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MODELLING OF 4WD VEHICLE DRIVEABILITY DURING TIP-IN/TIP-OUT EVENTS

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Abstract

This paper describes a modelling method to investigate the dynamic behaviour of 4WD vehicle under a severe driving condition, where the driver applies a rapid tip-in on the accelerator pedal in 2nd gear to achieve maximum engine torque. This is followed by a tip-out event by releasing the accelerator quickly.

The Tip-In/Tip-Out events are one of important elements to assess the vehicle driveability. During these test events, the vehicle is expected to generate low frequency vibration between 2 Hz and 10 Hz and gives discomfort feelings induced by resonance effects on sensitive human organs. The aim of this paper is to develop a 4WD vehicle model in a modern object-oriented multi-body simulation tool and study its driveability.

1. INTRODUCTION

Not many evidences have been seen on the research carried out to examine the eminence of the vibration response in 4WD vehicle system predominantly in the Tip-In/Tip-Out events. The basic test method is to study the interaction between the driver and vehicle in terms of driveability at low speed events. The results can be analysed to identify the root cause of the low frequency responses. From the past investigation [1], the exposure of low frequencies to the driver leads to a sense of discomfort induced by resonance effects on different human organs such as upper torso, lower abdomen and shoulders.

The previous effort [2] that influenced this work has resulted in the development of a single Torsional Model, a single Fore-Aft Model using the ADAMS modelling environment to study the above behaviours. Both models have been correlated with the ADAMS full nonlinear model as a reference model and the vehicle testing data. The development of these single models was due to the long computation time of the ADAMS full nonlinear model. Furthermore, it was not possible to use the full nonlinear model for a real time application due to the complexity of its architecture, which consisted of more than 550 Degree of Freedom (DoF). One aim of the work was to accelerate the process of tuning the properties of the system components to improve the vehicle driveability. However, these two models were not connected to represent the actual physical behavior of the vehicle system due to the lack of tyre model. Hence, it did not demonstrate the actual interaction between these two sub systems. In addition, there were limitations to the development of control strategies in real time using ADAMS. This is due to long computation time and the ability to host the model in a real time platform.

Considering the above factors, Dymola modelling platform [3] has been selected to develop the model of vehicle system based on the open-source Modelica language. For this research, the vehicle model was constructed from standard and extended Modelica libraries by setting up the vehicle parameters. The model was also integrated in a MATLAB/Simulink environment and implemented in the real time platform, which allows the development of control strategies.

From this experimental result, the vehicle model characteristics at low frequencies have been derived to give the prediction of the vehicle driveability behavior. In the remaining of this paper, the modelling methods and modelling details are presented in the next topics, which explained the results of simulation in Dymola, real time simulation and conclusion of this research.

2. MODELLING

The architecture of the vehicle model was based on a 4x4 platform with 2.2L Diesel engine mounted in East-West orientation. It consists of all drive-train subsystems and the chassis components acting on longitudinal direction.

The physical characteristics of the components were represented by combining the torsional and fore-aft elements with the correct mechanical properties. The interaction of the drive-train and chassis systems was captured by a nonlinear tyre model in longitudinal direction, which was derived from simulation data to give a true behavior of the tractive force on different road surfaces.

pitch and yaw to the vehicle system dynamics. Similarly to Torsional Model, no tyre model has been included to represent dynamics characteristics of the wheels, i.e. tractive force, slip ratio. The characteristics have been replaced by the forces taken from vehicle test data and used as the inputs to the front and rear wheels to excite the system.

2.3 Coupled Model

The architecture of the Coupled Model consists of 40 DoF from 4 sub-systems and driver environment, . Each component of the sub-systems has the combination of torsional elements, masses, inertias and mechanical properties. From *Figure 3*, the engine is mounted to the transmission bell housing and the energy from the combustion is transferred to input shaft via DMF. The integrated front differential inside the transmission splits the torque to the front half-shafts and Power Take Off (PTO) unit. The rear differential then divides the torque to right and left side of the rear half-shafts. Both of the front and rear half-shafts are connected to the tyre model to provide the longitudinal force as a function of vertical load and wheel slip ratio. The longitudinal force drives the chassis sub-system where each of the chassis components is inter-connected by the nonlinear compliances.

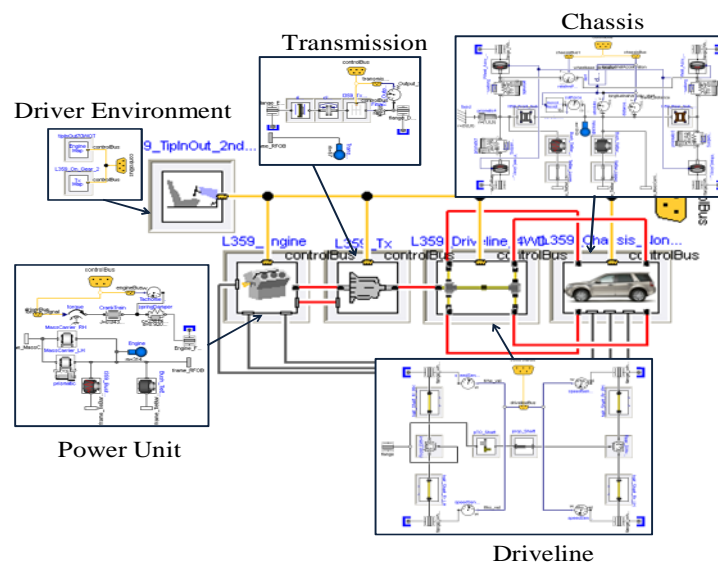


Figure 3: Coupled Model of 4WD Vehicle System

A generic tyre model has been developed based on the standard tyre parameters fitted on this particular vehicle. Car Maker software [5] was used to generate the 3 dimensional data, on tarmac surface with surface coefficient, $\mu = 0.85$. Fundamentally, the road surface coefficient is reduced as a function of vehicle speed; hence reduce the tractive force further. But this effect has been neglected in this analytical model. The input torque is controlled by the driver environment block to represent the combustion process to induce the crank train system. The gear ratio is fixed at 2nd Gear with the assumption that no mechanical losses in the gearing system.

By coupling the two single models as discussed earlier, the vehicle model was expected to be more robust and produce true physical representation of the sub-system behaviour. The common root causes of vehicle driveability were always associated with the rise rate of the torque demand as well as the overshoot of the oscillation, which can be sensed by most drivers [6, 7]. The explanation for this occurrence was due to the rapid excitation of torque demand to the driveline system and its compliances. With this Coupled Model, it has allowed more accurate study on driveability and to understand the energy flow from throttle demand to the seat rail via contact patch of the wheels.

To validate the accuracy and robustness of the Dymola Coupled Model, a correlation was carried out with an ADAMS full nonlinear model and had been verified with vehicle measurement data. The simulation parameters were set-up to be the same as the ADAMS full nonlinear model.

3. SIMULATION

3.1 Off-line Simulation

The standard practices of the Tip-In/Tip-Out tests are to accelerate the vehicle up to 45 kilometer per hour (kph) and coast down to 30 kph. An abrupt torque then applied within 0.12 seconds to excite the torsional system until it reached maximum torque. And finally the torque will be removed instantaneously from the crank train system. For the simulation set-up, the model was run up to 17 seconds to replicate the test condition above, with fixed step integrator of 0.001 seconds. The Euler solver was used together with Implicit Euler as the inline integration for real time solver.

Based on the single Dymola sub-systems simulation, the correlation results show a good agreement with the single ADAMS models in terms of crank speed and seat acceleration as presented in *Figure 4* and *Figure 5*. These results verified that the single Dymola Torsional Model and Fore-Aft Model can be used to form the Coupled Model. For the Coupled Model, the computation time against the ADAMS full nonlinear system has improved significantly from 9 hours 30 minutes to only 10.5 seconds. In *Figure 6*, the crank speed is having the same behavior as full nonlinear model but the oscillation is seen been phased in 2nd to 4th orders. However, these discrepancies can be neglected as the occurrence was due to the effect of the generic tyre model properties. The correlation result demonstrates that the Coupled Model has performed the same manner as the ADAMS full nonlinear model.

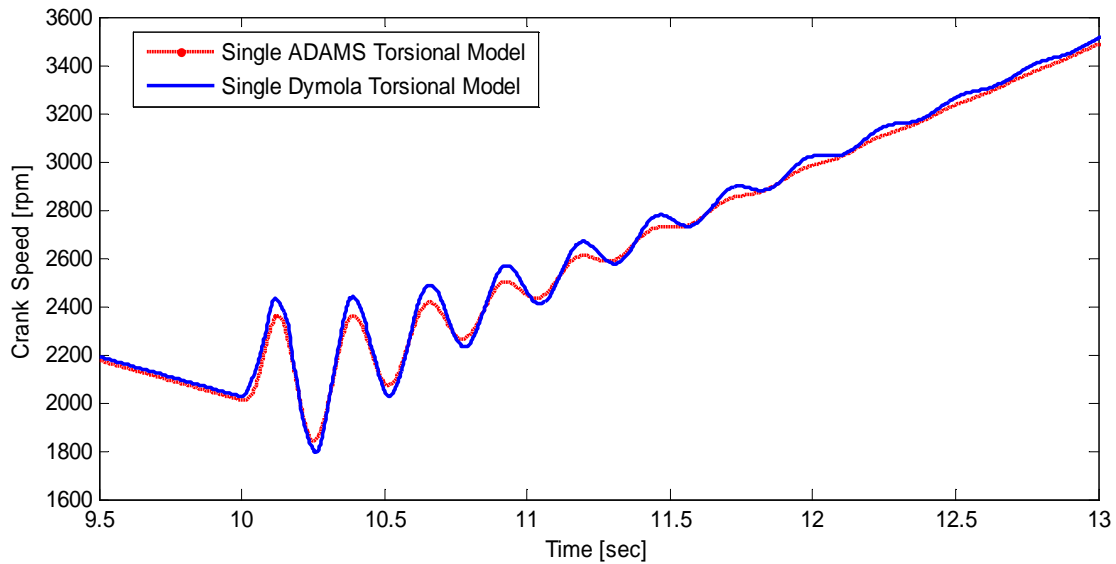


Figure 4: Correlation results against single ADAMS Torsional Model without tyre model

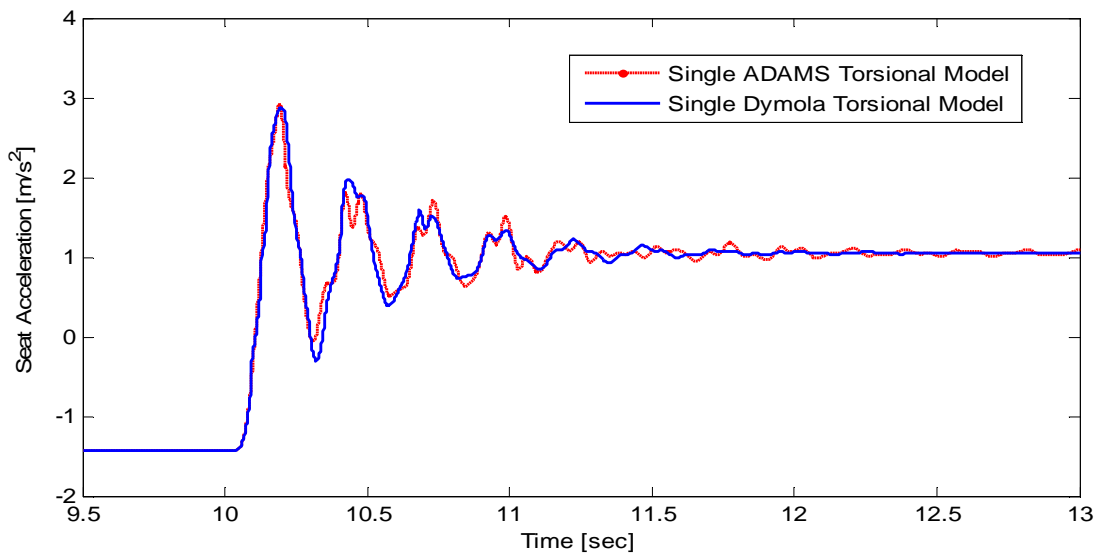


Figure 5: Correlation results against single ADAMS Fore-Aft Model without tyre model

The next stage of the simulation for the Coupled Model was to conduct the Tip-In/Tip-Out events and observed the response of the system based on 70% throttle position at 2nd Gear. *Figure 7* shows that the system generates shuffle frequencies of 3.42 Hz during Tip-In and 3.14 Hz during Tip-Out with exposure time of 2 seconds measuring at the seat rail. As discussed earlier, these frequencies range (2 Hz - 10 Hz) will stimulate the human organs that can cause the motion sickness. These shuffle frequencies were dominated by the half-shafts [4] and can be tuned by changing its mechanical properties. The overshoots of the shuffle frequencies, which attributes to the customer perception of vehicle driveability, exceed more than 85% for both events as shown in *Table 1*.

From the graph, the rise rates have also been calculated during Tip-In and Tip-Out to characterize the vehicle driveability. The Tip-In event has generated higher value of rise rate at 3.62 g/s as the result of the sudden excitation of the driveline system from steady state condition. When torque is removed instantaneously from the system, the rise rate on the seat acceleration is reduced by 30% compare to Tip-In event due to the nonlinearity and asymmetric of the system compliances. With these two outputs, it is possible to develop the driveability matrix in the next phase of the research to assess the vehicle behaviour acting on the transient torque response as well as developing the control strategies to damp the shuffle frequencies.

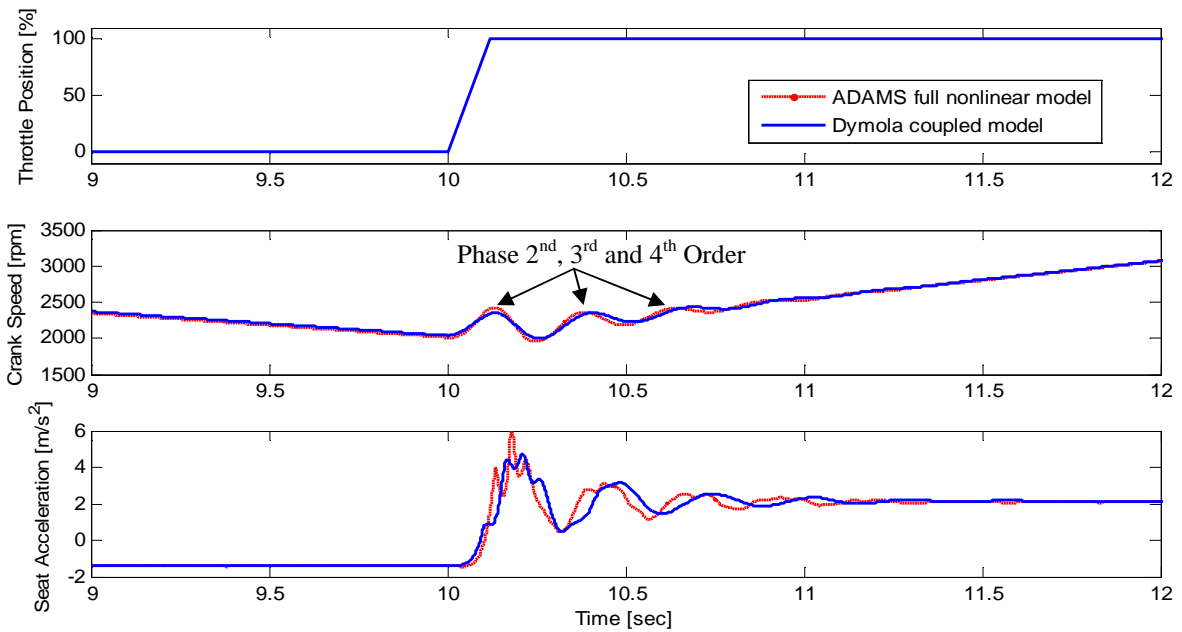


Figure 6: Correlation between Dymola Coupled Model and ADAMS full nonlinear model with Throttle Input 100% at 2nd Gear, surface coefficient, $\mu = 0.85$

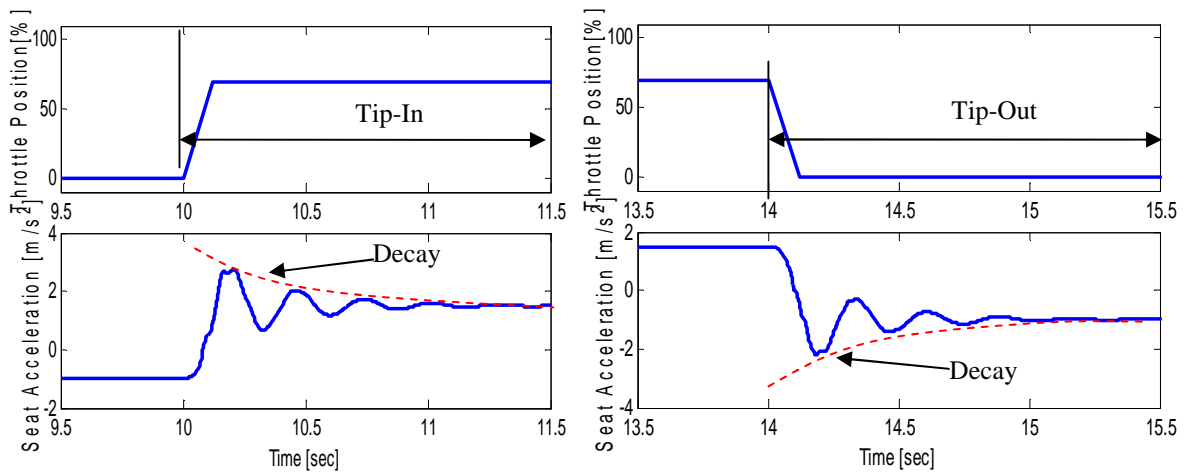


Figure 7: Tip-In/Tip-Out with Throttle Position 70% at 2nd Gear, $\mu = 0.85$

	Tip-In Event	Tip-Out Event
Damping Ratio, ζ	0.0366	0.0393
Natural Frequency, Hz	3.42	3.14
Overshoot, %	89.1	88.4
Rise rate, g/s	3.62	-2.36

Table 1: Overshoot and Rise Rate during Tip-In/ Tip-Out with Throttle Position 70% at 2nd Gear, $\mu = 0.85$

3.2 Real Time Simulation

For the future control strategies development, it is essential to see how the vehicle reacts in real time and to validate the response of the vehicle sub-system as close as the actual condition. From the real time simulation, the model has performed the same behaviour as the Dymola Coupled Model. The computation time for this model is 0.5 milliseconds to process every 1 millisecond of sampling time with no overrun. It indicates that the coupled vehicle model can be used in real time simulation for the control development.

4. CONCLUSIONS

The Coupled Model of 4WD vehicle system built in Dymola has performed equally as good as the ADAMS full nonlinear system. The model has produced frequencies that can affect the human organs during Tip-In/Tip-Out events. The oscillation is mainly dominated by the half shafts, where these frequencies can be shifted by tuning the properties of stiffness and damping. The simulation time has also reduced from 9 hours 30 minutes with an ADAMS full nonlinear model to 10.5 seconds with the Coupled Model with equally valid outputs. The Coupled Model also performs well in a real time test environment as seen by the measurement of the computation time to give a prediction of the vehicle behaviour during torque transient mode.

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