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Simulating driveability using Dymola and Modelica

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Simulating driveability using Dymola and Modelica

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Abstract

As the complexity of automotive powertrains increases it is becoming increasingly difficult for engineers to determine the optimum specification for the system. The proliferation of control systems also adds to the complexity of the task and increasingly simulation is being used to assist in the development of new products.

A model has been developed to enable component parameter studies and desktop calibration of driveability attributes using Dymola and Modelica. The model included the automatic translation of an existing, validated engine model from Simulink into Modelica using Simelica. The correlation process used is described and the results presented.

In addition, a study of the effects of differing levels of model complexity has been conducted to understand the trade-off between simulation accuracy and performance.

1. Introduction

Automotive manufacturers have a difficult task to reduce emissions, increase fuel economy and enhance the performance and driveability of new products while reducing the time and cost of development. Simultaneously these products are becoming increasingly complex adding to this already difficult task. Historically, the majority of the development has been conducted with extensive prototype vehicle testing. As the complexity of the products increases, additional testing is required and it becomes increasingly difficult to optimise the system.

Engineers are increasingly looking to simulation as a way to optimise the system and reduce the amount of testing required. Simulation has been used for many years to optimise fuel economy and performance but with the development of multi-domain modelling tools it is now possible to apply simulation to a number of other areas such as driveability.

In this paper the term driveability refers to the study of how the longitudinal acceleration response of the vehicle reacts to accelerator pedal inputs. The main area of study is the response during tip-in and tip-out manoeuvres. A tip-in manoeuvre is characterised by the vehicle initially either being driven at a constant speed or coasting down in gear and the driver rapidly applying the accelerator pedal. Similarly, a tip-out manoeuvre is characterised by the vehicle either in a state of acceleration or being driven at a constant speed and the driver rapidly releasing the accelerator resulting in the vehicle decelerating. It is not uncommon to find that the vehicle longitudinal acceleration oscillates in response to these manoeuvres and this is undesirable from the driver and passenger’s point of view.

In order to simulate driveability we need to incorporate the effects from many engineering domains. This means the model must include representation of the transient torque production of the engine, the behaviour of the engine control system, the behaviour of the entire transmission and driveline and the interaction with the tires and vehicle body. This requires the use of a multi-domain modelling tool, as we need to include effects spanning a number of engineering domains including control, fluid flow, thermodynamics, heat transfer and mechanics (1D rotational, linear and multibody).

In this project we have reused elements of existing simulation tools, which have been previously validated to simplify the development process. This has the added benefit that engineers are already familiar with these parts of the models and have confidence in them. We have reused elements from two different simulation tools and combined them in one environment to develop our driveability simulation capability.

A complete driveline model has been developed within the transmission and driveline department to study the vibrations within the drivetrain. These models have been developed using Dymola and include the effects of the driveline mounting system to the vehicle body structure.

The engine calibration teams have developed a Simulink engine and control system model that has been used to optimise fuel economy and emissions. By utilising this engine model in the driveability model we can quickly derive an engine model that simulates the effects we require. Using Simelica we can automatically translate this existing engine model from Simulink into Modelica and then couple this to the existing driveline models.

This project has utilised the Modelica VMA [1], see figure 1, which has allowed us to easily develop a number of driveline models that include different levels of detail. We can then easily swap between these to investigate the effect on the results. This enables us to fine tune the model for use in different applications, as we understand the trade off in simulation accuracy against computation time.

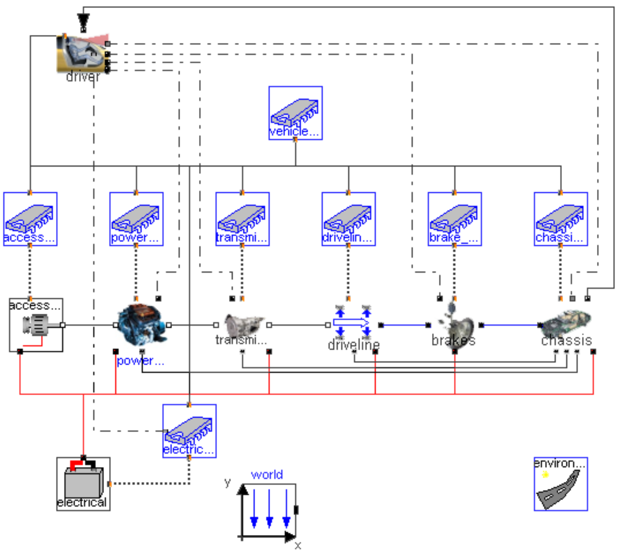

Figure 1: Top level of the Modelica VMA

2. Understanding Driveability

Driveability, in this context, is primarily about the excitation of the first torsional mode of the powertrain and is often referred to as the shuffle frequency [2]. This torsional mode is excited by the sudden changes in the engine torque that result from the accelerator pedal inputs. The driver and passengers feel the torsional mode as an oscillation in the longitudinal acceleration. The acceleration is typically measured using accelerometers mounted on the driver's seat track as this nominally equates to the acceleration experienced by the driver.

The assessment of driveability is primarily a subjective one carried out by engineers during testing. For simulation purposes it is necessary to utilise objective measures that can be used to assess whether a change in the hardware or calibration has improved or degraded the response. A typical acceleration trace due to a tip-in manoeuvre is shown in Figure 2. The key characteristics of this response that assist the engineer in judging whether the vehicle exhibits good or bad driveability behaviour are the peak overshoot, damping, and the frequency of oscillation. To tune a system to deliver good driveability on the desktop requires targets to be set for these objective measures.

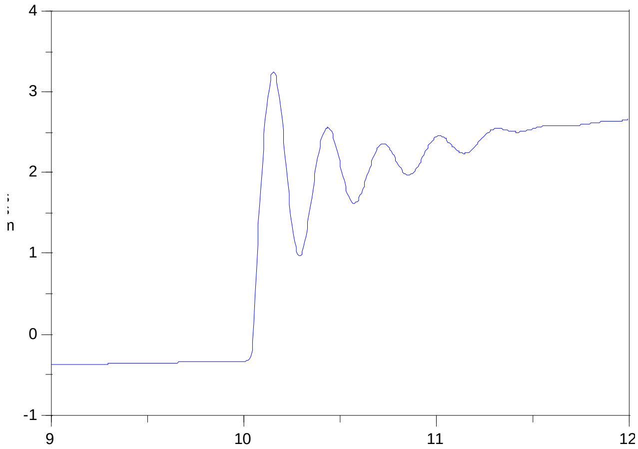

Figure 2: typical longitudinal acceleration due to a tip-in manoeuvre

The frequency of the first torsional mode is usually in the range of 2-10Hz and depends primarily on the flywheel inertia, gear ratio and driveline compliance [2]. As driveability is only concerned with the first torsional mode we can use a low order model and we can ignore effects in the model that occur at frequencies above 30Hz. This means we can use a mean-value engine model and do not need to model the torque cyclics as a function of crank angle. It also allows us to simplify the driveline model.

3. The Model

3.1 Engine and Engine Controller

The Engine and Controller models have been imported into Modelica from an existing Simulink model. This approach has been adopted for the engine and its controller as a suitable model has already been developed and validated in Simulink. This model has been used for a number of different simulation tasks and the development engineers have experience and confidence in this model.

Development engineers are usually sceptical of new methods and it takes a significant amount of time and effort to convince them that the new methods can be used reliably in place of extensive testing.

Re-using existing models that engineers have previous experience and confidence in eases the process of introducing new methods.

The engine model itself is a mean-value engine model that predicts the airflow into the engine and includes maps that characterise the torque output based on the airflow, air-fuel ratio, spark timing and engine operating temperatures. The controller model is an accurate model of the actual controller rather than an idealised or simplified version. It includes the functions to control the spark timing and fuel injection.

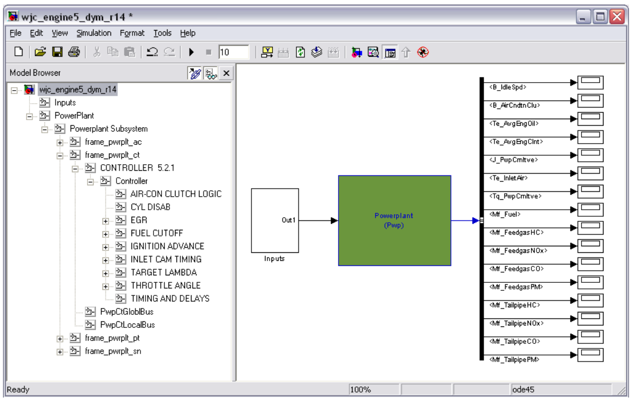

Figure 3: Simulink model created for automatic translation

The Simulink model of the engine and controller was developed to work within the Simulink version of the VMA [3]. The Modelica VMA is based off the Simulink VMA and uses the same system decomposition. The Simulink VMA makes extensive use of signal buses to pass signals between the various subsystems and this approach has been modified in the Modelica VMA to make use of the physical connectors available in Modelica. To enable the automatic conversion of the Simulink model into Modelica using Simelica [4] we first had to extract the engine and controller blocks from the VMA and create a new model that includes a set of constants as the inputs to a dummy bus system, see figure 3.

The translation process is straightforward and Simelica generates a Modelica representation based on the AdvancedBlocks library [4]. To ensure the correct translation of the model it is necessary to import the parameter data into Simelica as well as the Simulink model. Figure 4 shows Simelica with the engine and controller model open. On the left hand side of the window we see a hierarchical view of the model and any block that may potential create a problem during the conversion will be identified in the list on the right hand side of the window.

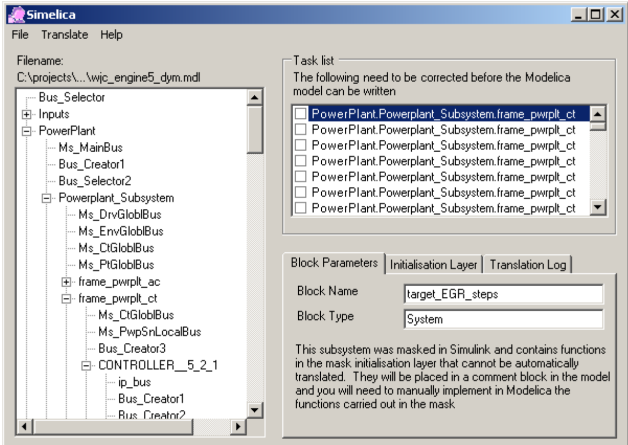

Figure 4: Simelica view of the Simulink engine model during translation

A number of potential problem blocks are identified within the Simulink model although none of them actually cause an issue. The reason they are identified as potential problems is that they are masked subsystems that include some initialisation commands. These are potential problems as the initialisation commands can be any Matlab statement including user-defined functions. It is not possible for the translator to automatically handle all of these so they are identified for the user to review and check in the translated model. The initialisation commands in the identified masked subsystems are actually being used to calculate some additional parameters and can be automatically translated.

3.2 Transmission Model

The gearbox is modelled as a referred torsional system. The stiffness, damping, inertia and backlash are all referred back to the input shaft and modelled at one point. Although this approach does produce the overall correct mathematical effect it doesn't allow us to study the detail of how the gearbox internals respond during a driveability manoeuvre.

Ideally the gearbox model would include all the gears and shafts with the inertias, stiffness and lash data being applied at the appropriate points in the system. Unfortunately the data required to develop a model with this amount of detail was unavailable for the test vehicle.

3.3 Driveline Model

Driveability is affected by a large number of components including the propshafts, halfshafts, differential, driveline mounting system, vehicle suspension and tires. A number of different driveline and chassis models have been developed so that we can investigate which of these effects has the greatest impact on the accuracy of results and simulation performance.

In its simplest form, the driveline model includes the propshaft, differential and halfshaft but does not include the driveline mounting system. The actual propshaft installation in the test vehicle comprises of a rubber coupling at the end of a two-piece propshaft connected by a universal joint. In the simplest model this is reduced to two inertia elements connected by a linear spring-damper and the overall effective inertia, stiffness and damping of the propshaft is calculated from the values of the individual component values.

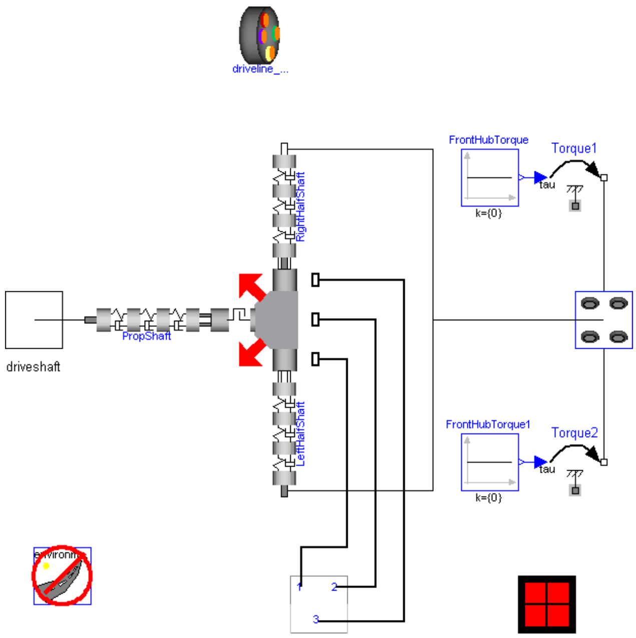

Figure 5: Complex Driveline model

A step up from the simple driveline models is to include the driveline mounting system. Within the driveline subsystem this involves modifying the differential model to include the reactions at the bearings that act on the differential case. Using a MultiBody model of the differential these reactions are translated into forces acting on the differential mounts. The actual mounts are included in the chassis subsystem.

The reaction of the driveline torques on the differential case is modelled by coupling the 1D driveline models to a MultiBody representation of the differential, see figure 6.

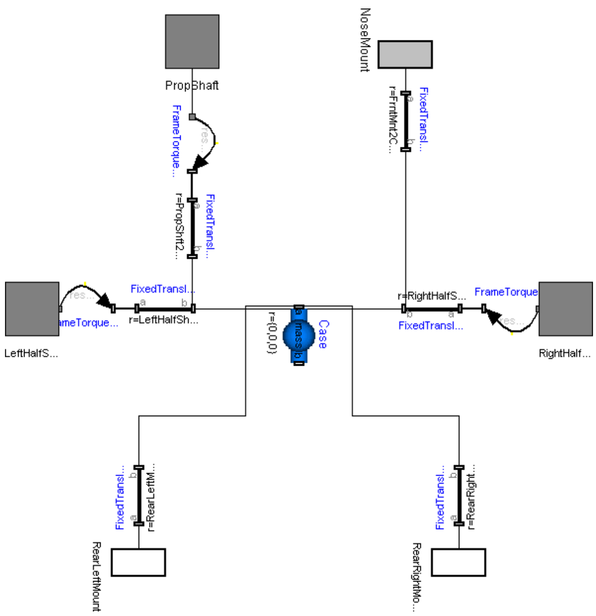

Figure 6: Differential model

3.4 *Chassis Model*

The chassis model includes the vehicle body, suspension, tires and, in the complex models, the driveline mounting system, see figure 7. The vehicle body and suspension models take account of pitch and bounce in response to the acceleration of the vehicle body to calculate the load acting on each tire. A Pacejka tire slip model is used to account for the variation in vertical load and the effect that this has on longitudinal tire slip.

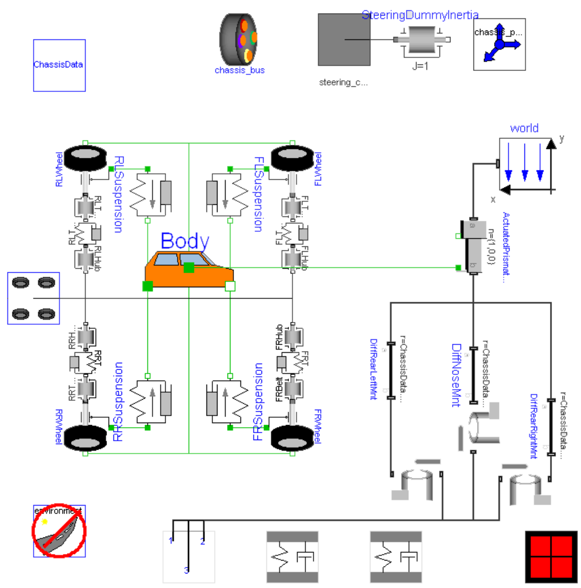

Figure 7: **Chassis model**

The basic chassis model can be extended and the differential mounting system added using the MultiBody library. The mount models use different non-linear stiffness and damping values in the x, y and z directions and include a spherical joint at the interface to the differential case.

3.5 The Driver model

The driver model has a number of tasks to perform to enable the simulation of the driveability tests we are interested in. The driver has to control the vehicle to an initial speed and after the vehicle has stabilized at this speed the driver has to control the accelerator pedal to replicate a tip-in or tip-out test. See figure 8 for example traces of the accelerator pedal position during these tests. All of the tests are conducted in a fixed gear and without the use of the brakes.

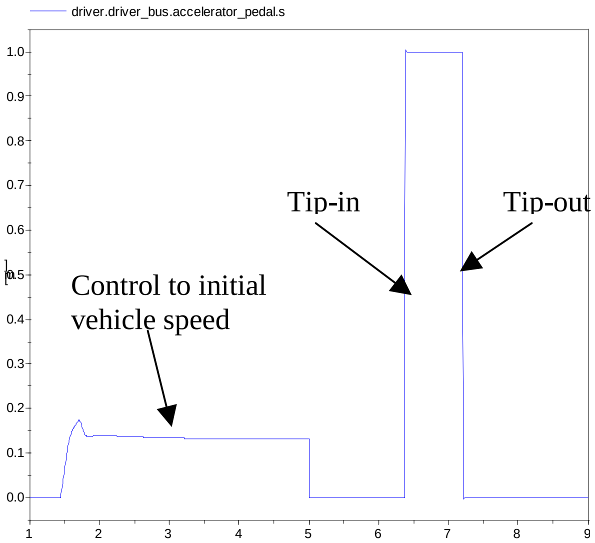

Figure 8: Accelerator pedal profiles

3.6 The Rest of the model

The remainder of the model is simple as it only has a small impact on our area of study. The accessories subsystem is modelled as a series of torque losses and inertias representing the major components (alternator, power steering pump, air-conditioning unit).

The brakes subsystem includes individual wheel brakes but none of the actual braking system is modelled. The position of the brake pedal as controlled by the driver is multiplied by a scale factor to enable an appropriate amount of braking force to be exerted when required.

4. Correlation

4.1 The approach

The correlation of the model is achieved using the most complex versions of each of the subsystem models. As we have such a complex model the correlation has been approached on a step-by-step basis, which enables the correlation of different parts of the model at each step.

The first step is to ensure that all the subsystem models have undergone some degree of correlation. This step can be used to ensure that each subsystem is behaving in a physically correct manner and that, where possible, the subsystem is correlated in isolation to appropriate test data. For example, we can correlate the engine model to data collected on an engine dynamometer.

The whole vehicle model can then be correlated in a number of steps to help ensure that the overall system performance is correct. The correlation exercise was divided into a number of simple steps, simplest first, and gradually increasing the number of systems until the full vehicle model undergoing a tip-in test was correlated.

4.2 Coastdown correlation

The first test was to compare the model to a coastdown test from high speed (>120kph) with the vehicle in neutral. Correlating the model to this test enables us to confirm that the parameters for the aerodynamics, tire rolling resistance, driveline losses and total vehicle inertias are consistent with the measured vehicle. Figure 9 shows the results of this stage of the correlation;. Due to experimental variation it is essential that a number of real tests are carried out and that the model correlates to fit within these results rather than to match an individual test exactly.

Once correlation of the coastdown with the vehicle in neutral is achieved, we can start to include additional effects into the correlation. The next step is to look at the in-gear coastdowns as this step enables us to verify that the gearbox loss models and engine braking effects are consistent with the test vehicle. As the engine braking effect should have been correlated as part of the Simulink engine model correlation we can focus on the gearbox loses if there is a discrepancy between the test and simulation.

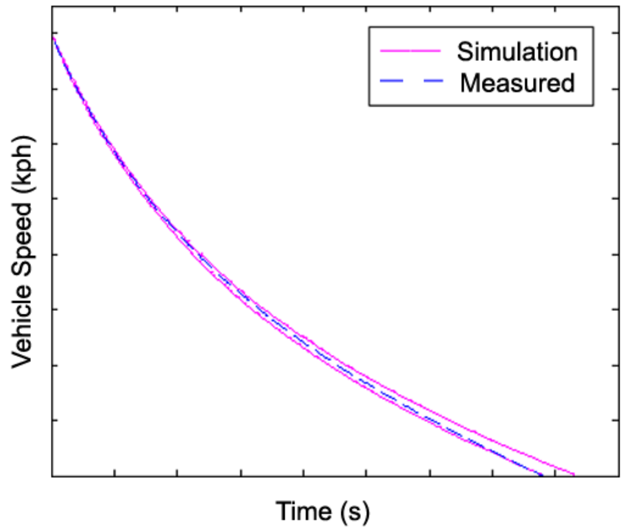

Figure 9: Coastdown Correlation

4.3 Acceleration correlation

Acceleration tests enable confirmation that the engine torque, gearbox, driveline losses and total inertias are consistent with the test vehicle. In this case discrepancies between the model and the test results have been observed, see figure 10.

It is difficult to identify the root cause, but the most likely candidate was differences in the predicted and actual engine torque. This is a result of the engine installed in the test vehicle having unknown characteristics and the original engine model having been correlated against a different engine installed on a dynamometer. Other possible causes of the discrepancy are the loss models in the engine, gearbox and auxiliaries (alternator, power steering pump, etc).

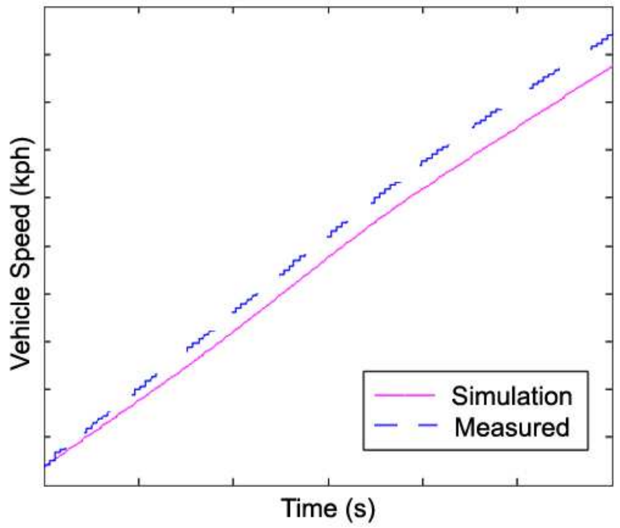

Figure 10: Acceleration Correlation

4.4 Tip-in correlation

There are two types of tip-in test that the model has been correlated against. The simpler of the two to correlate starts with the vehicle moving at a constant speed, the other with the vehicle coasting down in gear. When correlating the model against the tip-in test from a constant speed we are seeing the response of the vehicle longitudinal acceleration being influenced by the stiffness and damping within the entire drivetrain. The test from the coastdown condition adds in the effect of the backlash within the gearbox and driveline.

Achieving good correlation to the tip-in tests demonstrates that the gearbox and driveline stiffness, damping and backlash values are consistent with the test vehicle and that the model is suitable for the development of driveline components and engine control strategies and calibration. By correlating the results across a range of different engine speeds and in different gears we can establish confidence that the response of the model is accurate across a wide range of conditions.

Figure 11 compares the longitudinal acceleration of the vehicle for a number of tip-in tests at different engine speeds in second gear. Overall, there is a good correlation between the simulation and test results across this range of engine speeds. There is a slight discrepancy in the results and the model appears to be over-damped when compared to the test vehicle, as the oscillations in longitudinal acceleration after the first peak are smaller than measured in the vehicle.

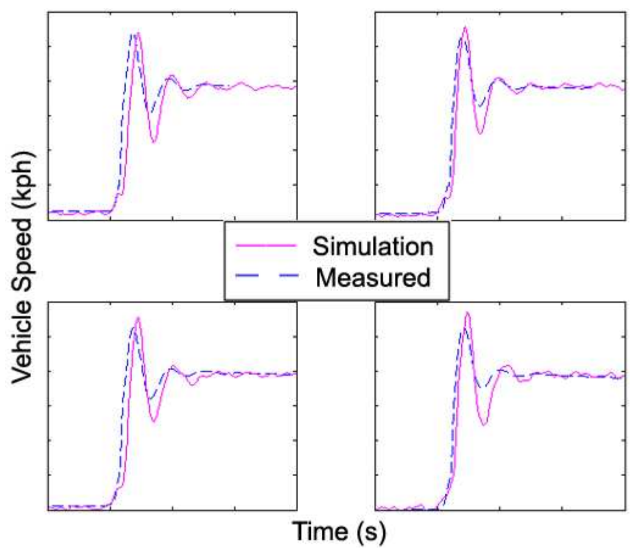

Figure 11: Correlation of tip-in results at various engine speeds in second gear

5. Model detail study

Due to concerns about the computation time required to accurately simulate driveability an investigation into the required level of model complexity has been carried out. Using the various different driveline and chassis models we have developed, the trade-off between simulation accuracy and computation time can be studied. Table 1 summarises the different details that are included in the different driveline and chassis subsystem models.

	Basic	Medium	Complex	Extreme
Propshaft – Driveline Subsystem				
1D single element	Y			
1D one element per shaft		Y	Y	
1D multi element per shaft				Y
Halfshaft – Driveline Subsystem				
1D Single element	Y	Y	Y	
1D multi-element				Y
Differential – Driveline Subsystem				
1D	Y	Y		
MultiBody			Y	Y
Chassis Subsystem				
Half-car body model	Y	Y	Y	Y
Non-Linear diff mounts			Y	Y

Table 1: Summary of model detail

Figure 12 compares the results of running each version of the model through the same tip-in test. By increasing the complexity of the model we improve the correlation of the model but at the cost of computation time as shown in table 2.

The results for the complex and extreme models are almost entirely coincident apart from some very small deviations not visible at this scale. As the extreme version of the model takes over two and a half times as long to simulate as the complex version and the results produced are almost exactly the same as generated by the complex model we can drop the extreme version of the model from further comparisons. Comparing the results produced by the complex model with our test results shows a very good agreement during the tip-in test as discussed in section 4.

The metrics shown in table 2 can be used to help objectively judge how close the correlation is. The peak overshoot is a measurement of how much bigger the first peak is than the acceleration achieved after the oscillation has settled. The settling time is the time taken from the tip-in event at the throttle for the vehicle acceleration to settle to within ±2% of the acceleration achieved after the oscillation has settled.

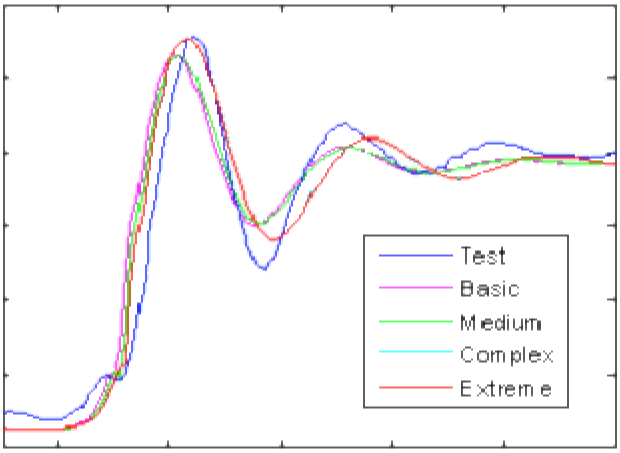

Figure 12: Comparing model detail levels

	Simulation Time (s)	Peak Overshoot (% of ss accel)	Settling Time (s)
Basic	133	48.9	1.04
Medium	133	49.2	1.05
Complex	512	57.5	1.12
Extreme	1370	57.5	1.12
Test	n/a	58.5	1.11

Tests carried out using a PC (AMD64 3800+, 2GB RAM) running Windows XP

Table 2: Simulation time vs. model detail

The complex model achieves a good correlation but the simpler models still achieve a reasonable level of accuracy. As these simpler models take one quarter of the time to simulate than that taken by the complex model it would be extremely useful to be able to use these versions for conducting extensive sensitivity studies. By maintaining two versions of the model we can use the simple model to quickly identify a range of suitable values for a model parameter and then apply the complex model to determine the optimum value.

6. Conclusions

Robust models have been developed and correlated for the simulation of driveability. These can now be used for the desktop calibration of many engine controller functions and to carry out parameter studies to develop the specifications for the driveline components.

The use of Simelica has enabled us to easily reuse models that have been developed in Simulink. The translation process is entirely automatic ensuring that we can easily update our Modelica model as the Simulink models are developed further.

The sensitivity study has enabled us to judge which versions of the model are suitable for which tasks. For example, to carry out a large parameter sweep to help target further effort we can use the basic model as it gives an adequate level of accuracy and quick simulation times. To fine-tune a calibration we need to use the complex model as this provides a very good correlation but takes longer to simulate.

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