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DOTTORATO DI RICERCA IN
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CICLO XXIX

Model-in-loop development and experimental assessments on noise and vibration effects for Hybrid powertrain.

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MOTIVATION

The automotive industry receives direct and indirect stimuli (by European legislation and public demand) to reduce carbon dioxide emissions, fulfil noise regulations and to improve fuel efficiency and passive safety for production vehicles. Strict requirements and ambitious targets for green, silent and safe mobility need interdisciplinary solution approaches in various fields. However, modern vehicle industries are very much dependent on the skill of engineers to timely introduce new methods and technologies in the development of new products. Current university based education is focusing on classical fields like mechanical engineering with “NVH” or “Light Weight Design” aspects or on the other side on electrical engineering covering “Electrification” & “Hybridisation”. Future challenges in automotive development (e.g. E-mobility) require a more interdisciplinary approach in the education of future researchers and engineers.

Thus, engineers can capture the requirements of conflicting demands and help for the development of ‘green’ products. The GRESIMO project (Best Training for Green and Silent Mobility) aims to bring together early career researchers and experienced colleagues, from across NVH, Light Weight Design (LWD) & Hybridisation/Electrification (H/E) disciplines, and from a broad range of professional backgrounds. The focus of GRESIMO was to motivate and encourage early stage researchers for scientific work in the new interdisciplinary fields between the 3 mentioned research areas as shown below in Figure 1. The main goal of project consortium was to provide high level education, training facilities and technical supervision for PhD fellows in these new disciplinary fields between NVH, LWD & H/E. (ITN GRESIMO Annexure I, 2011)

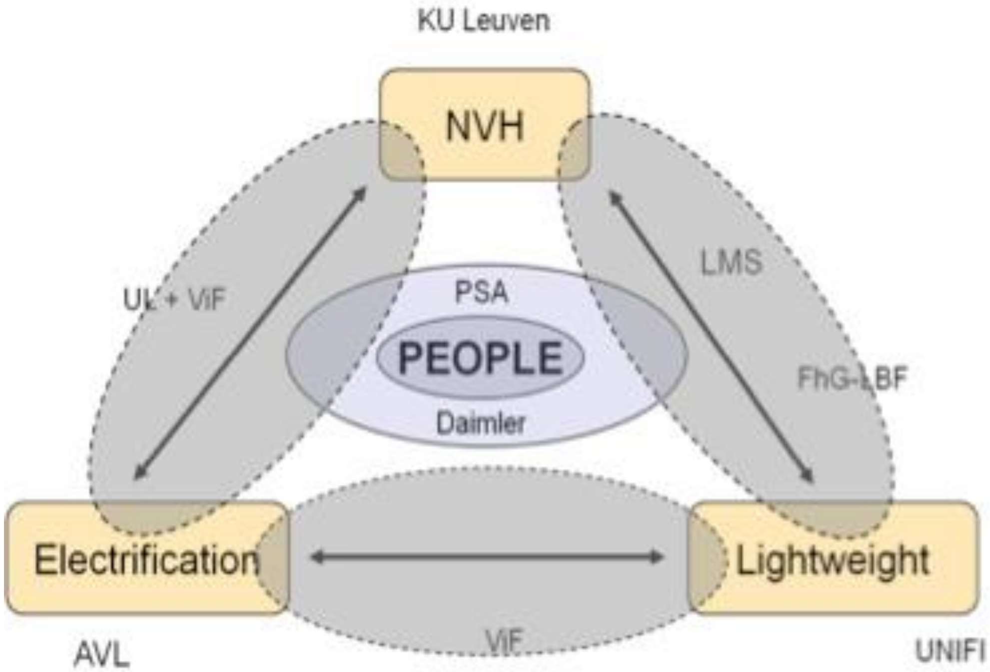

Figure 1: Established & interdisciplinary research areas (ITN GRESIMO Annexure I, 2011)

Thus, this thesis is part of GRESIMO project for advanced research methodology, whose support was to provide researcher a specific education in theoretical as well as practical training i.e. in one or two of the mentioned interdisciplinary fields. The two interdisciplinary sectors which were considered within this work are Hybridisation/Electrification (H/E) and Noise Vibration and Harshness (NVH). However the present work describes and summarizes the methods and models that are applied and developed within the various tasks during the GRESIMO project for scientific research.

ABSTRACT

The increasing use of virtual technology is considered as a great support for the real world in terms of developing and analyzing for the better products. Moreover today for finer vehicle’s in the automotive industry, the simulation platform is the base of research and development phase. Universally industries use various modelling approach for better understanding and further explore in the automotive systems. However, there are different vehicle classification within industry and one of the leading vehicle system configuration is Hybrid Electric Vehicle (HEV), which has by far shown a great attraction overall.

The following thesis presents the analysis effort made within the simulation methods and models to further investigate the noise and vibration effects for hybrid powertrain. During the complete research work, three major assessments were carried out:

1. The first computational model represents the co-simulation of complete vehicle, where multi-physics simulation approach is carried out to principally understand the noise and vibration effect for hybrid powertrain. A special emphasis is made on combining simulation methods and models for achieving non-stationary conditions, where the requirement is for complete vehicle model.

2. The second model concentrates on evaluation of seat vibration with respect to human perception and based on the transient operating conditions, i.e. specifically focus on start-stop systems of micro / mild hybrid powertrain. A detailed analysis model was build to study the starting behaviour of hybrid configuration named - Crankshaft Starter Generator (CSG) and further understand its effects on human factors.

3. The third and final model deals with interpretation of earlier research work i.e. combining the complete vehicle model and start-stop system to simulate the complex transient events in a driveline. Furthermore the simulation model is more focused on the operation based, where stop and go scenario is analyzed with its effects on transmission dynamics.

The prime target for this research is to elaborate and define the modelling approaches and methods for electro-hybrid powertrains and their incorporation in simulation software. However, these multi-physics simulation models are used to investigate different types of applications and to further analyse the noise and vibration effects for transient operating conditions.

Moreover, the outcomes of the research work have achieved a thorough study and analysis of noise and vibration effects (N&V) for the dynamic conditions of electro-hybrid vehicle powertrain. In addition to it for the future aspects, this work can be used to optimize and test the profound models with the real world application and validate with measured data to scrutinize noise and vibration effects of hybrid powertrain.

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1 INTRODUCTION.

1.1 Literature overview.

Internal combustion engines have been used immensely for a century as a power source of road vehicles. However, in the same period vehicle ownership increased to a level that finite fossil fuel resources and environmental and health impact of the emissions have become great concern for a few decades. To overcome this concern, various new trends have emerged into market such as use of alternate fuel for vehicles, Fuel-Cell technology, Plug-in-Hybrid Electric Vehicle, Electric Vehicle and Hybrid Electric Vehicle.

In this case, HEV (Hybrid Electric Vehicle) has dominated the list with its deep history, as it all began in the year 1899, where the first hybrid vehicle (Lohner-Porsche Mixte Hybrid) was developed by Ferdinand Porsche. With the increasing adoption and usage of dual technologies, there is also a growing recognition of the attributes related to this dual technology system such as dynamics, drivability and refinement that can have an adverse effect on customer acceptance. There are a number of new challenges associated with this refinement, i.e. in particular with their sound quality. [1, 2]

Moreover, HEVs are well known as smooth and silent drives, but due to increase in complexity, additional vibro-acoustic effects may concern the design of vehicle behaviour. To overcome these issues, significant amount of research and development is carried out in the field of modelling, simulation and control of hybrid powertrain systems. Modelling and simulation play a crucial role for the improvement of hybrid powertrains, since design validation by hardware measurement is costly. [3] In addition, for past 30 years, simulation of the noise and vibration behaviour of automotive drivelines became an integral part of the powertrain development process. Considering the fact that the N&V of Hybrid vehicles, which are different from conventional vehicles. Below Figure 2 shows the overview of the NVH effects in Hybrid powertrain model.

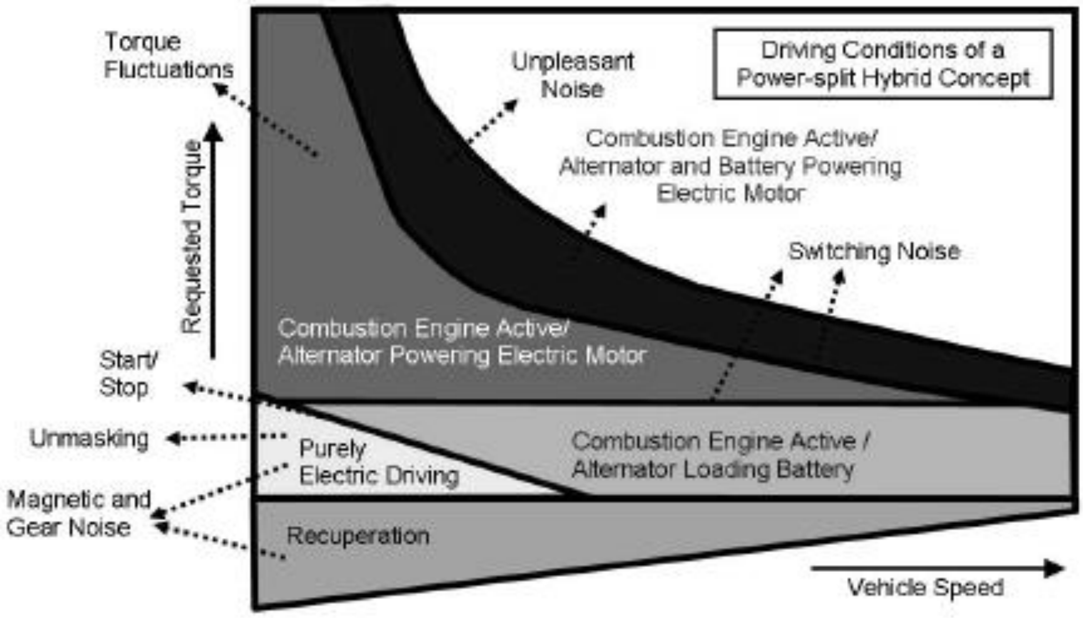

Figure 2: Noise and Vibration phenomena in different driving condition (Georg Eisele et al., 2006)

Furthermore, vibrations related to human body are again a major concern for many centuries, as it effects, not only physically but mentally, too. Vibrations have a broad range of exposure related to different areas like engineering, ergonomics, medicines etc. Each area has its own method for assessing the magnitudes. However, when it comes to engineering assessments, oscillation relies mostly on magnitude of acceleration. The current thesis also targets at Whole-Body Vibration (WBV) and its related issues. Whole-Body Vibration occurs when the body is supported on a vibrating surface.

As described by Griffin, [4] there are three principal situations: sitting on a vibrating seat, standing on a vibrating floor, or lying on a vibrating bed. To predict, measure and compare these vibrations different approaches were developed and International Standard Organization (ISO) and Journals have been established devoting this subject. However, for an evaluation of whole body vibration ISO 2631-1 is considered as a major reference for many authors. On this due consideration, the simulation conducted in the present thesis follows the method as defined in ISO 2631-1 (1997) and by M. J. Griffin.

In order to model and predict potential vibration problems and solutions, dynamic analysis are majorly preferred. The dynamics of such a system are often governed by complex relationships resulting from the relative motion and joint forces acting between the components of the system. Due to system complexity and inclusion of relatively many bodies within the system, the computational mathematics is a faster and classical approach for clarification. These numerical methods are principally based on analytical techniques of Newtonian, D'Alembert and Lagrangian mechanics. As these method leads us to a set of differential equations that can be expressed in matrix format and can further be solved using numerical algorithms. [5]

2 DYNAMIC EFFECTS OF COMPLETE VEHICLE MODEL

2.1 Introduction and overview

With current and future HEVs (Hybrid-Electrical Vehicles), additional phenomena and effects have entered the scene and need to be taken into account during layout/design as well as optimization phase. Beside effects directly associated with the e-components (namely electric whistle and whine), torque changes caused by activation/deactivation of the e-machine give rise to vibration issues (e.g. driveline shuffle or clonk) as well. This is in particular true for transient operation conditions like boosting and recuperation. Moreover, aspects of starting the Internal Combustion Engine (ICE) using the built-in e-machine in conjunction with the dynamic behaviour of torsional decoupling devices become increasingly important.

In order to cope with above-mentioned effects a multi-physics simulation approach is required. The following assessment proposes a simulation approach that incorporates the domains of the ICE thermodynamics, the mechanical driveline system, the electric components, the vehicle, as well as the fundamental control functions. A special emphasis is put onto non-stationary transient operation, which requires a full coupling between the involved domains. Moreover, the aspect of a combined 1D/3D mechanical modelling is outlined, with the background of scaling model fidelity for components of particular interest and importance (e.g. Dual Mass Flywheel, Centrifugal Pendulum Vibration Absorber, and transmission gear stages).

The models and results presented in this computation were widely based on research activities with respect to the coupling of the different involved domains such as:

- The combustion engine (incorporating its fundamental thermodynamics),
- the transmission and driveline,
- the vehicle,
- the e-machine,
- the (transient) driving/operating scenario,
- As well as basic control mechanisms.

A particular focus is put onto aspects of powertrain induced interior noise and vibration comfort. The emphasis of this investigation is mainly to outline the methodology and feasibility of the proposed multi-domain simulation utilizing a prototype model. The ultimate prove of the applicability of the method by correlation to experimental data is not mentioned within this thesis, rather part of extended future work.

2.2 **Simulation Panorama**

The model presents a complex yet common simulation environment and methodology for the entire vehicle model in terms of Multi-body Dynamics (MBD). However, the present work exposes only few dynamic effects, i.e. with respect to the non-stationary operating conditions where it requires a full coupling between the involved domains. The current model represents the Mild Hybrid working strategy, where an integrated starter/alternator (ISA) combines the functionalities of both the starter motor and the alternator. While the internal combustion engine (ICE) produces most of the propulsion, force and ISA functionality is mostly related to start\stop. Moreover, the system is also capable to provide power assist and regenerative braking. The ISA can be placed in different locations within the drivetrain, such as coaxially, non-coaxially, within the auxiliary drive, as well as in the transmission. In our particular case, it is mounted directly on the Crankshaft (coaxial structure). [6]

A combination of the AVL's Simulation Tools is used to simulate the N&V behaviour of the complete powertrain. All software used in this computation integrates its own behaviour and functionality and accordingly reacts with the other one. Figure 3, highlights the general workflow of the complete vehicle model, stating its exact conjunctions and details in relation with the other one.

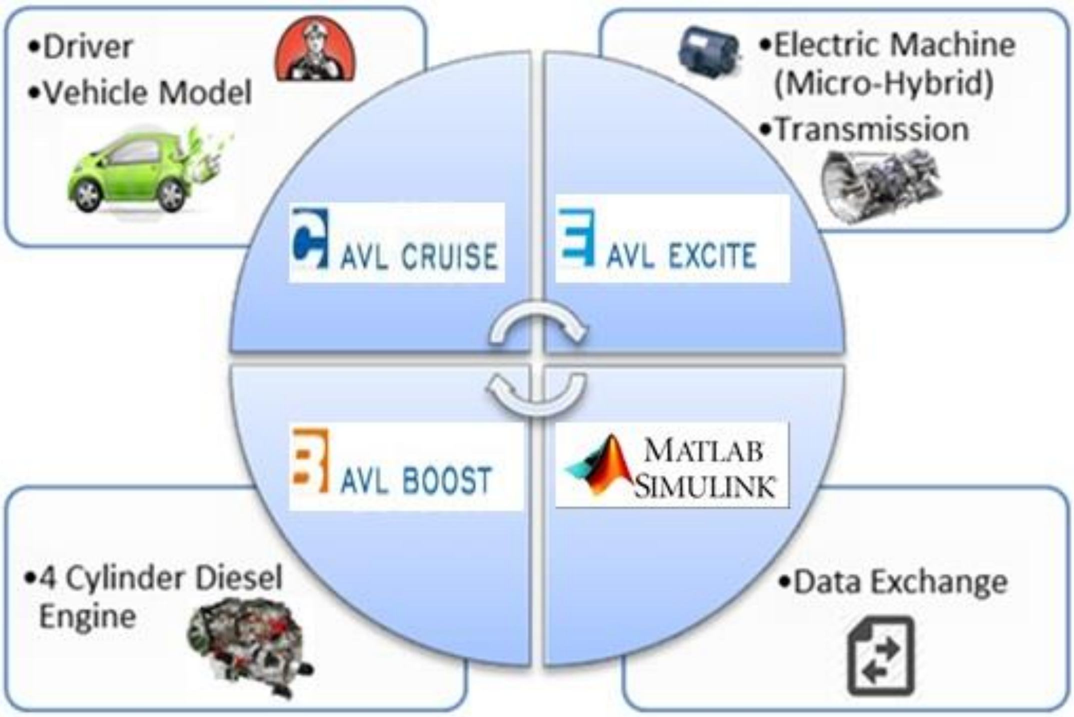

Figure 3: General workflow of complete simulation model.

As described, the complete hybrid vehicle model is a research oriented prototype model briefing a simplified 4-cylinder Diesel Engine in combination with a manual transmission. All the important vehicle data used in the whole system are being assumed for proper simulation process.

2.3 Enumeration of sub-models

2.3.1 Internal Combustion Engine and Gearbox – Mechanical representation

In order to reduce the noise pollution and to fulfil the rising comfort expectations, low noise vehicle engines and power units are required. Thus, comprehensive simulation methodologies and software tools are necessary during the design phase to analyze the complex physical events of noise generation. To understand and investigate this complex system, a dynamic approach is made where the computational model represent the required mechanical components and their interaction in terms of vibro-acoustics.

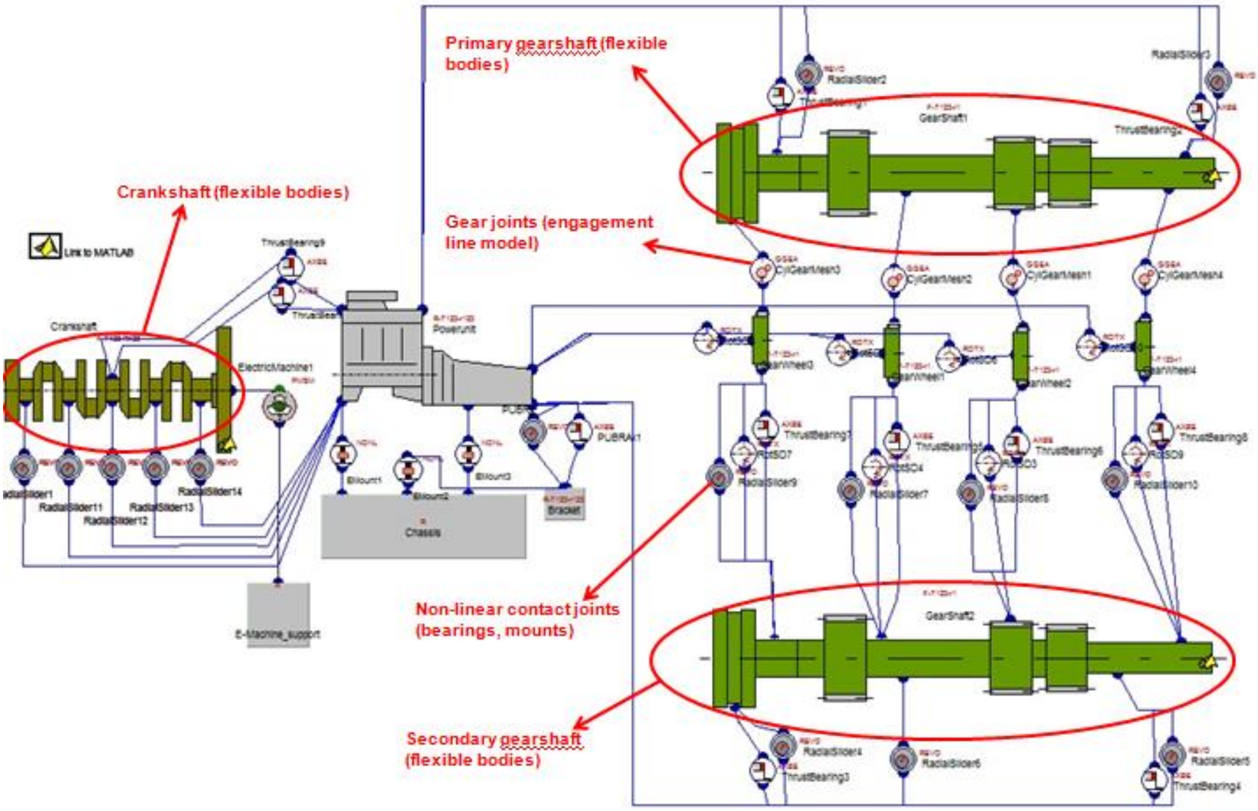

Figure 4: Engine representation and Transmission model.

Figure 4, shows the model, considering a set of flexible and rigid structures called "bodies". These bodies are interlinked via highly non-linear contact models referred to as joints. These joints are force-based and may represent simple stiffness/damper-units but also in-depth component representations such as slider and rolling bearing elements and gear meshes.

All the bodies mentioned within the model are defined with 6 Degrees of Freedoms (DoFs), which are 3 translational DOFs and 3 rotational DOFs. Depending upon the selection, they can perform global motions as well as elastic deformations (vibrations). Connections to the adjacent domains are established via the crankshaft, at its node representing the flywheel. The chassis body is used to support the complete powerunit that represents the engine structure and the gearbox housing. The connection between chassis and powerunit is realized using special bracket joints.

In the present analysis, the investigation on chassis body is disabled, as it is meant for model representation. However, for further analysis the chassis could be beneficial to understand the engine mounts dynamic behaviour and investigate the NVH role at relevant interior components (e.g. driver's seat rail, steering column).

The gearbox shown here is modelled via the primary and secondary shaft. The primary shaft is interconnected to the body representation of the (loose) gears using gear contact representations and loose gears are supported on the secondary shafts. Fixed gear wheels and synchronizer units are not modelled separately but incorporated into the body representations of the shafts and loose gears. For the gear engagement one of this loose gear (contact gear) is connected by a coupling torque generated by a torsional spring/damping unit to its carrier shaft, while the other non-engaged gears remain uncoupled in torsional direction. For this particular application, the complete simulation is carried out with only one gear in connection. Moreover, changing of gears is possible by adjusting the coupling torque in the torsional connections between loose gears and secondary shaft.

The overall system is solved based on time domain analysis, due to its high non-linear MBD system. The mathematical model is based on Newton's equation of momentum and Euler's equation of angular momentum. The equations of motion of each flexible body are based on a Floating Frame of Reference approach (FFoR), while using a body-fixed co-ordinate system. The origin of the reference frame is uniquely defined by imposing algebraic constraints, so called reference conditions. They ensure that the elastic deformations stay small in order to use a constant stiffness matrix. However, in every time step the system applies a backward differentiation formula (BDF) time integration scheme with adaptive BDF order method. The equations of motion of the complete multi-body system are a general implicit second order differential-algebraic equation (DAE) of the bodies.

During each time step, the resulting nonlinear algebraic equation system is solved by a modified Newton-like iteration. As the bodies are connected by joint force laws, iterative decoupling of the full multibody system is completed on body level. A Newton-like iteration step is performed for each body. In each iteration step, the joint forces are kept fixed. Once iteration step is finished, the joint forces are updated and predicted for the next iteration step of the bodies. [7, 8, 9, 16]

The coupling between the flexible bodies (crankshaft and transmission shafts) and MATLAB/Simulink is based on defined nodes for selected DOFs. The linked DOFs are forced to undergo the same motion for all the connected domains. The connected MATLAB/Simulink-interface and with it the other simulation domains are called up for each iteration performed to integrate across the current time interval. Due to model stability and work flow of the vehicle model, it is simulated on constant step size, but with the potential of variable step solver (referred in chapter 2.3.4). In order to simulate MBD vehicle model, the time required to calculate the complete drive cycle is approximately 3hrs (However, considering CPU specification as the limitation). Furthermore, after completing the simulation model the results are transformed into frequency domain for analysis.

The important components used in model are:

Electric Machine – The Electric-Machine used for the ISA is a Permanent Magnet Synchronous Machine (PMSM). The model comprises a fundamental wave approach with an inbuilt inverter, sensor, battery and control system. A torque controlled electric machine is used, allowing the machine to be switched on and off any time during the given calculated cycle. [10]

Gear Stages – The representation of the gear stages is done using so called engagement line models, where all force interactions are assumed to be taking place in the line of action (=pressure line) of the gear mesh. The model is capable to account for gear whine by applying a variable meshing stiffness. Uni-directional stiffness/damper units within each of the pressure lines of the driving and backlash flank side allow for consideration of gear backlash. In this respect, the model is also capable to resolve impact driven effects such as gear rattle as well as driveline clonk. [10]

Link to Matlab joint – Mainly used to couple the individual software tools and for data exchange assistance. The joint realizes the interaction between the bodies through displacement/velocity and acting force/moment, while running along with the simulation. [10]

2.3.2 Internal Combustion Engine Thermodynamics

The engine model describes a simplified 4-cylinder turbocharged direct injection diesel engine simulated with BOOST RT. As shown in Figure 5, it comprises of components stating compressor, turbine, injection ports, mass flow element, and cylinder's, engine and exhaust ports.

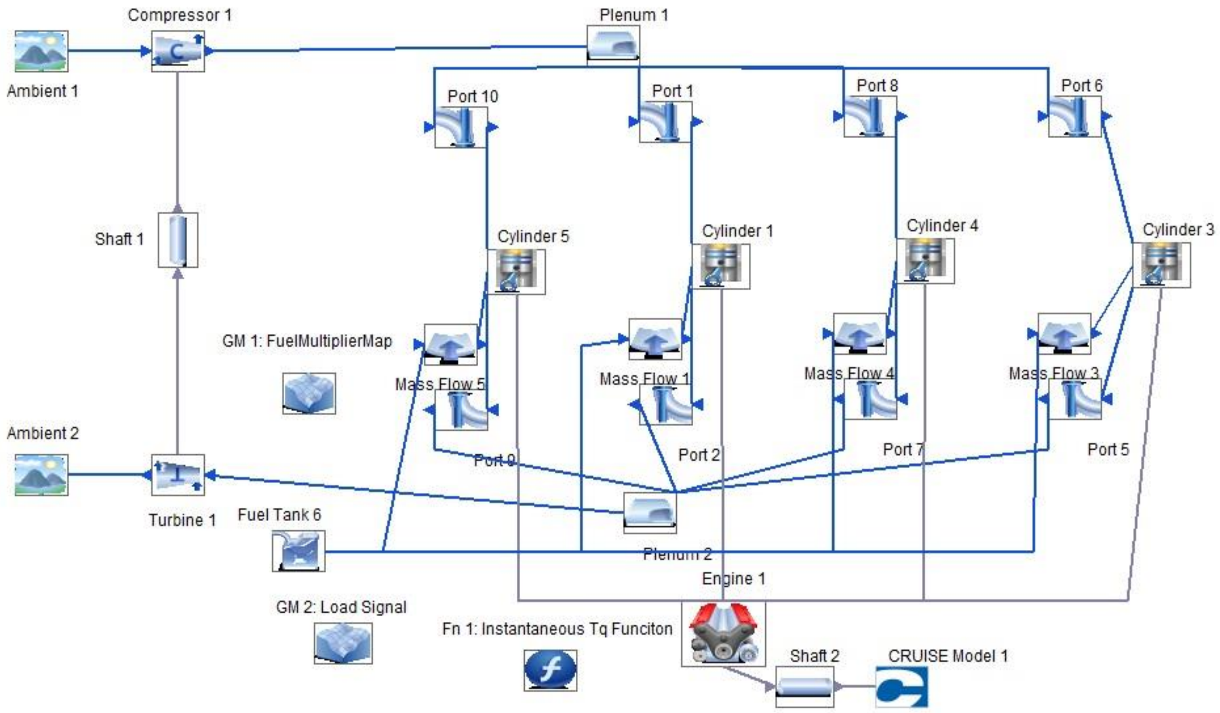

Figure 5: Turbocharged Diesel Engine model.

The model stimulates a simplified air path crank angle resolved cylinder model through filling and emptying approach for transient operation. Due to its model simplification, it is capable of analysing detail thermodynamic engine model with due consideration of transient condition. For the complete vehicle model to be scrutinized, it additionally contains the other domains i.e. the vehicle model, which implicates the engine quantities through databus channels and mutually receiving the crankshaft quantities via MATLAB.

Sub-divisions of ICE model are as follows:

Mass (air/fuel) flow – The model represents a turbocharged direct injection diesel engine, the amount of fuel injected into the cylinder is prescribed via mass flow elements i.e. with defined flow profile. These elements are further connected to fuel tank element. The suction of inlet air path is from the ambient element through the compressor and plenum to the inlet ports. Similarly for the exhaust air path, this is constituted by the outlet port, plenum and turbine.

Cylinder – In order to account for crank angle resolved behaviour, each cylinder elements calculate their thermodynamic processes with consideration of conversation of mass and involved species and energy. The balance equations are used within the background to consider the exchange of mass flows or pure energy flows with the arbitrary number of attached elements. The calculations carried out within the cylinder are undertaken on crank angle basis with respect to time domain. [11]

Engine – The engine element incorporates and controls all the four cylinders and effectively calculates the engine parameters. Furthermore, the element receives the angular velocity from the attached mechanical element (Shaft 2) and in return delivers instantaneous torque to the flexible crankshaft model.

2.3.3 Control, Vehicle and Driveline

Figure 6 represents the model consisting of the engine (Figure 5), clutch, shaft and flanges (connection to mechanical represented crankshaft and transmission shafts), single transmission ratio, differential, brakes and wheels, driver and the vehicle. The whole vehicle model is further controlled by the driver's profile, which has pre-defined manoeuvre.

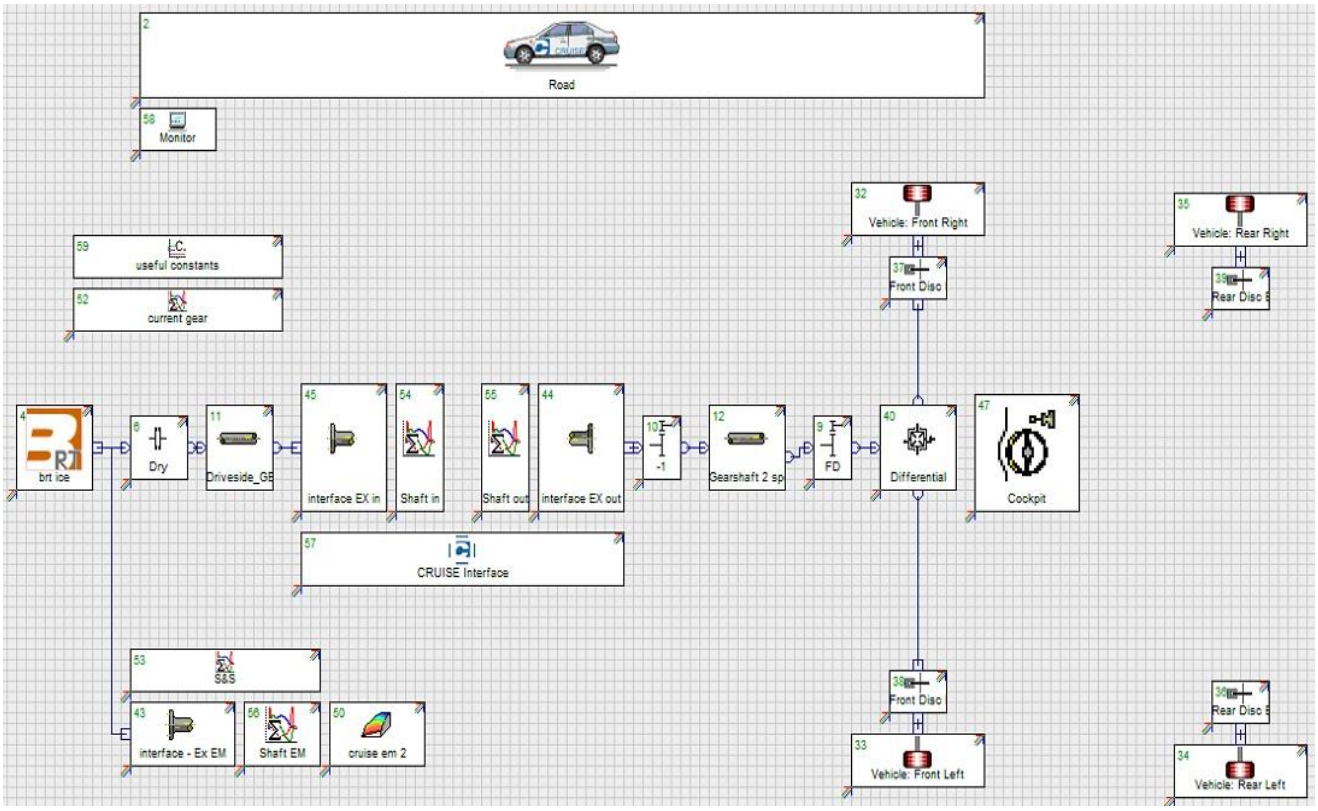

Figure 6 : Cruise hybrid vehicle model.

The most important components of the vehicle model are:

Vehicle – The vehicle component contains general data of the vehicle, such as nominal dimensions and weights. Road resistances and dynamic wheel loads are calculated for road and dynamometer runs based on the dimensions and the load state. The wheel loads are calculated while taking into account longitudinal vehicle dynamics. (E.g. from the effects of the acceleration, aerodynamic drag, rolling resistance). [12]

Interface(C-Interface) – This interface component defines the coupling between the vehicle model and the engine housing/gearbox and thermodynamics part models. The usability of C-Interface component is at each time step, the user defined outcoming vehicle data is interchanged and forwarded to the server application through databus channels and vice-versa for incoming data through channels. [12]

Driver – The driver component within the model guides the vehicle to the user defined driving cycle or manoeuvre. However, it takes over the control of all ABC (acceleration, brake, clutch) pedals.

2.3.4 Formation and Operation (Coupling)

The main emphasis of the first computational model was to create a multi physics simulation model for a Mild-Hybrid powertrain vehicle. The second important aspect was to build a methodology for the existing software tools to be used and its fidelity towards complete vehicle model.

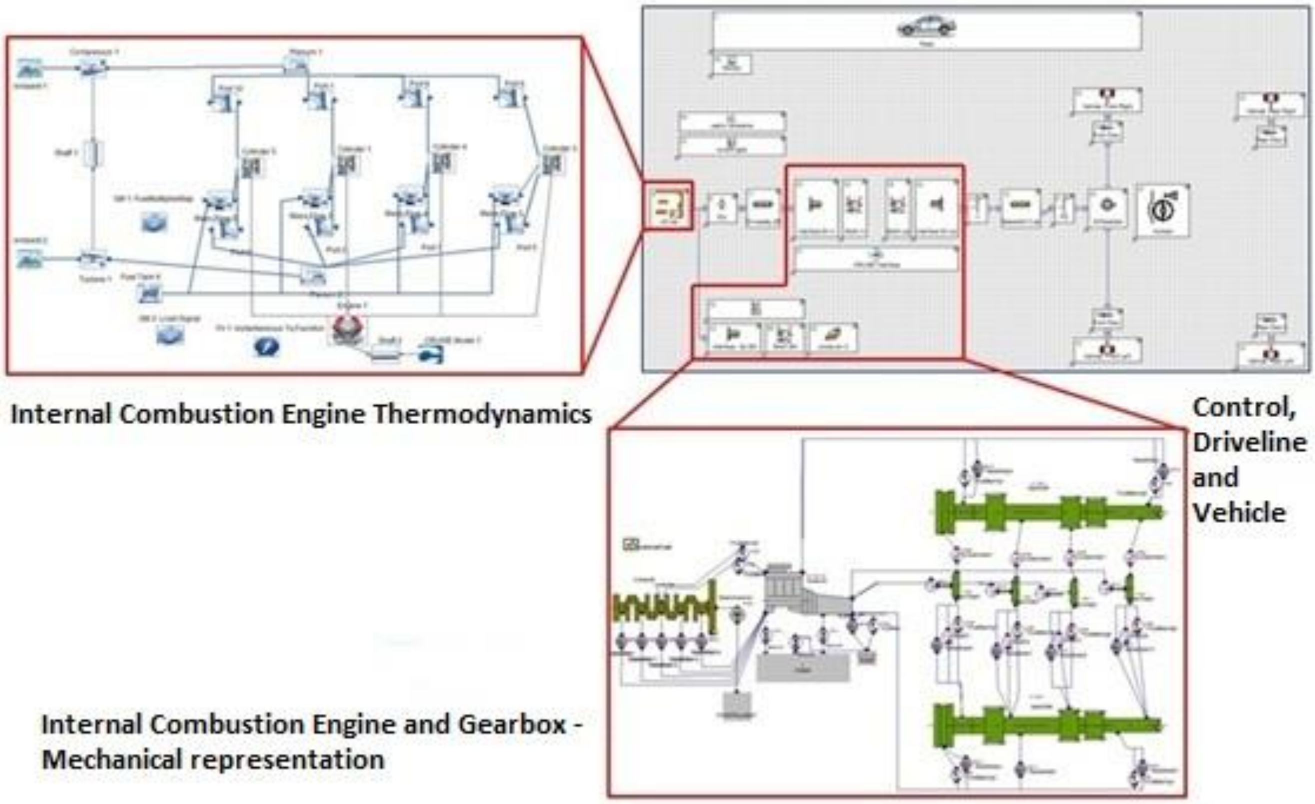

Figure 7: Formation of complete model.

Figure 7 shows an overview of the complete vehicle structure with an image of HEV model. The simulation begins with the electric machine (used as ISA) that cranks the engine with pre-defined torque. The cranking torque of the e-machine is delivered to the flexible body crankshaft modelled within ICE representation. The crankshaft is further coupled with thermodynamic engine model, which is embedded with the vehicle model. In simple terms, the thermodynamic engine model is inter-connected with the mechanical representation powerunit model and the transmission model. The MATLAB tool is used as communicator between the part models.

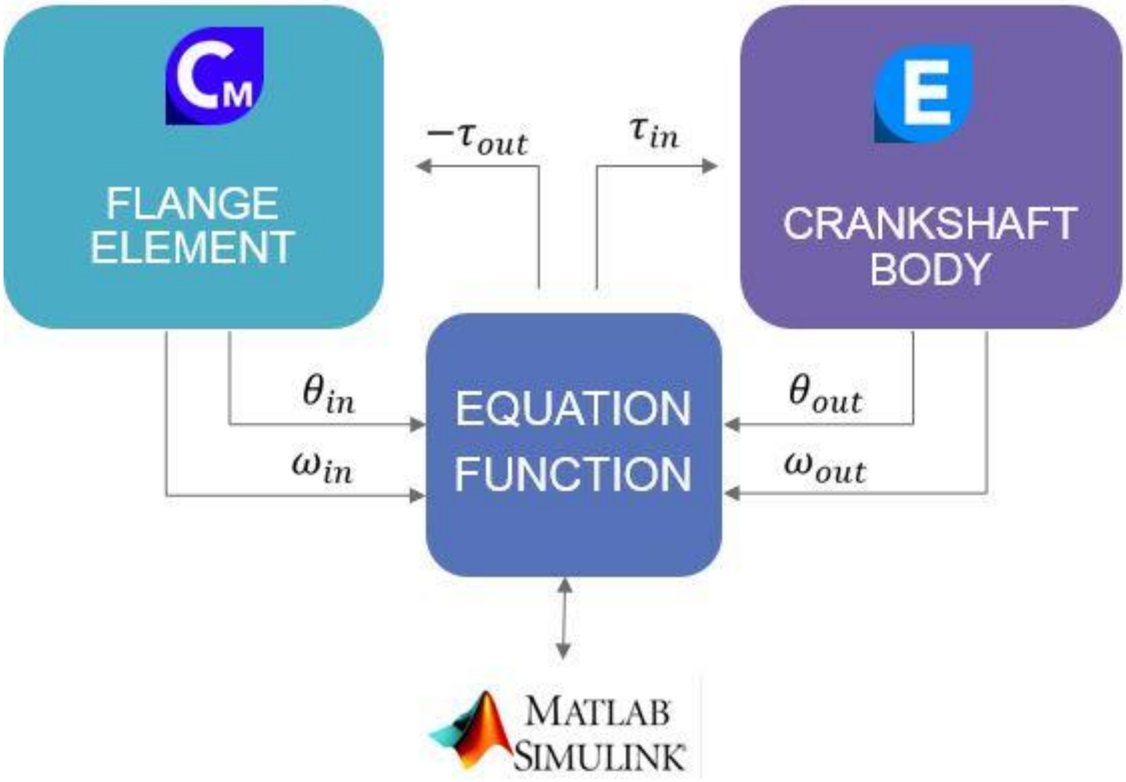

Figure 8: Background of transferred quantities

As described earlier, the crankshaft receives torque from electric machine, which in return provides crankshaft rotational quantities. Figure 8 gives the general background on how the rotational quantities are further passed onto the Engine and Transmission through Link to Matlab joint. These state vectors / quantities (=displacements and velocities) are linked via DOF's for the selected node and are interchanged at each time interval. In vehicle model, the state vectors are migrated to flange element, which acts as a coupling tool through C – Interface component. In addition, the formulation of the moment is done with the help of these state vectors i.e. via stiffness – damping equation and this is how the engine is cranked at its initial state.

Figure 9, gives overview of inter-changed state quantities from one domain to the other.

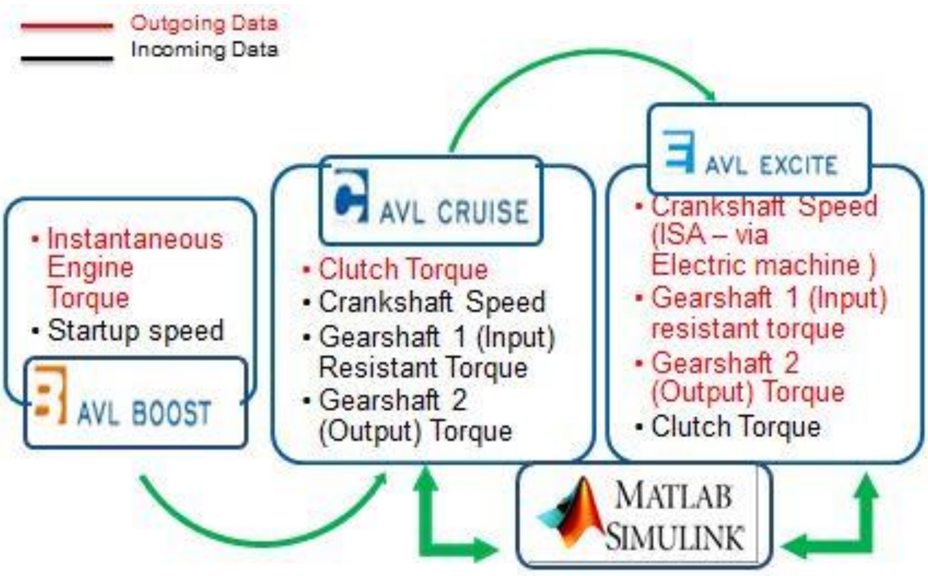

Figure 9: Data workflow between software's

The general workflow of driving cycle starts with the engine, which is cranked for a period of 0.25 seconds by electric machine and thereafter, the combustion phase takes over the control. However, during this combustion phase, the driver handles the main control of the engine with due respect to the mentioned profile (Figure 10). The combustion engine ramps up with an output of instantaneous torque that is transferred to the frictional clutch, which represents a simple yet dynamic clutch model. The clutch torque is then guided to the primary gearshaft, where detailed transmission model representing a 4 speed manual gearbox modelled with 3D elastic bodies, synchronizers and gear joints (Figure 4). Moreover the coupling of detail transmission model and driveline model is completed with respect to support of transferred quantities and flange elements, as shown in Figure 8.

For the transmission analysis, exclusively the 1st gear is engage during the complete driving profile, as the N&V effects are more pronounced in the launching and slow driving phase. On the other side, non-engaged gear stages, i.e. the rotational connection via the synchronizer units are left open as a freewheeling. However, the development of the gear meshes and the loose gears are capable to account for gear rattle excitement by the speed fluctuations of the combustion engine. And as mentioned earlier for proper support, both the gearshifts are inter-connected by joints (axial and radial bearings) and its connection to the transmission housing. The state quantities (displacement and velocities) of the secondary gearshaft are then forwarded to the remaining driveline representation of the vehicle model via MATLAB.

The simplified driveline model consists of final drive, differential and wheels with no slip is consideration. Thus complete vehicle model considering engine, ICE mechanical representation and transmission and vehicle model is thus controlled and manoeuvred with a standard driver element based on custom-made driving profile.

The complete vehicle model, as shown in Figure 7 is simulated for a period of 10 seconds following a driving profile with a Tip-In/Tip-Out event, as depicted in Figure 11. All the domains and the simulation models are ran on fixed step size solver, as they are efficient to highly non-linear and strong stiffness systems. Fixed step solver also helps all the model state quantities to process in each iteration at every time interval. The interlink between the software's represent a simple logical approach of 'Venn Diagram theory', (as shown in Figure 10) where the common state quantities between the software's are interchanged during constant time interval. [17] However, further investigation on coupling methodology can overcome the major limitations, such as use of different solver, computation time and modelling usability. With the recommendation in the direction of a so-called communication grid solution, where solvers of each domain run independently with their own optimum solver type and variable step size. The inter-domain communication is then imposed by a fixed time grid where the coupled stated quantities become synchronized. The advantage of using grid solution is where, each and every domain can perform better within its environment and further accomplish the requirement to build complete vehicle model.

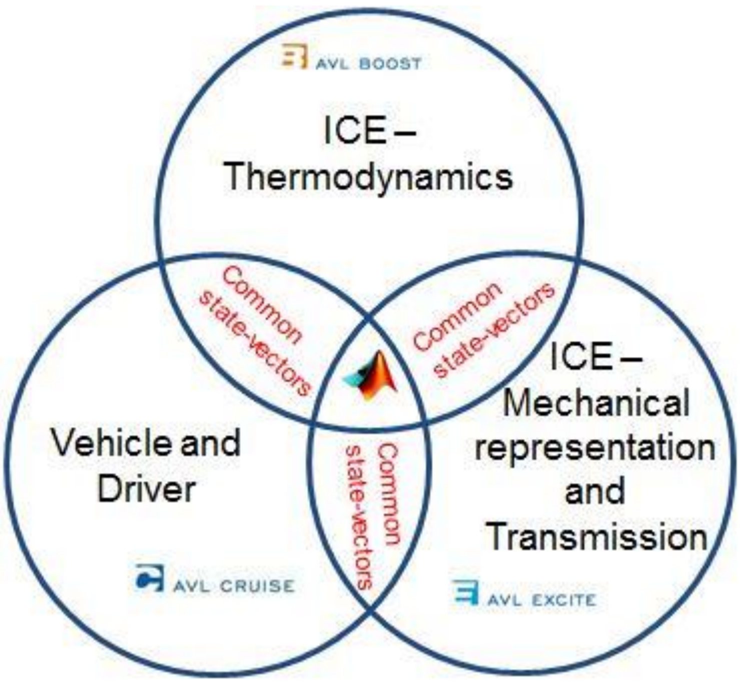

Figure 10: Venn Diagram theory

2.4 Scenarios and Results

The current model is capable to simulate a variety of N&V issues; but it addresses few important scenarios, while setting an example for complete hybrid vehicle model. The fundamental driveline and vehicle data assumed for the simulations is outlined in Table 1.

Table 1: Data Specification and Assumption.

Engine	4 cylinder – Four stroke Turbocharged DI Diesel
Stroke	90.4 mm
Bore	83 mm
Connecting Rod Length	150 mm
Compression Ratio	16.5:1
Transmission	4 Gear – Manual (only 1st gear is engaged for simulation)
Electric Machine	Permanent Magnet Synchronous Machine (6 Magnetic pair poles)
Gear Ratio	2.81 (1^{st} Gear)
Vehicle	No Wheel Slip and Cornering
Final drive ratio	3.421

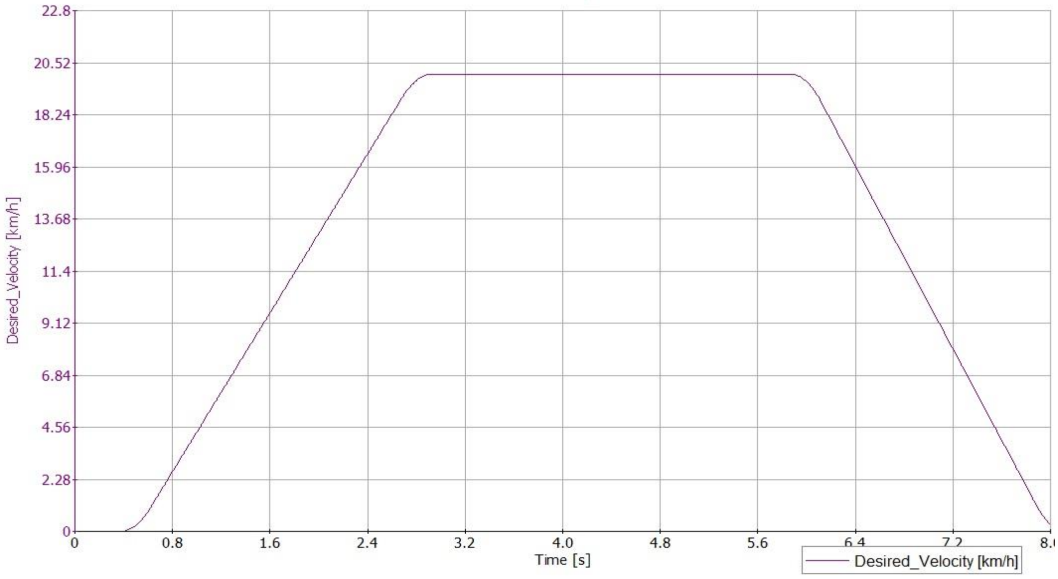

Figure 11: User defined driving manoeuvre (Driver's profile)

2.4.1 Scenario 1 – Torque Ripple

Torque Ripple – Engine Start/Stop: There is consistently an image of noise and vibration during start/stop of an engine and in sudden acceleration/deceleration process, but with HEV it seems more prominent. This effect is related to the rapid torque application from electric-machine onto the IC-engine. In this assessment, the torque ripple is examined with respect to starting behaviour of an IC-engine. In general scenario electric-machine grants the required torque to the IC-engine for start-up and this sudden torque leads to vibration. In technical terms, the vibration at engine start can be divided into two phases. While the 1st phase before ignition (=cranking) is determined with respect to compression reaction forces and pumping pressures in the cylinders. The other phase is just after the ignition, which dominates the rapid engine torque increment caused by combustion. [13]

Nevertheless, in this simulation model only one phase is characterized, where the vibration occurs due to instant torque from an electric machine.

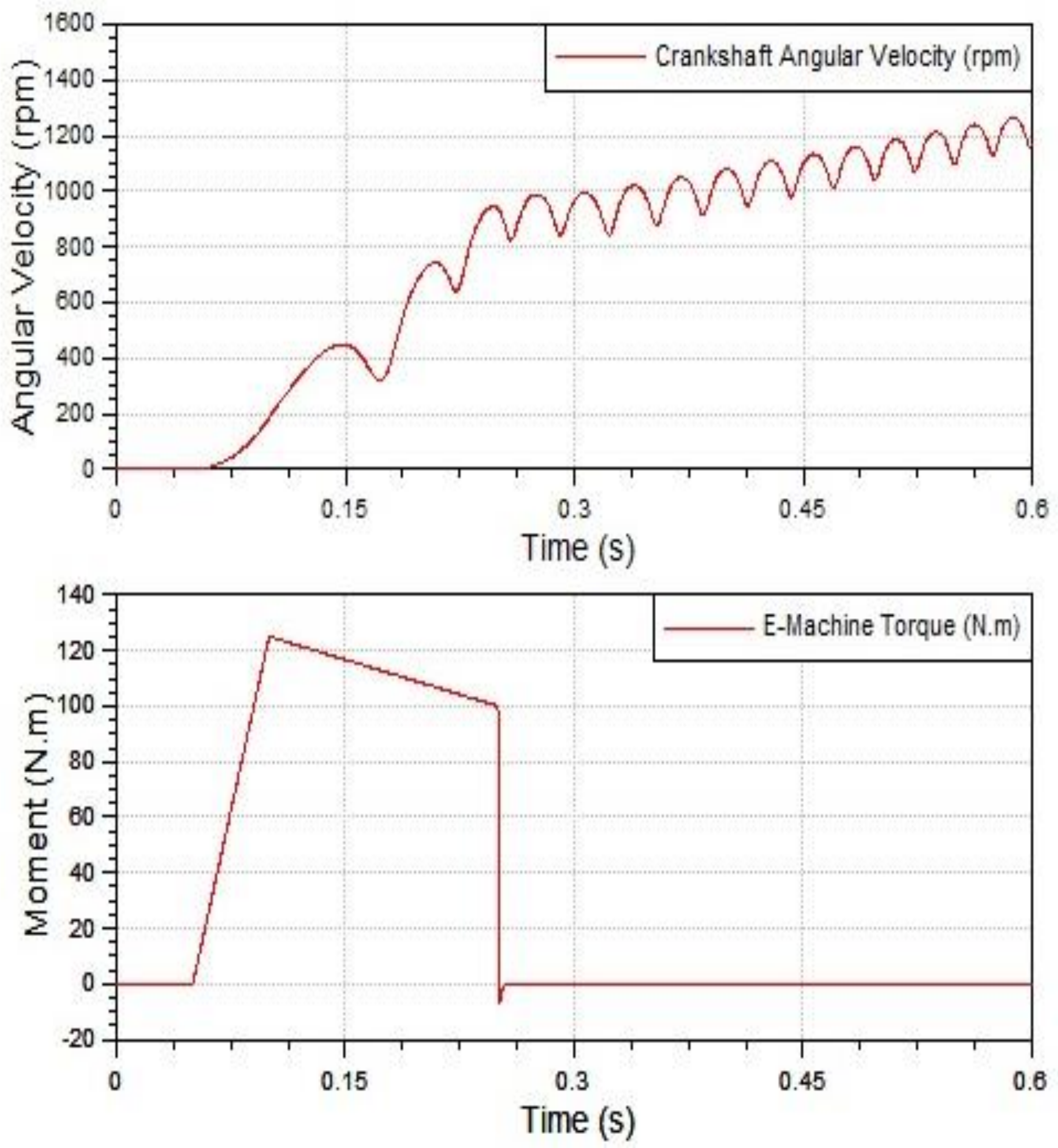

Figure 12: Crankshaft Angular Velocity and Electric Machine Torque.

Figure 12 (upper) shows the Start-up angular velocity of the crankshaft and Figure 12 (lower) represent pre-defined reference torque of electric machine.

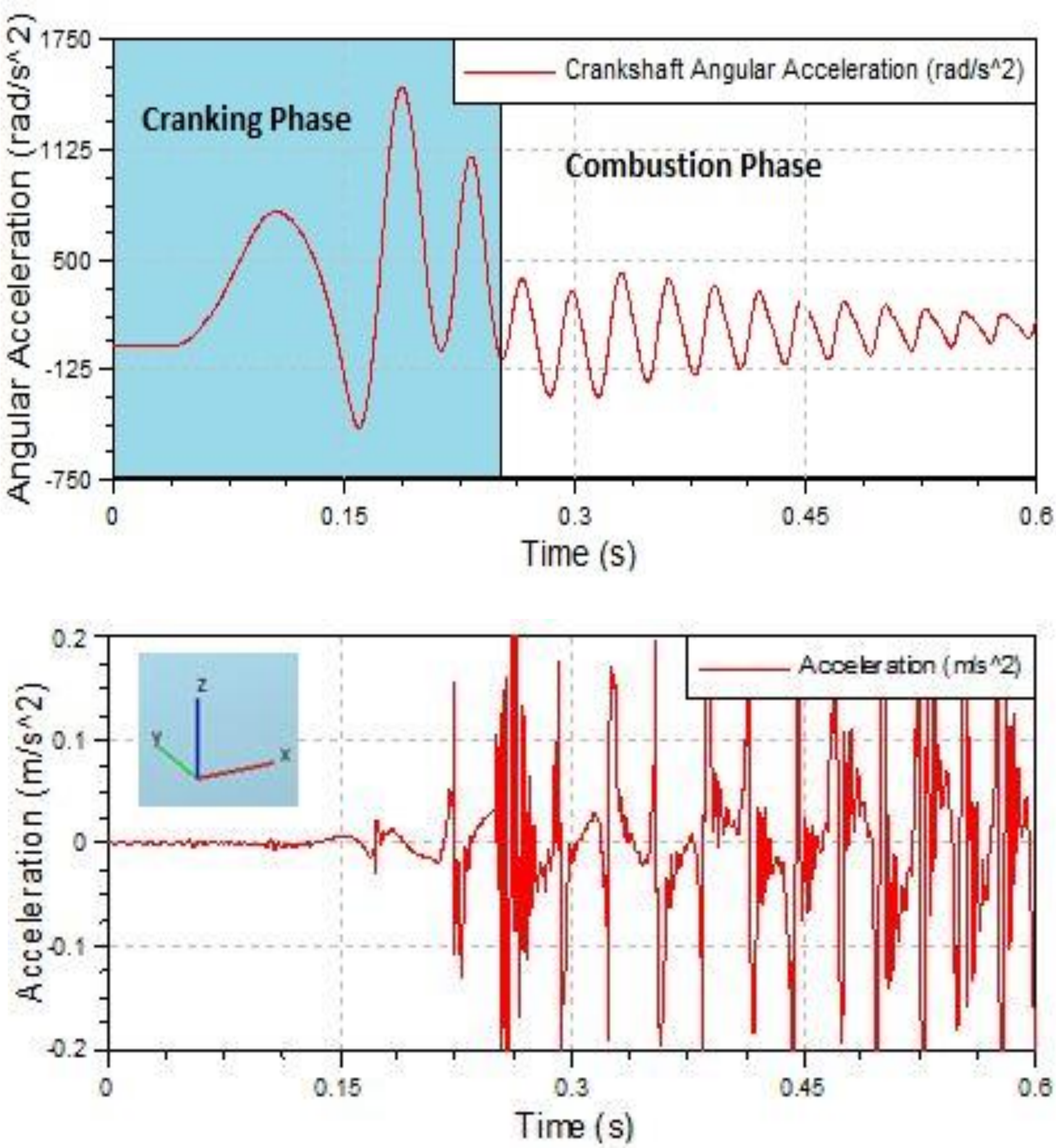

Figure 13: Angular Acceleration and Acceleration of Crankshaft (Global motion)

Figure 13, represents an overview of torque ripple found on angular acceleration and acceleration of the rotational crankshaft acting in linear direction (z-direction, towards cylinder axis direction). During the cranking phase i.e. from 0 – 0.25s, the engine is cranked with the help of electric machine. As shown above, the angular acceleration peak during this period is caused by the sudden moment, which is due to the combustion pressure generated inside the cylinder. From 0.25 – 8s, the vehicle is powered by IC-engine, where the combustion phases is active. The combustion phase is the phase where a high temperature and pressure exothermic redox chemical reaction between fuel and an oxidant arises. This combustion introduces to a sudden engine knocking that is felt during the starting phase. The combine effect of electric machine and start of IC-engine i.e. in between 0.25 – 0.3s, introduces to a complex level of noise and vibration effect within the system (as shown in Figure 13 (below) crankshaft acceleration). This is further perceived by the driver through structure borne noise, while generating an audible noise with it.

The results shown above are in agreement to previous research work conducted by Ito, Yoshiaki. et.al. [13] The effects shown in the present work are related to the crankshaft, whereas Ito, Yoshiaki. provides an overview for floor acceleration. However, to overcome this ripple various control strategies are available, but the focus of present work, the main objective is to model and verify whether it is possible to simulate the effects using the available tools, with consideration of complete vehicle model.

2.4.2 Scenario 2 – Driveline shuffle and clunk vibration

The vehicle shuffle and clunk is known as a low frequency longitudinal vibration generally occurring in the range of 2-10Hz. This low mode transient vibration depends on gear ratio, inertias and stiffness and can be a result of engine tip-in, tip-out, automatic or CVT gear ratio change, clutch engagements and braking. [14] It is caused by the 1st Eigenmode of the drivetrain, where the powertrain vibrates against the vehicle inertia. The excitation to this mode is given by sudden torque changes due to acceleration or deceleration phases of the vehicle. The engine tip-in/tip-out manoeuvre is a standard for the assessment of vehicle shuffle. The shuffle is typically most pronounced in the tip-out phase where the engine torque changes from drive towards coast condition and further can be controlled by appropriate engine torque management strategies.

Upper diagram in Figure 14, compares the target speed profile of the vehicle (green curve; converted to crankshaft rpm) with the rotational speed of the crankshaft (red curve). The difference between the actual and the desired profile is mainly caused by the properties of the driver model. As already outlined previously, the maximum shuffle can be observed in the phase of the tip-out starting at ~2.8 s. In the subsequent period of nearly constant velocity the vibration decays due to the driveline damping.

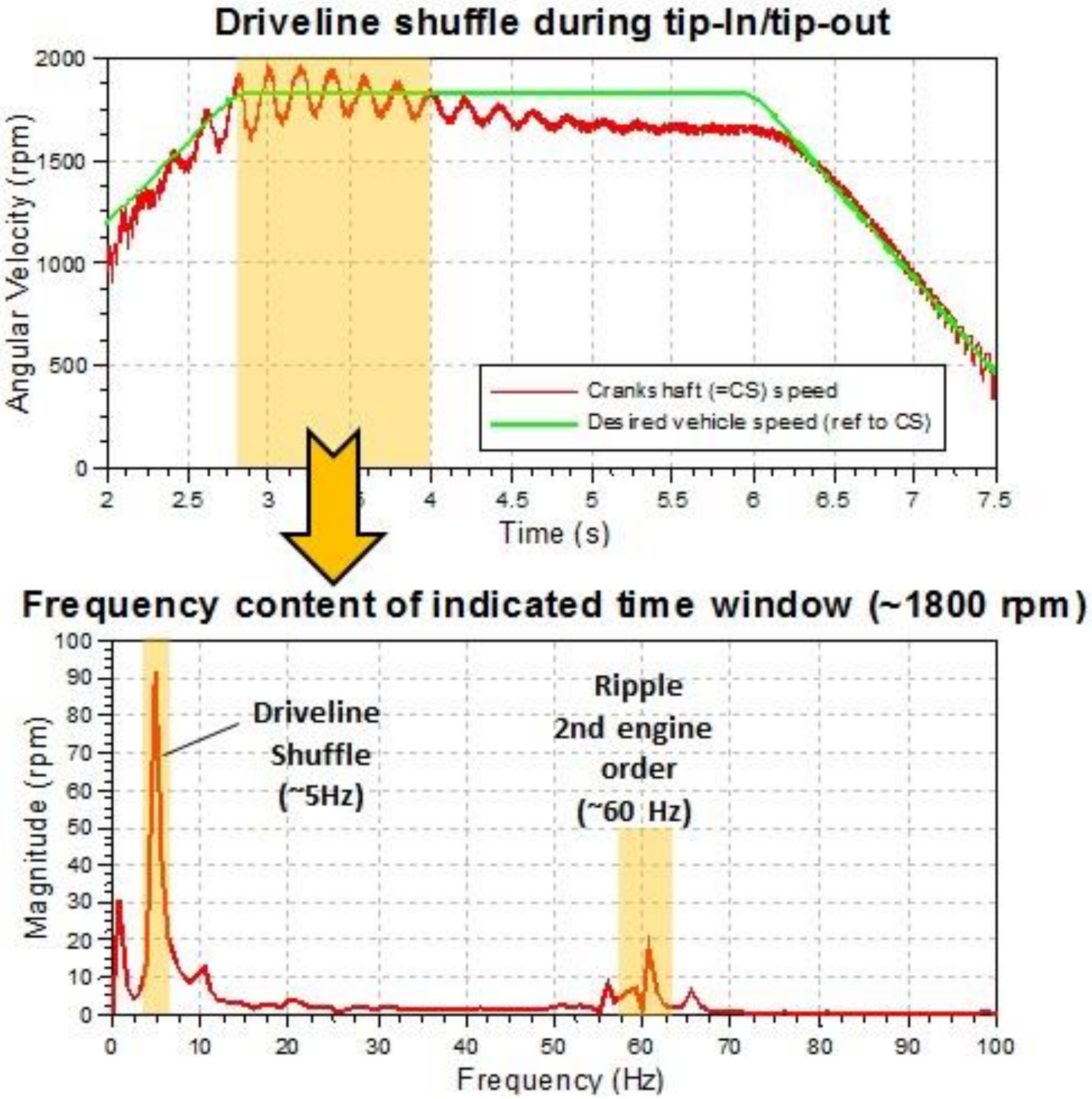

Figure 14: Driveline Shuffle.

The lower diagram of Figure 14 represents the low frequency content of the vibration indicated during engine tip-out region between 2.8s and 4s. Beside the dominating shuffle frequency (~ 5Hz), also the ripple due to the 2nd engine order contributes to the observed speed fluctuations. The peak left to the shuffle (~0.1 Hz) is caused by the shape of the profile itself.

2.4.3 Scenario 3 – Gear Whine

Generally, gear whine is manifested as a narrow banded tonal sound that is excited due to the periodic fluctuations of gear mesh contact properties, commonly known as Transmission Error (TE).[15] The related noise is manifested at the gear mesh frequency (according to pinion's speed and number of teeth) and its multiples (harmonics). Due to its pitched, thus annoying nature, gear whine noise is audible up to very high frequency levels (e.g. 10 kHz) which may extend the frequency range of practical interest significantly. Consequently, the applied gear contact model as well as the representation of the flexible structures must be highly efficient. [16] The multi-flank pair contact can be characterised as an advanced force-law which gives a relationship between the dynamic motion of the connected gear wheels and the constraining forces/moments acting on the gear. However the results shown is simulated with standalone gear model, due to its requirement regarding computational efficiency and load cases, but the same gear transmission model can be used in the full system model too.

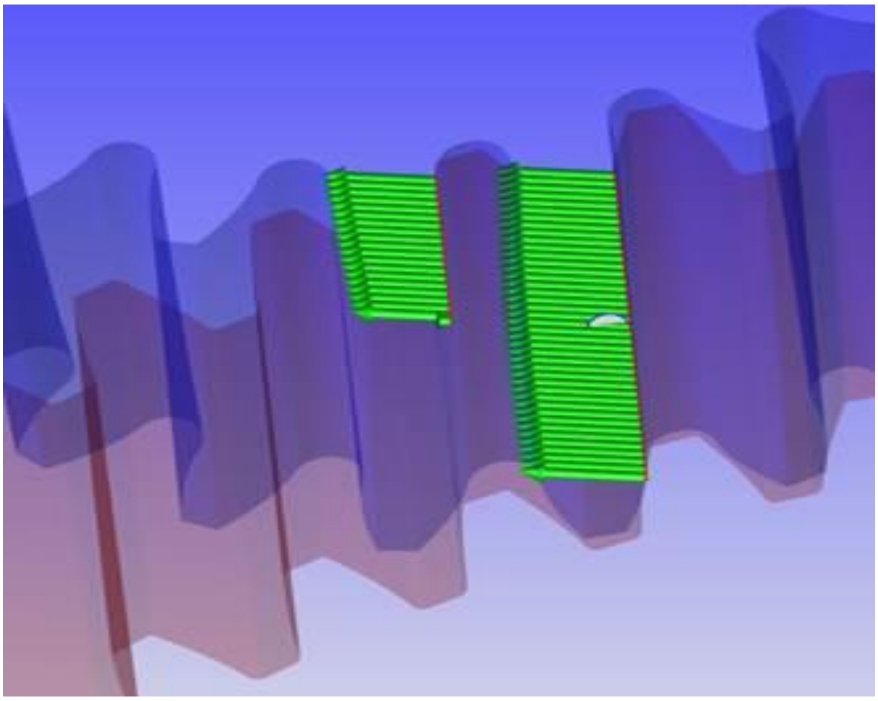

Figure 15: Normal Force distribution at engaged gear

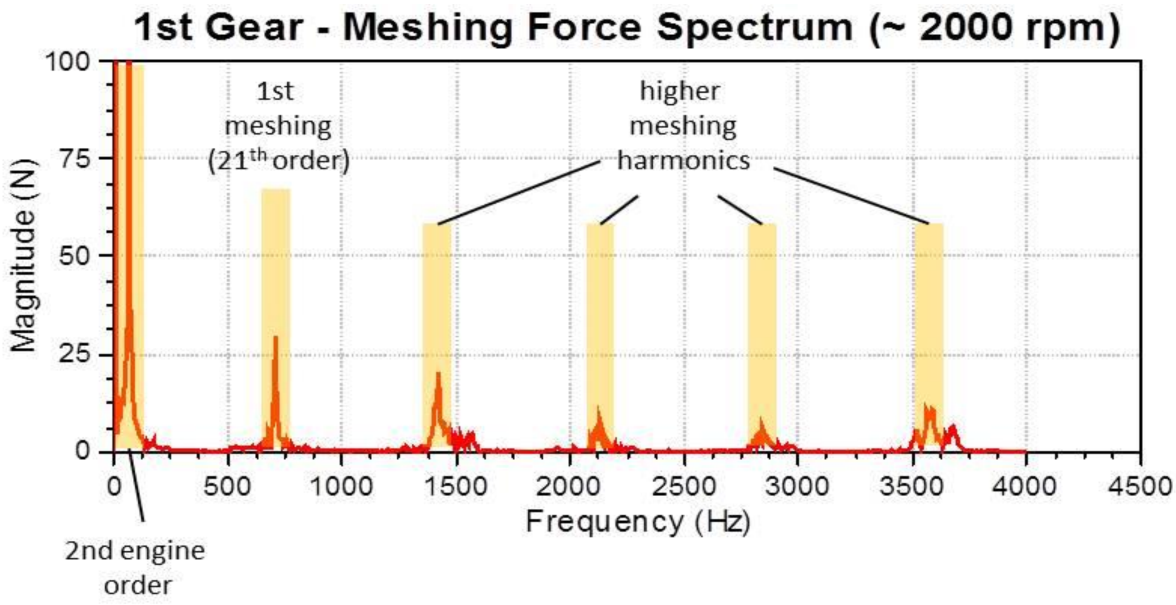

Figure 16: Frequency spectrum of meshing force variations found at constant phase of tip-in/back-out manoeuvre (4 – 5 s)

Figure 15, gives the picture of a typical meshing force distribution at a time instant where two flank-pairs are in contact at the same time. According to the overall contact ratio of the engaged 1st gear (ε_γ = 1.78), the number of pairs in contact fluctuates between 2 and 3. Among other effects as shape tolerances and errors, this variation is the main source for the fluctuation of the meshing force, which finally excites the gear whine. Figure 16 shows the frequency spectrum of a short time windows extracted in the constant phase (approximately at 2000 rpm) of the tip-in/back-out manoeuvre. One can see the 1st meshing order, which corresponds to the number of teeth at the pinion (z_1 = 21). Since the imposed fluctuations are not purely sinusoidal, additional higher order harmonics are developed which reach up to very high frequency levels.

Beside the contributions from the gear meshing, at lower frequencies the fluctuations caused by the combustion engine speed variations (2nd engine order = 67 Hz) become visible.

Modulation effects between low frequency vibrations (e.g. vehicle shuffle and engine excitations) give rise to meshing order sidebands, which additionally alter the tonal character of the induced gear whine noise.

2.4.4 Scenario 4 – Gear Rattle

Gear rattle is a broad banded noise (typically between 500 Hz to 3 kHz) induced by impacts of loose (=unloaded) gears, while travelling through the gear's operational backlash [16]. The actual source of the gear rattle is associated with the speed fluctuations imposed by the combustion engine. Isolation components between the engine and the transmission input - such as the Dual Mass Flywheels (DMF), Torsional Vibrations Dampers (TVD), or nowadays-centrifugal pendulum absorbers (CPVA) – are capable to reduce speed fluctuations significantly. However, the remaining rotational vibrations are transferred to the transmission input shafts and may expose loose gears of disengaged gear stages to bounce within the backlash. Although the resulting impacts are typically of low force level (up to 100 N), they act like frequent little hammer hits exciting the gearbox housing to structural vibrations finally percept as trash or rattle sounds. Beside the noise itself, the trashy character suggests a low-built quality or even serious damage to the gears, which in results to frequent subject of customer complaints.

Figure 17, depicts gear contact forces in the mesh of the disengaged 2nd gear observed in the constant phase of the tip-in/back-out manoeuvre. The frequent loss and gain of contact is an indicator for the loose gear vibration that finally causes gear rattle.

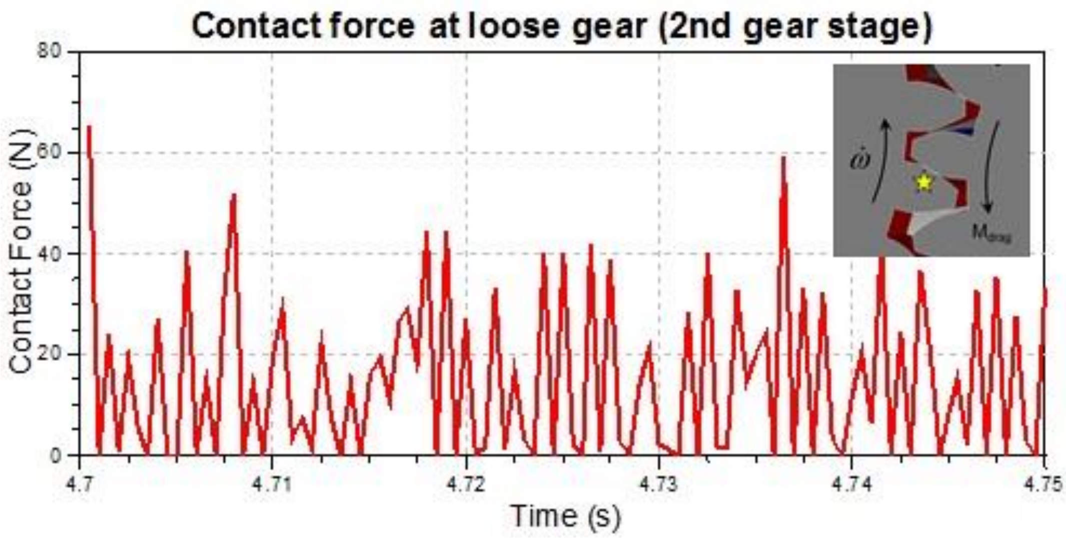

Figure 17: Gear Rattle on Loose Gear

With respect to the simulation of gear rattle, assumptions regarding the magnitude of the stabilizing drag torque acting on the loose gears as well as the damping in the gear's backlash are essential for the quality of the prediction. In this context, the trend towards minimized transmission losses often promotes gear rattle effects.

2.5 **Assessment conclusion**

The model puts a special focus on building the complete vehicle MBD model based on the methodology of combining and coupling the different modelling domains within a comprehensive simulation framework. However, the second and most important aspect is to not only build the model and simulate, but also influence the dynamic effects related to the noise and vibration.

The benefit of coupling all the domains is that each domain model here is capable of investigating its own dominant behaviour under the complete vehicle condition. It even helps to understand and thoroughly analyse each sub-model within each domain and its effect on noise and vibration modes, which can be used to improve and develop advance sub-system.

The work presented outlines the prediction of various vibrational effects in a mild-hybrid driveline in conjunction with a predefined driving manoeuvre and its control via a driver model. The consideration of the crank-angle resolved thermodynamic behaviour of the combustion engine accounts for speed fluctuations - as main source of booming noise and gearbox rattle - and provides online driving torques throughout the transient driving profile of interest. Beside the aspects of torsional drivetrain vibration, such as shuffle and associated clonk, the approach is capable of in-depth 3D-modeling of components, for instance the gearbox internals including the gear meshes.

The simulated results can further be use verify with the test data and to examine under different load condition and operation. However the methodology does increase the complexity of computational modelling, but it can be improved and further develop based on the numerical time-integration scheme. It leads in the direction of an overall adaptive method, which shall ensure that the different domains can be simulated with optimal computational performance.

3 SEAT VIBRATION AND ITS EFFECT ON HUMAN FACTORS

3.1 Architectures of Hybrid Electric Vehicle (HEV)

Hybrid vehicles are generally classified in many different configurations and has even been recognized with different terms such as micro, mild or full hybrid, whose purpose are quite similar to the one's mentioned below.

The most commonly used configurations for electric propulsion are:

- Series HEVs:
  - Figure 18, shows the architecture of Series HEV, where the ICE is the main energy converter that converts the original energy in gasoline to mechanical power. The mechanical output of the ICE is then converted to electricity using a generator. The electric motor moves the final drive using electricity generated by the generator or electricity stored in battery. The electric motor receives electricity directly from the engine or from the battery or both to drive the vehicle.

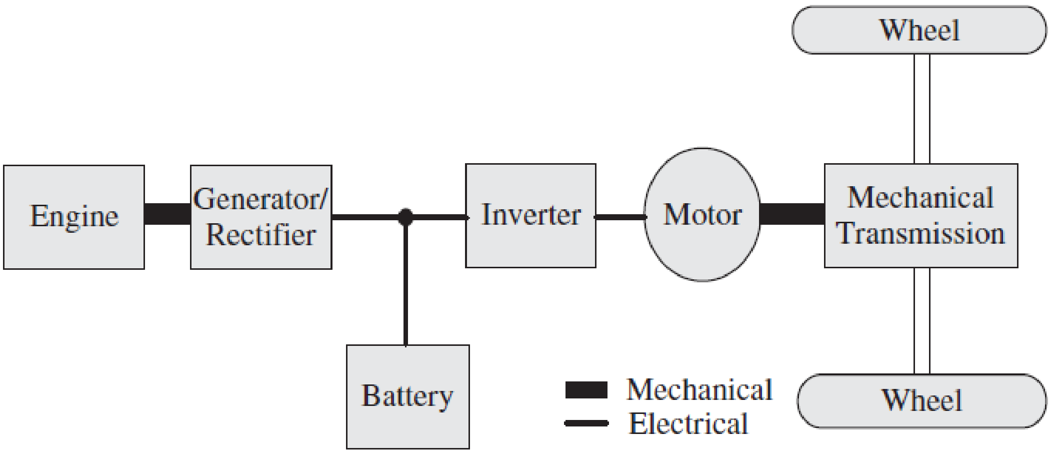

Figure 18: Layout of Series HEV. [18]

- Parallel HEV:
  - Figure 19, shows the architecture of Parallel HEV, where the ICE and the electric motor can both deliver the propulsion power in parallel to the wheels. The ICE and the

electric motor are coupled to the final drive through a mechanism such as clutch, belts, pulleys and gears. Both the ICE and the motor delivers power to the final drive, either in combined mode or each separately. The electric motor can even be used as a generator to recover the kinetic energy during braking or absorbing a portion of power from ICE.

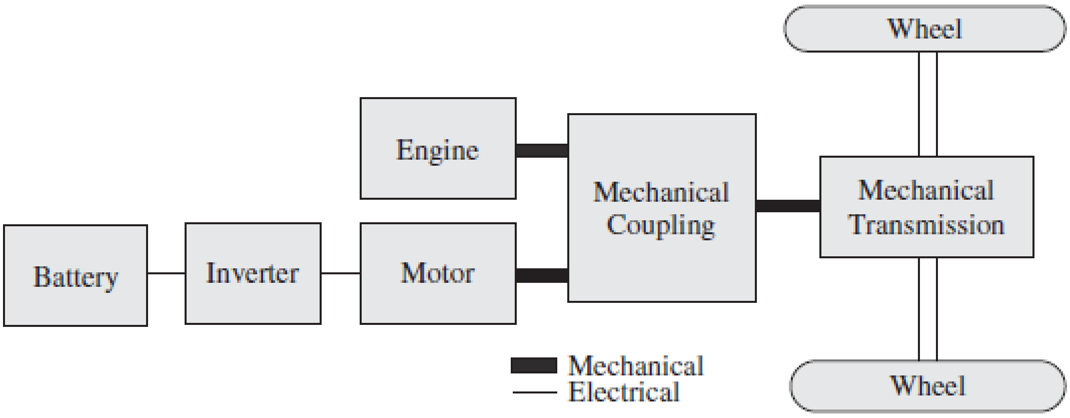

Figure 19: Layout of Parallel HEV. [18]

- Power-Split HEV:
  - Figure 20, shows the architecture of Power-Split HEV, which are complex and incorporates the function of both series and parallel HEVs. Therefore, it can be operated as a series or parallel HEV. The vehicle is powered from the ICE and electric motor in combination with the mechanical coupling. However, the operating condition of architecture plays a crucial role within power-split HEV.

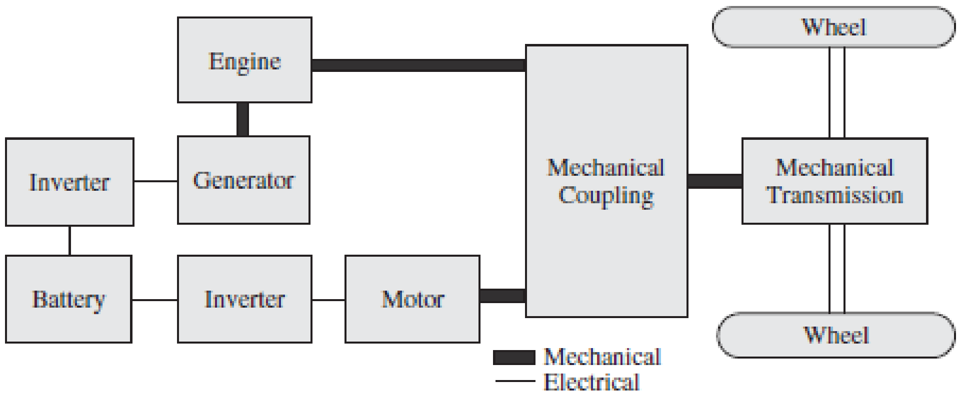

Figure 20: Layout of Power-split HEV. [18]

Moreover, for current deep investigation the HEV powertrains are further detailed and identified by its functionality and the placement of an electric machine within the configuration, as shown in Figure 21.

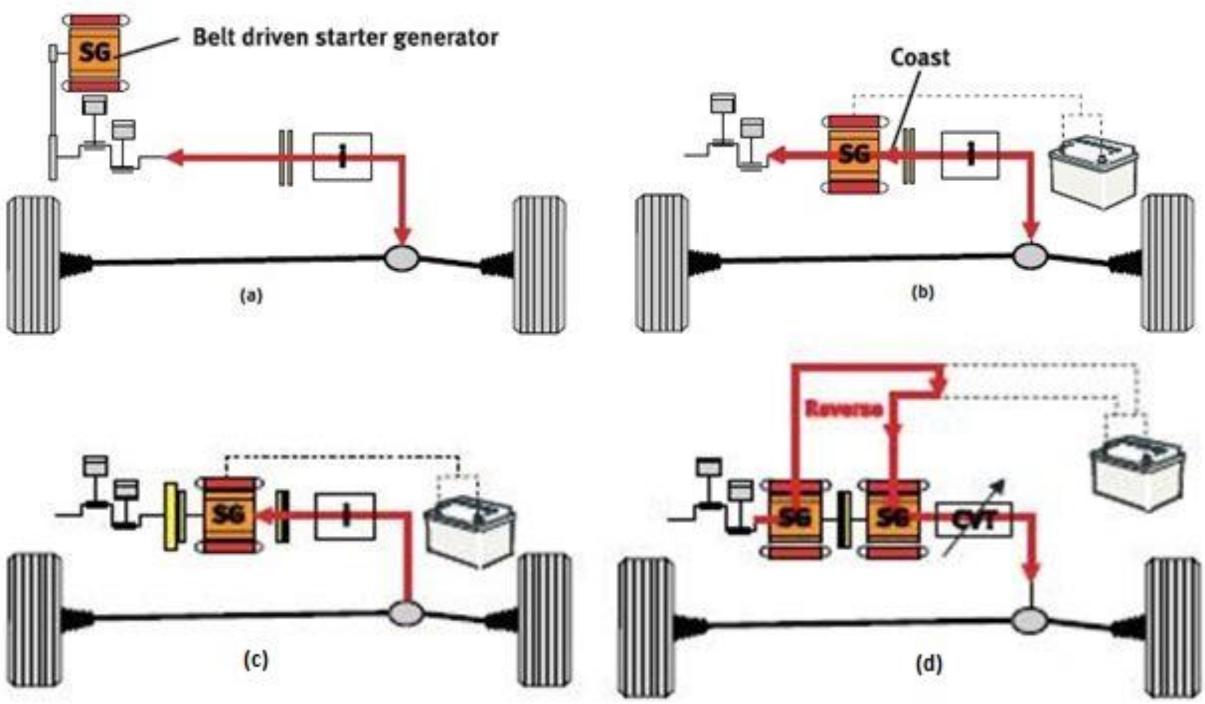

Figure 21: a) Belt Starter Generator, (b) Crankshaft Starter Generator, (c) Starter Generator between two clutches, (d) Reversal travel in serial hybrid with CVT. [19]

3.2 Crankshaft Starter Generator (CSG)

3.2.1 General overview

Figure 21(b) 'Crankshaft Starter Generator' (CSG) – demonstrate the hybrid powertrain used for the current investigation. This technology has pronounced its major step within hybrid technology for many OEM's, and it is considered to acquire the major market share in the coming years. And the reason it is popular, is because of its smooth engine start-stop function, which is more often used for better efficiency and driving conditions. It is even known to be the beneficial technology with respect to cost effective and one of the most adequate ways to reduce CO_2 emissions that works precisely and quickly.

The configuration described acts as a bi-directional power converter, i.e. exchanging mechanical energy into electrical energy and vice-versa. It works as an electric motor that starts an ICE almost soundlessly and considerably faster than any conventional starter. And as a generator, it produces power for various electrical consumers in the vehicle, but with higher capability compared to conventional systems. [19] With the CSG, an electric machine is mounted directly on crankshaft i.e. placed right in between combustion engine and transmission, as shown in Figure 21(b). The main objective of this powertrain configuration is its start-stop functionality, and if further upgraded with sensor technology and energy storage system, it facilitates important functions like recuperation, coasting and electric driving i.e. Parallel/Power-Split HEV. Figure 22, represents an example of CSG powertrain development from ZF.

Figure 22: Example of Crankshaft Starter Generator – 'Dynastart', ZF (http://www.zf.com/)

3.2.2 Effects related to CSG

For past many years noise and vibration (N&V) of complete vehicle has been the key issue for most of the OEM's. However, significant efforts are still invested in order to tackle noise and vibration issues. Nevertheless, the effects related to HEVs are different compared to conventional vehicles, due to frequent starting and stopping of the engine and electric motor to conserve fuel and with the increased complexity of the powertrain. The operation of an ICE is relatively different in current configuration, due to start-stop function, which stresses out not only ICE but even the components associated with the system.

However, in normal case the ICE is usually divided into four operating states: cranking (key start), idle, engine on, and engine off. [19] The current work only focuses on non-stationary conditions, i.e. during start-stop action, where the vibration from the engine occurs in both before and after ignition. When there is no ignition, there is compression and pumping pressure in the cylinders. After ignition, there is sudden torque change in the engine due to combustion. In present work, the investigation is performed only when there is no ignition, i.e. analyzing the rippling effects on engine starting phase, where the electric machine cranks an engine, in particular:

Torque Ripple (Cranking Effect): The model proposed in this chapter is based on a CSG configuration, where an electric machine is directly coupled with the crankshaft. The engine-starting phase is subdivided into two different stages. First stage, where the engine is cranked by the electric machine and the second stage, where the ignition takes place. However, during the cranking stage, the starter grants the required torque to embark an ICE. This sudden torque excites vibrations in the powertrain due to compression reaction force and pumping pressure in the cylinders. As soon as the ICE reaches its targeted speed, ignition starts, which further leads to another vibration. Thus, this complete starting phase combines the vibration effect and it is further transferred to the driver via the vehicle structure. The phenomenon is accounted as an important factor in hybrid powertrain, due to the frequent restarting of the ICE especially during stop and go traffic.

3.3 **Frequency weighting & Vibration Dose Value (VDV)**

Frequent vibrations are a major concern for human beings on day to day basis. Furthermore, measuring this vibration on standardised level is even more uncertain, due to its complexity. Hence to designate these measured evaluations, the weightings were generalized by ISO.

In the current simulation model similar approach is considered due to frequent restarting of ICE, where the severity of vibration is simulated based on standardised weightings. To further model the standardised weightings, the frequency weighting filters are necessary, which are designed and evaluated according to the defined frequency range i.e. with the help of sixth-order Butterworth filter, while using its upper and lower limits. The Butterworth filter is an ideal filter for not only completely rejecting the unwanted frequencies, but also supports to uniform the sensitivity of a given wanted frequencies. The Butterworth filter has flat response in passband and rolls off towards zero within the stopband, which is different when compared with Chebyshev 1&2 and Elliptic filters. Thus Butterworth filter is highly recommended to calculate frequency weightings in general.

Figure 23, shows an example on how the frequency weightings are used to filter the acceleration produced from different axis.

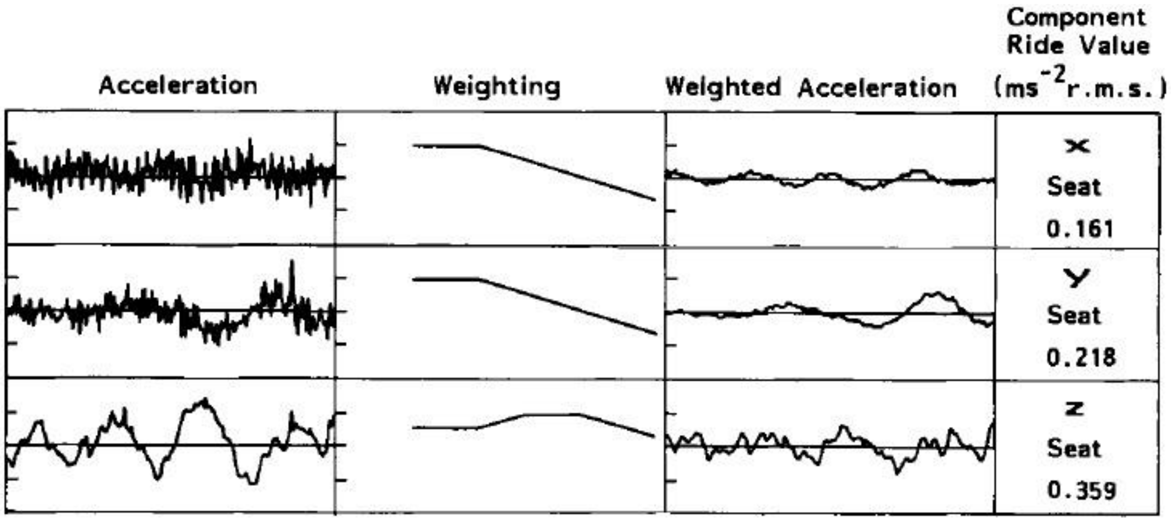

Figure 23: Example of frequency weightings on different axis. (M.J. Griffin, 1990)

Figure 24 and 25, describes the similar approach made to calculate the weighting curves as mentioned by ISO 2631-1 and M. Griffin, which guides in measuring vibration contribution along different directions. In practice, weighting methods have proved to be necessary for consistent assessment of vibrations regarding their severity for measured data. For current model evaluation of vehicle acceleration, the weighting used are Wc, Wd, We and Wk. [22, 23]

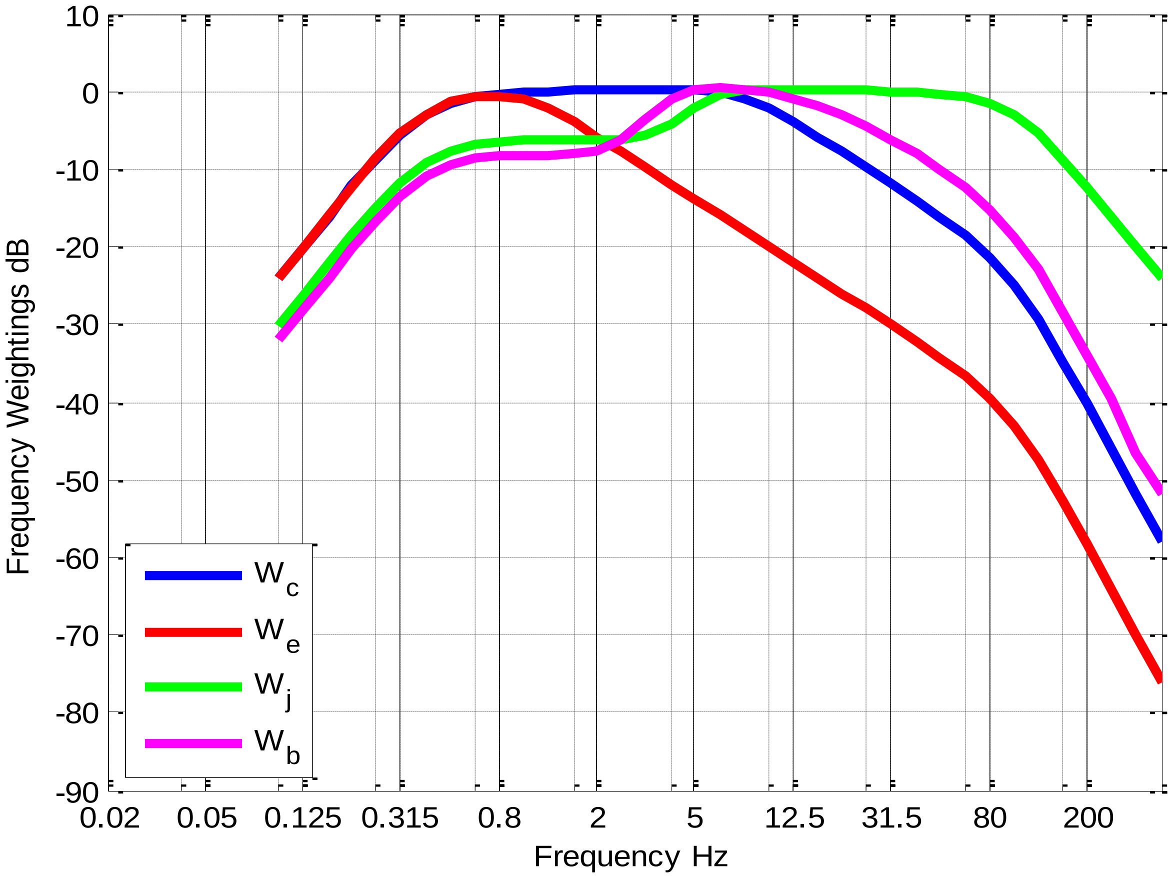
Frequency Weightings dB
10
0
-10
-20
-30
-40
-50
-60
-70
-80
-90
0.02
0.05
0.125
0.315
0.8
2
5
12.5
31.5
80
200
Frequency Hz
W_c
W_e
W_j
W_b

Figure 24: We is used for Rotational axis and Wc is used for Longitudinal axis, Wb is used for interference with activities, Wj is used for hand-transmitted vibration.

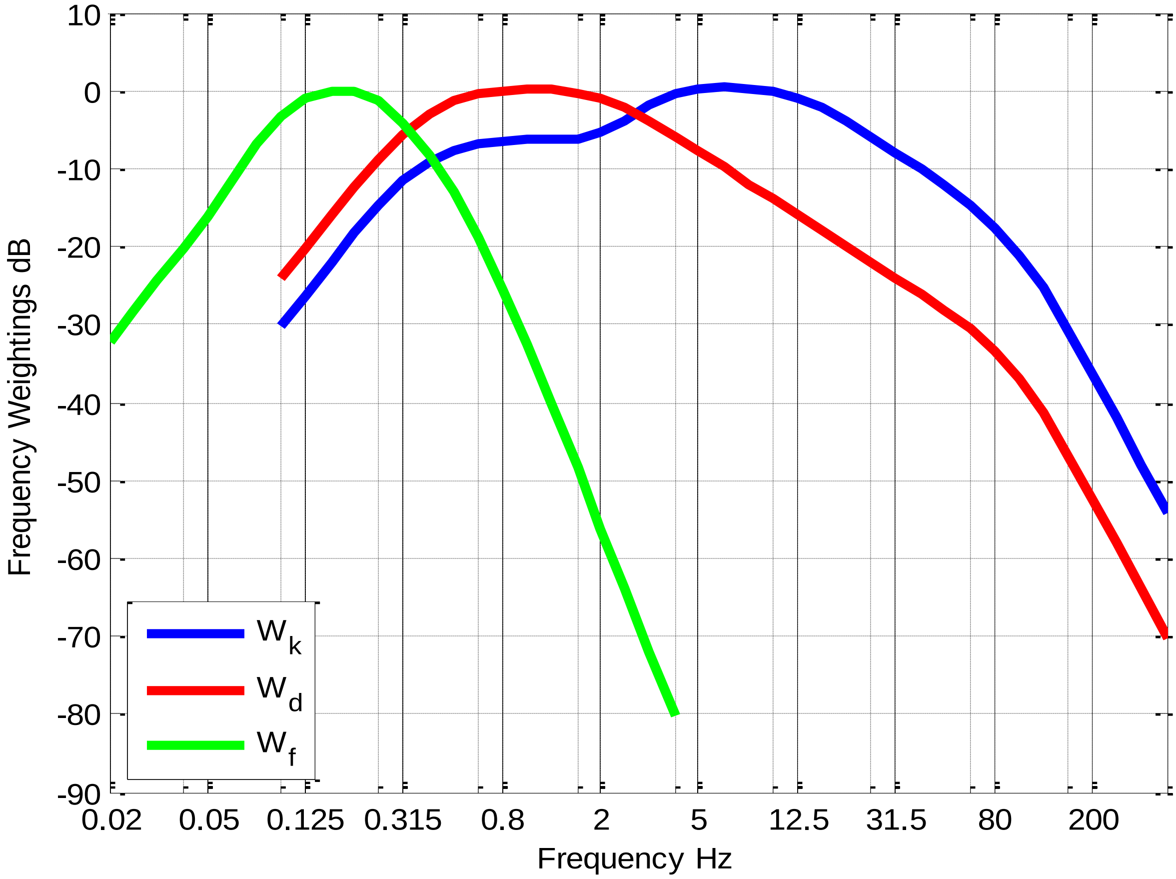

Figure 25: Frequency weighting curves, where Wk is used for Vertical axis, Wd is used for Lateral axis, Wf is used for motion sickness.

Moreover, to further assess the severity of vibration within the system, different methods are depicted. This method is recognized when the crest factor limits are exceeded, especially for transient, shock and non-stationary motions, for instance. The common methods used to quantify vibration magnitudes are Root Mean Square value (RMS), Maximum Transient Vibration Value (MTVV), Peak-to-Peak Value (PTP) and Vibration Dose Value (VDV).

In some cases, the motions are inconsistence with intermittent periods of vibration, where neither peak nor average measures reflect the duration of motion event: usually the peak value is determined by the magnitude at one instant while the r.m.s. magnitude can either increase or decrease with increasing duration. For this inconsistence motion whose vibration characteristics vary from moment to moment, and where the determined r.m.s. magnitude is not always possible

(=can yield to different conclusion), in this case cumulative measures are more reliable commonly known as ‘dose’ (M.J.Griffin, 1990). Thus to quantify high crest factor motion in our simulation model the vibration dose value is considered to be an appropriate method.

- **Vibration dose value** – It is the dose value which is achieved mathematically by integrating the fourth power of the frequency-weighted acceleration, over the period of motion. Unlike other methods, the VDV incorporates both a frequency weighting and a duration weighting. [18] The VDV increases with increasing measurement time.

$$VDV = \left(\int_0^T a^4(t)dt\right)^{\frac{1}{4}} \tag{1}$$

Equation 1 defines the vibration dose value, where *T* is the time history and a is the frequency weighted acceleration. The VDV is measured in $ms^{-1.75}$ and is generally used to measure the different discomfort levels for whole body vibration.

3.4 Driver seat vibration

The popular image of a ‘comfortable seat’ is that of a soft-cushioned seat having a luxury appearance and feel and giving an instant sensation of conscious well-being. (M.J. Griffin, 1990) However, the evolution of an optimum design usually takes into account many factors including:

- The range of seat user’s e.g. physical dimensions.
- Activities involved e.g. reading, relaxing and driving.
- Maintenance of seat e.g. wear, tear and cleansing.
- Environmental aspects of the seat e.g. thermal and dynamic properties.

This chapter is primarily concerned with the evaluation of the dynamic response of seats, when driving. As the seat is not rigid, it will modify the vibration entering the body and the exposure will depend on the location of vibration sensor. The vibration may occur in more than one axis and it may enter the body at more than one location, as in this case of a seat and a back support. The driver feels the structure borne vibrations through its contact on vehicle such as via steering wheel, pedals and in particular driver’s seat, where the majority of them are sensed. In general the standard describes a 12-axis measurement procedure for comfort evaluation of a seated person, which includes the acceleration data from the backrest, seat and feet. [22]

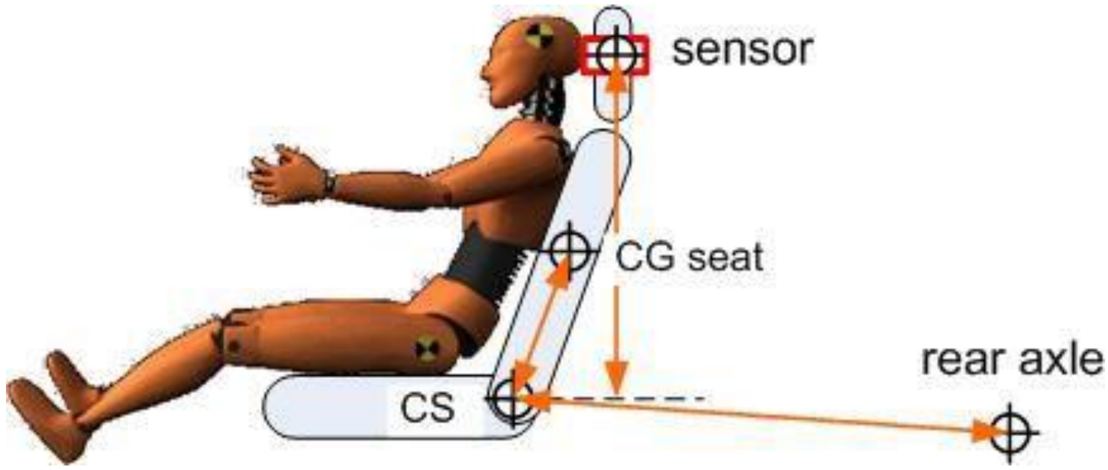

Figure 26: Driver Seat Sensor (AVL VSM)

Figure 26 shows the replica of driver's seat, describing the seat sensor position with respect to center of gravity (CG) and vehicle.

3.5 Simulation models and Results

3.5.1 Simulation models

The simulation models described in the current work are an extension of previous models mentioned in Chapter 2 and published by Parmar et al. [25] The model uses the CSG configuration; where BOOST RT explores engine thermodynamics, detailed crankshaft and electric machine are modelled in EXCITE PowerUnit, and Driver and Vehicle model are represented in CRUISE, in order to represent the driver seat within the model AVL VSM is used. AVL VSM is an additional implementation within the model, especially to support the driver related dynamics, in our case to simulate seat behaviour. Figure 27 shows the general overview of the complete model including AVL VSM.

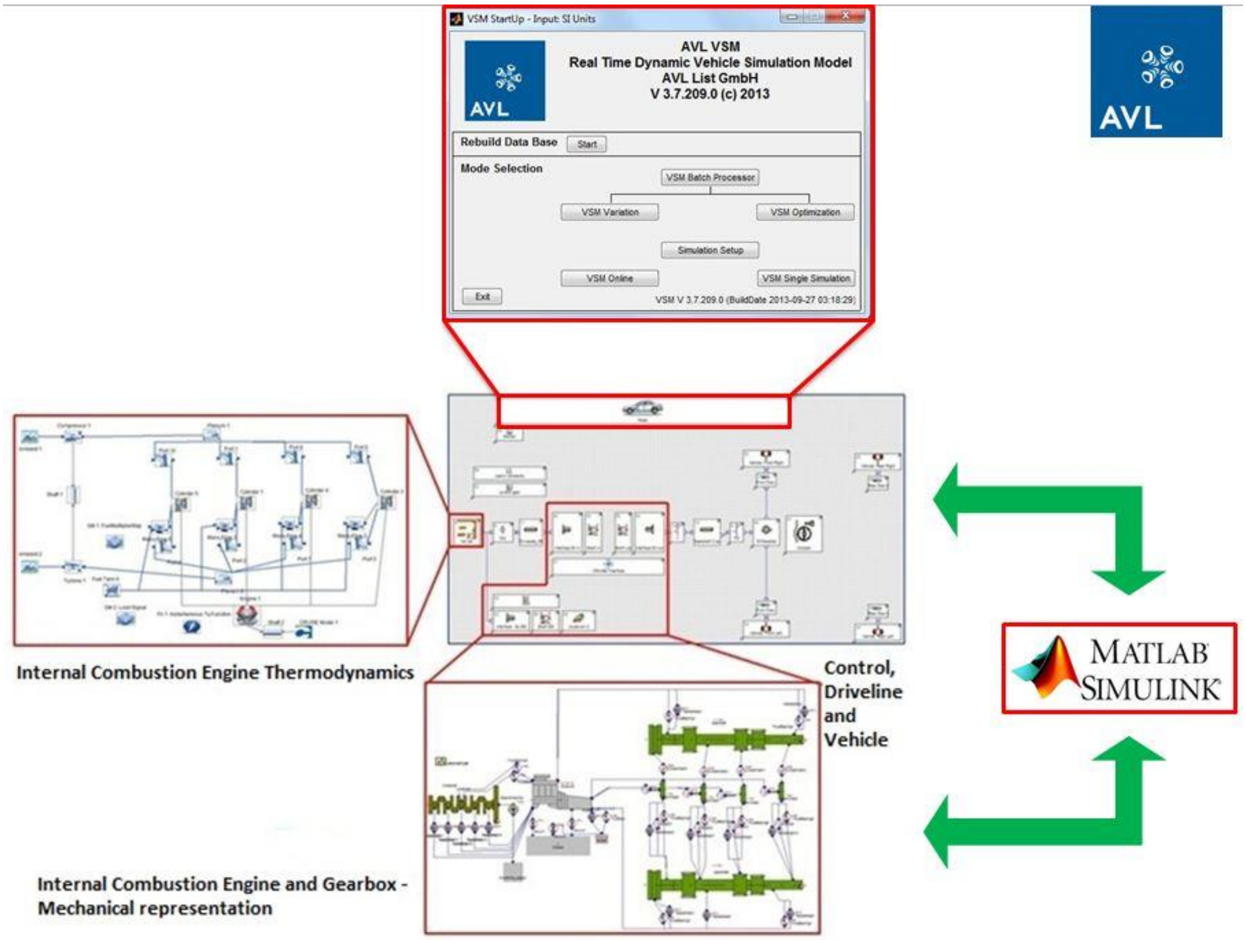

Figure 27: Overview image of complete model in co-junction with VSM

Below Figure 28, represents a 4 cylinder turbocharged direct injected gasoline (TGDI) engine, which comprises of turbocharger components, injection system, exhaust system, emission control system. The model implicates a detailed crank angle resolved cylinder model, while embedding the air path via filling and emptying approach, due to transient operation. A basic ECU system is build up to control engine air aspiration and injection by governing throttle and waste gate position, intake valve phasing and injection shape. [26] The major difference from the current engine model to the one mentioned in pervious chapter is: in this assessment a detailed thermodynamic model is developed, in order to define and achieve a thorough investigation on Engine starting behaviour and its effects related to it, however in this case only cranking phase is analysed.

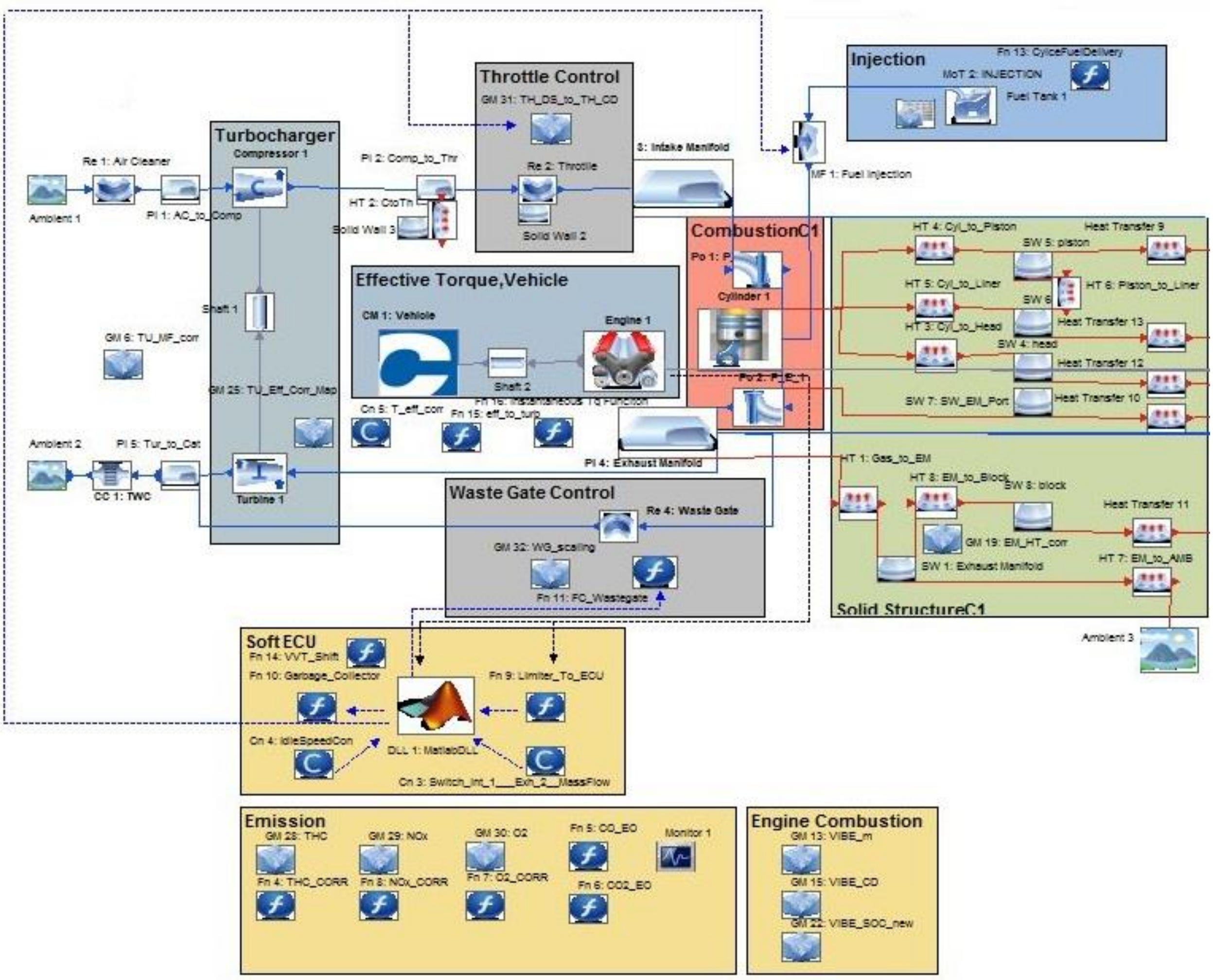

Figure 28: AVL BOOST RT – Engine thermodynamic model (Crank angle resolved).

For further dynamic investigations, a multi body dynamic model is build to introduce detail crankshaft and electric machine and their in-between coupling. The model represents a condensed flexible crankshaft body that defines 6 Degrees of Freedom (DOFs). To stabilize and give strong and stiff support within the crankshaft environment, bearings (joints) and powerunit are modelled, which serves to represent the engine/gearbox housing (as shown in Figure 29). In addition to build the CSG system, an electric machine (PMSM) is coupled in between crankshaft and powerunit integrating fundamental wave approach with an inbuilt inverter, sensor, battery and control system. The torque control system is used to handle the overall control system of electric machine i.e. on and off switch.

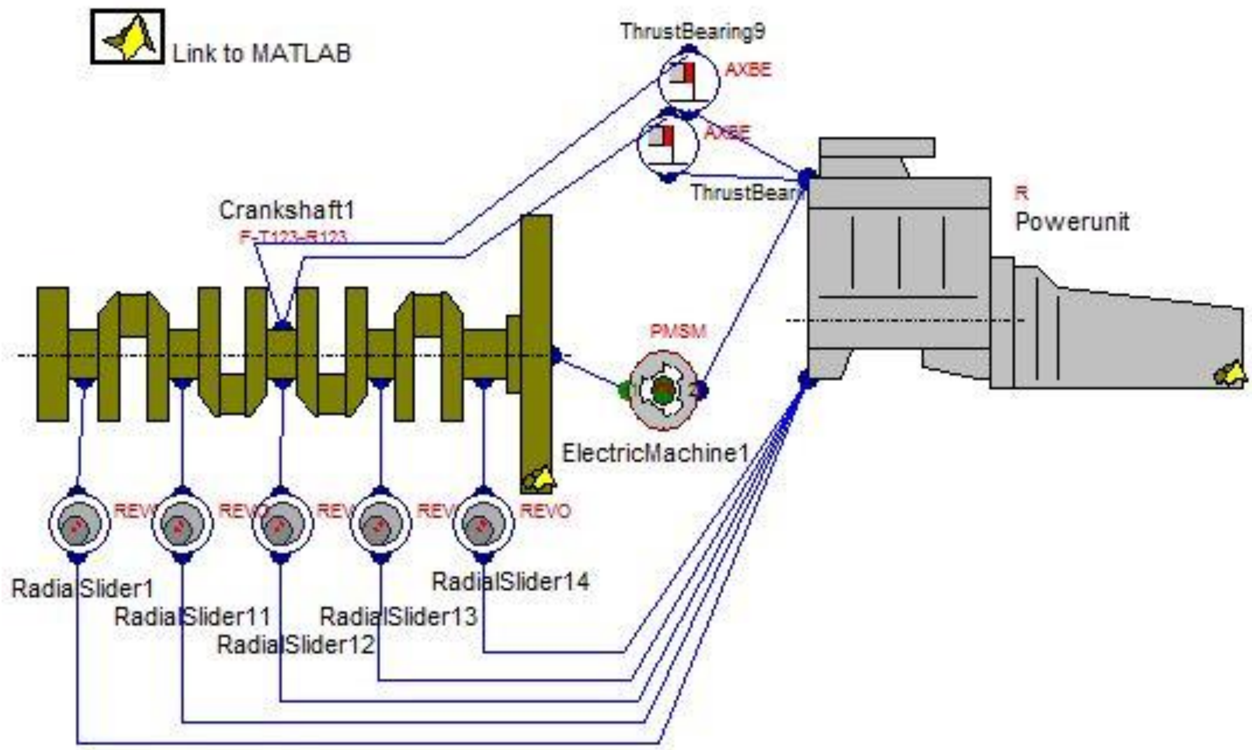

Figure 29 : AVL EXCITE – Dynamic CSG model.

Moreover, In order to represent a realistic vehicle model scenario, a complete hybrid vehicle layout is implemented, by means of AVL CRUISE as shown in Figure 30. As mentioned earlier, for the representation of the driver’s seat sensor, as well as for the support of other drivetrain components (engine mounts, chassis representation), are done by means of AVL VSM. The placement of driver seat sensor is as shown in Figure 26. The data mentioned within the simulation models are application based assumption values, however the parameters defined are completely in accordance with OEM’s criteria.

In addition, similar workflow of chapter 2.3.4 is considered to build the co-simulation of complete vehicle model, with an exception of detail transmission model, which for this assessment is simplified within the vehicle model, due to the main focus on engine cranking phase.

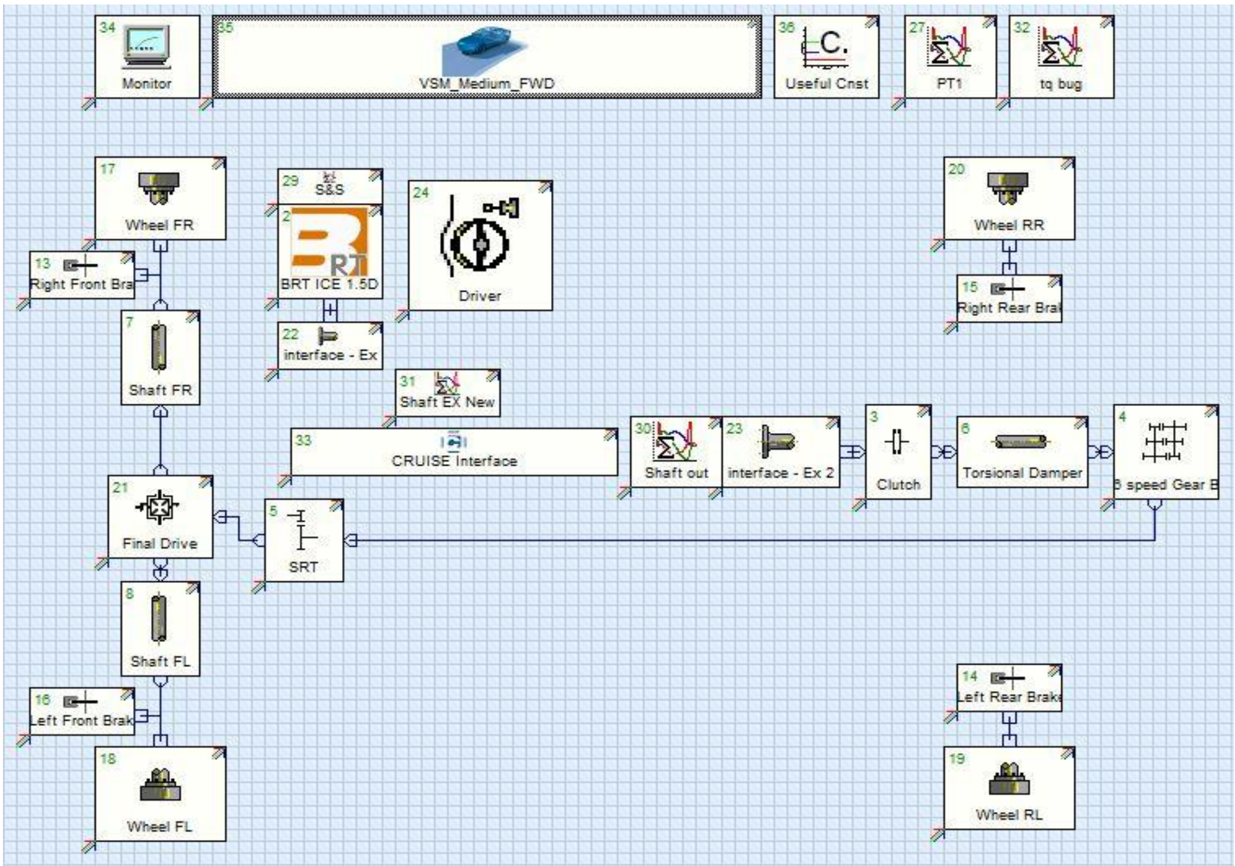

Figure 30 : AVL CRUISE –Vehicle Model

3.5.2 Results and discussion.

The results discussed below have a prime target on the vibration caused by starting phase of the ICE and further its dynamic relation with the driver's seat. The simulation is carried out in the time frame of 2 seconds, focusing on the cranking phase (first part of the engine start), where the electric machine grants the required torque to the crankshaft. As stated, this sudden torque excites the movement within the ICE (source), which is further transferred to the connected components such as whole powerunit, chassis and car body (path), and finally felt by driver through seat (receiver) via structure borne vibration.

Although, the occurrence of complete vibration event takes just a fraction of seconds, but the shock observed is much higher compared to the conventional vehicles, both for the components related to the engine block, and for the driver himself via driver's seat. The period of particular interest is from 0.02 – 0.20s, where the electric machine guides the ICE to ramp-up at a certain speed, while providing an elevation torque with a peak of around 165Nm.

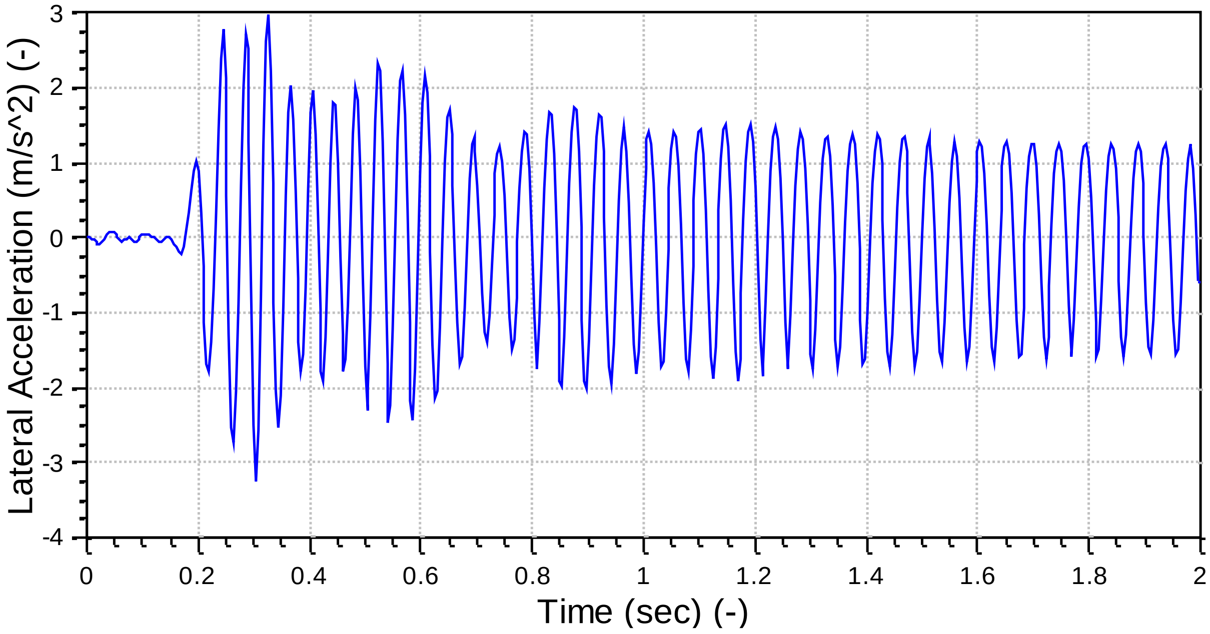

Figure 31: Lateral acceleration (y-axis) at seat sensor position.

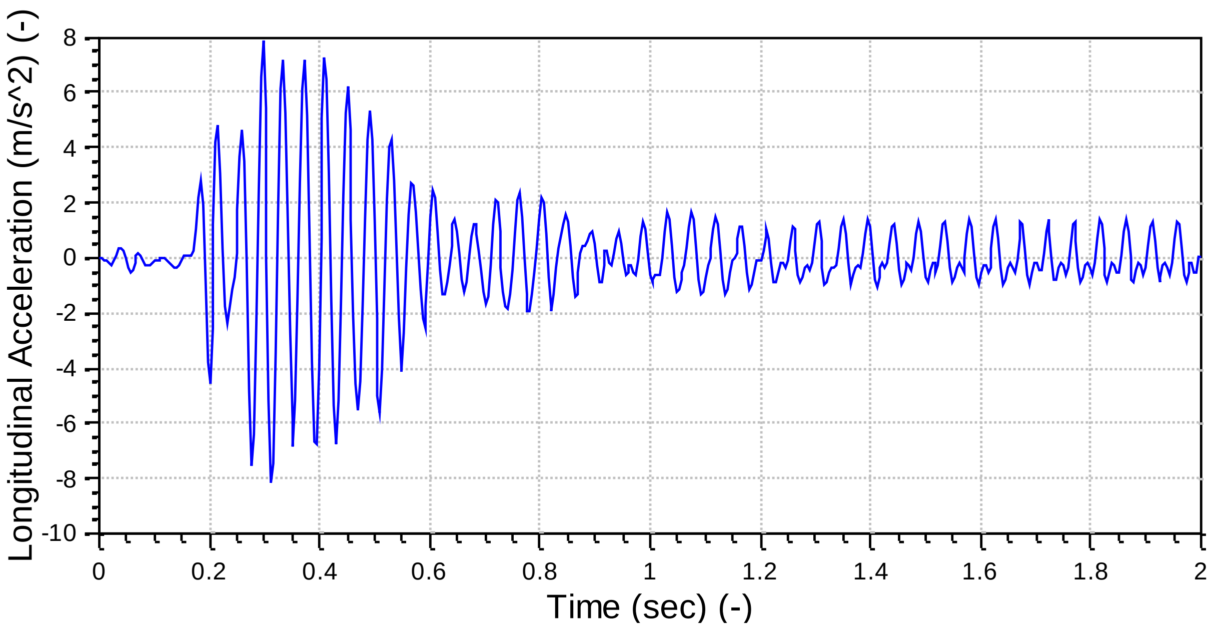

Figure 32 : Longitudinal acceleration (x-axis) at seat sensor position.

Figure 31 represents the lateral (y-axis) acceleration simulated at seat sensor position (sensor position shown in Figure 26). Figure 32 shows the longitudinal (x-axis) acceleration simulated at seat sensor position. The simulated values on lateral and longitudinal acceleration (y and x axis) usually contain a relative small acceleration range when compared with z-axis, and have the least impact on the vibration propagation and discomfort of the driver.

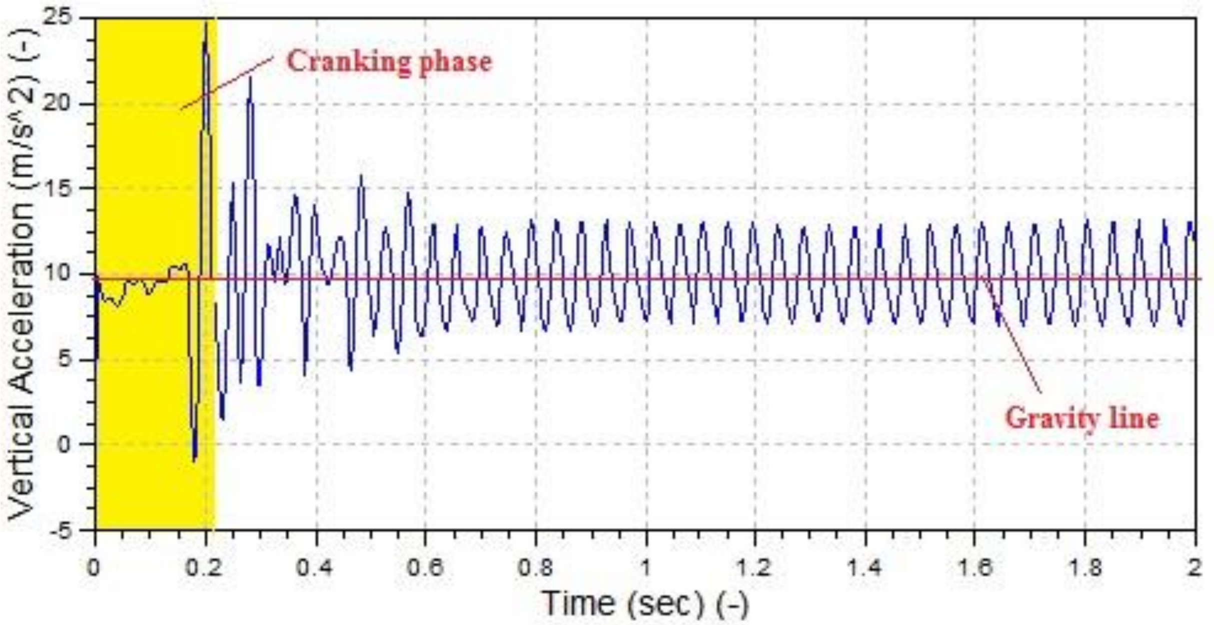

Figure 33: Vertical acceleration (z-axis) at seat sensor position.

Figure 33, represents the vertical acceleration (z-axis) measured at seat sensor position, which is relatively high (peak within yellow colour period), as it defines the magnitude with respect to gravity. The area highlighted in yellow represents, as mentioned before, the cranking phase of the engine, which is ramped up by the electrical machine. Figure 33, gives the lower ($-1.417ms^{-2}$) and upper range ($24.715ms^{-2}$) of acceleration magnitude, which indicates greater sensitivity when compared with x and y-axis accelerations. The jerking mode formed during this period, creates a major impact due to sudden torque transfer. This low frequency vibration impinges human body more often, due to frequent start/stop events.

In addition to it, the involvement of combustion phase can add more vibration (knocking) to the system, which all together increases the acceleration period range and so does the impact to the driver. However results shown above are only concentrated on cranking phase and not on the combustion phase, which can be the part of future work for deeper analysis.

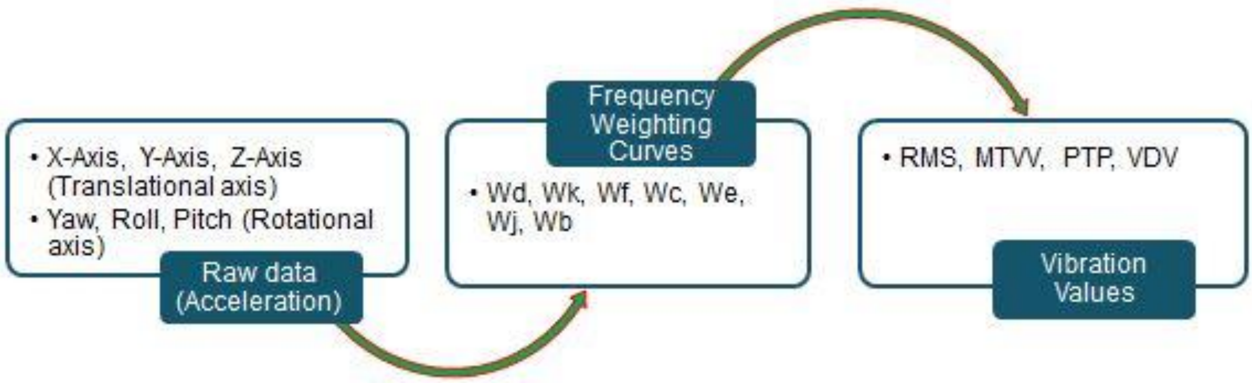

Figure 34: Flow chart for vibration data processing.

To further analyze this source impact vibration on human body sensitivity, the vibration values are calculated. For evaluation of these values, the frequency-weighting curve is used, which filters the simulated raw acceleration, as shown in Figure 34. The weighting curves are further distinguished according to the measured axis. However, for current investigation, only vertical acceleration (z-axis) is analyzed and filtered, due to its high magnitude and as it represents the direction, in which the dynamic effects are most perceptible for the driver.

For further assessment of the whole-body vibration, the vibration dose value is utilized (as shown in figure 34 the workflow to evaluate VDV). This is resolved by using frequency-weighted acceleration and fourth power integration method, as mentioned in chapter 3.3 and equation 1.

Below Figure 35, represents the VDV integrated from the vertical acceleration at seat sensor position. As form the results, it is encountered that VDV in Z-direction had higher impact compared to X and Y directions. The period at which an engine is cranked, and which leads to the sudden growth of VDV (red arrow – Sudden Rise). The highest value measured for VDV was $3.297ms^{-1.75}$.

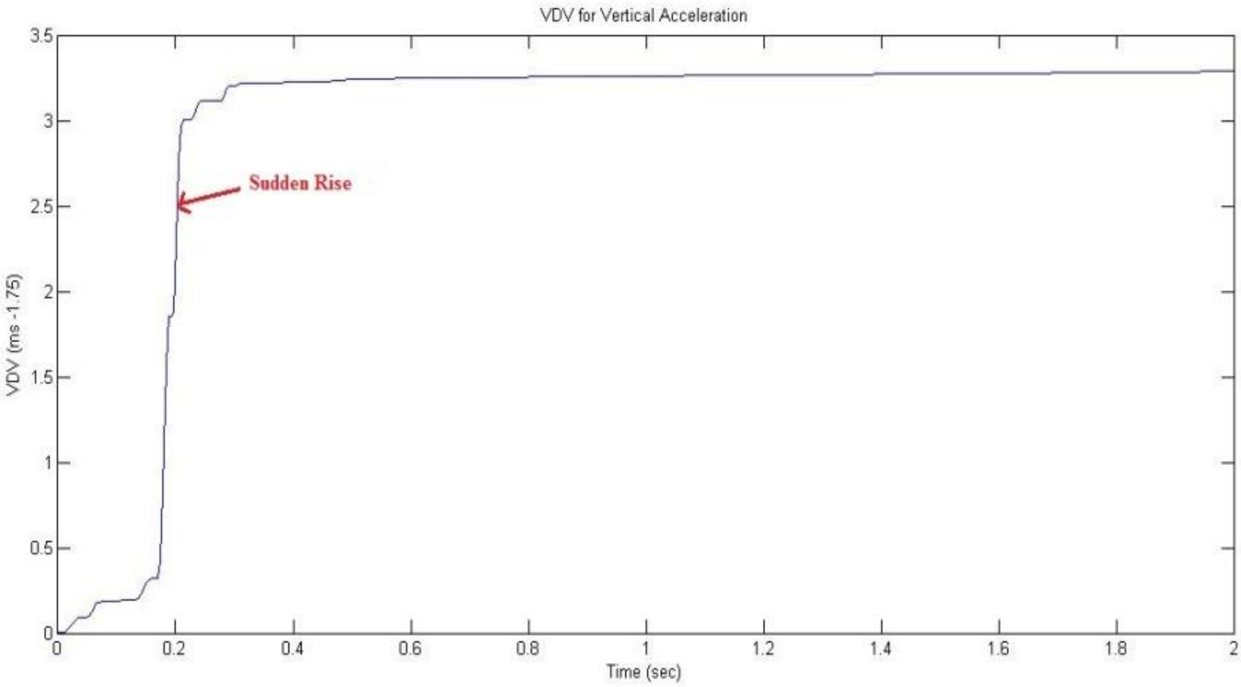

Figure 35: Vibration Dose Value for vertical acceleration (z-axis)

Furthermore, to sense the impact on complete vehicle system, the total value of VDV for all the directions (X, Y, and Z – axis) can be considered and can further increase the value when compared to the evaluation of single axis, This increase in value influences the deterioration of the comfort level for the driver due to repetition in time. As discussed by M.J.Griffin, at low frequency (below 1 or 2 Hz) the dynamics related to seats have little influence, but in the region of 4Hz, it noticeably amplifies vertical vibration, so that a tolerable 1 – 2 ms^{-2} r.m.s becomes an unacceptable magnitude. It is even been known that vibration magnitudes and durations that produce vibration dose values in the region of $15ms^{-1.75}$ will usually cause severe discomfort. Short and repeated vibrations (such as shocks, transient operations) close to $15ms^{-1.75}$ action level, which have high magnitudes, may appear unpleasant or even alarming to the affected person.

3.6 **Assessment conclusion**

The research work presented within this computation model outlines the 1st attempt in predicting seat vibration effects in micro/mild hybrid powertrain. The model further introduces the methodology of combining and coupling the different modelling domains within a comprehensive simulation framework and outlines first results of driveline vibrations. The schematic results shown above are based on the cranking of an engine, which describes the severity for whole body vibration. Based on an this observed severity, the vibration dose value was calculated, as VDV provide useful indications of subjective reactions to shocks (cranking engine) and intermittent vibration, where r.m.s. (root mean square) methods are not suitable.

For future aspects, the simulation models can further be extended in terms of evaluating not only start cranking phase operation but also combustions phase. This complete starting phase together will emerge bigger impact on the system. In addition for complete driving manoeuvres where even stop of an engine is equally as important as like start of an engine, due to frequent start/stop strategy.

Moreover, few more topics related to powertrain mounting and electric whining noise can also be taken into account for further N & V evaluation, which includes different scenarios like boost and recuperation. These vibration exposures to human body vary and differ mostly over time and even from one person to the other. The measuring mechanism (frequency weightings) involved for various effects gives firm conclusions, but there are scopes for greater understanding of all the phenomena's. Lastly, the simulated values within the assessment can further be isolated by means of a control algorithm, which helps to reduce the vibration magnitude and consequently enhance the comfort level.

4 COMPLEX TRANSIENT BEHAVIOUR IN COMMERCIAL VEHICLE

4.1 General overview.

As mentioned earlier, hybrid powertrains provides many advantages from fuel saving to energy consumptions and many more. One of the major advantage is start-stop or stop and go function, where the engine is turned off each time the vehicle comes to a complete halt - such as at traffic lights, or bus-stop halt – and again restarts automatically. Furthermore today's commercial marketplace offers highly efficient diesel engine and gasoline hybrid drives, but still diesel engines are highly restricted within the city environment, due to its exhaling pollutant. So to fulfil this requirement, diesel hybrid application would be the perfect solution for commercial vehicles, especially in case of Bus, which is known as the common public transport mode worldwide.

Therefore, for better understanding of behaviour in diesel hybrid application, a detail study is carried out in this final assessment, which represents the combination of first and second computation model. However, the target is to simulate and investigate the dynamics of complete vehicle model, keeping in mind driver's profile i.e. stop and go scenario in Bus application, where it is quiet common for a Bus to halt for few seconds at Bus-stop station and start again to drive.

In technical terms the dynamic scenario is described as: Vehicle stop → Engine standstill → Vehicle start using e-machine → Activating ICE to a certain vehicle speed → Turning off e-machine → Shifting from 1st to 2nd gear.

Thus to investigate above mentioned scenario a detail model is developed within this assessment. And further to ease its complexity and for better understanding of dynamic behaviour it is divided into sub-models. The sub-models consist of:

- Internal combustion engine – 6litre, 6 cylinder Diesel engine.
- Lepelletier six speed automatic transmission.
- An electric machine for hybrid powertrain.
- And vehicle model with driver profile controllability.

As mentioned earlier, the co-simulation (coupling) of complete vehicle model including sub-models is very much similar to earlier two assessments. However, the main focus here for this third assessment, is to study and understand the dynamic behaviour of stop and go scenario within hybrid powertrain in our special case for diesel application.

4.2 **Sub-models structure.**

4.2.1 Internal combustion engine

This sub-system represents 6litre 6cylinder turbocharged diesel engine in steady state as shown in Figure 36. The model shows the simplified model of turbocharger, i.e. compressor and turbine with basic input parameters for simplified turbocharged model. Whereas compressor target the pressure ratio as the most essential input and the turbine will act according to this setting, having aim to achieve specified pressure ratio. In this model it is done by waste gate calculation, which is used to let some of the gasses to bypass the rotor. Mechanical efficiency of turbocharger is considering when turbine power is driven to compressor. The following engine functionalities are covered by the model. (AVL Cruise M documentation)

- Engine turbocharging is modelled by full model of Compressor and Turbine connected with diesel engine,
- The fuel injection is modelled using the components Direct Injector and Fuel Tank, which injects fuel input directly to cylinder component.
- The engine air path model applies a filling & emptying approach in combination with a representative cylinder.
- The cylinder thermodynamics are calculated using a single zone model where the rate of heat release is approached by Vibe function.
- Finally the model is further connected to detailed dynamic crankshaft body coupled with Lepelletier automatic transmission model within Excite PU.

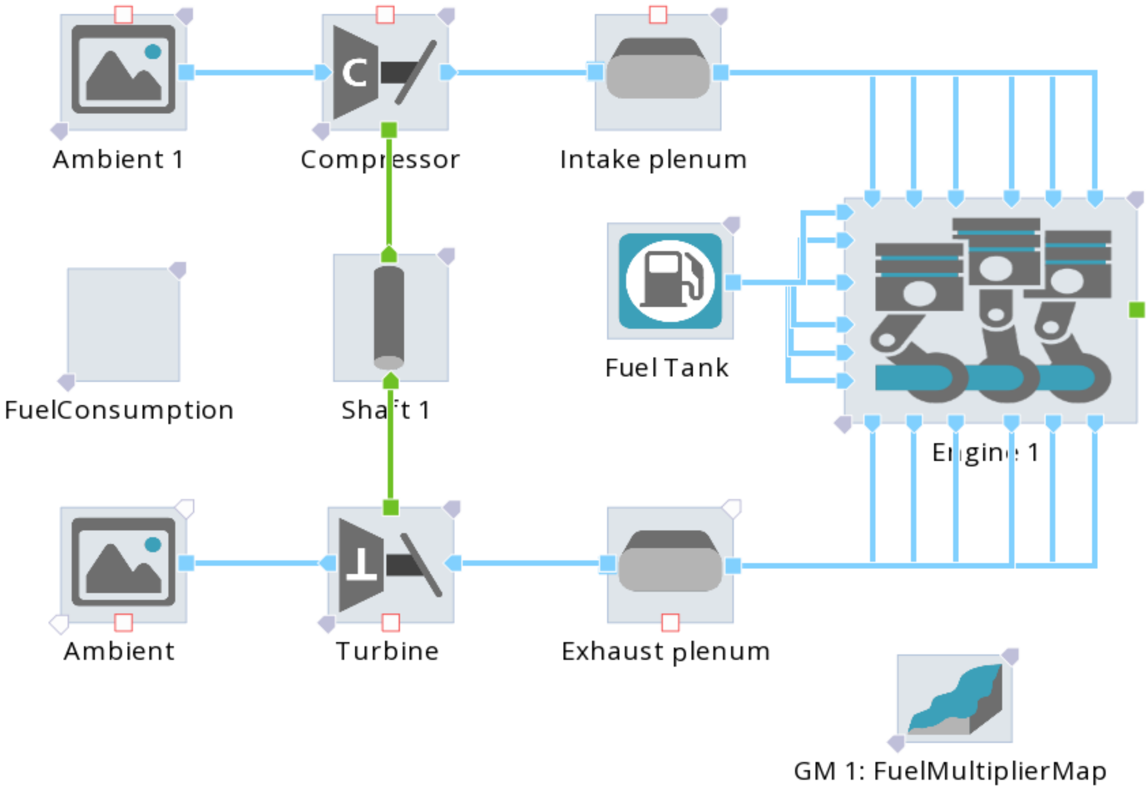

Figure 36: Cruise M Engine model layout

The model can further be divided into 3 major subgroups:

- The intake system
- The cylinder system
- The exhaust system

The intake system (Figure 37) – The intake system consists of following components: System Boundary - Compressor - Plenum. System Boundary ensures boundary conditions before intake system and represents state of the outside air. Plenums represent volumes of the system (of pipes for example), where also Heat Transfer can be considered. Intake air is compressed, so that higher pressure is available at intake plenum.

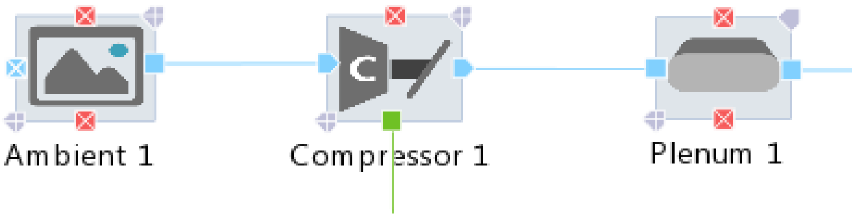

Figure 37: Intake system of engine model

The cylinder system – The cylinder system is enclosed in the Cylinder Block subsystem within Engine block, as shown in Figure 38. The Combustion Chamber is the most important part of it and it contains geometrical data and combustion information. It is connected to the intake and exhaust system via the Port components and their exposed connection ports. They contain valve lift and flow coefficient curves, as well as geometrical data. The heat generated inside of the Combustion Chamber is transferred to the engine structure that is represented by 3 Solid Wall components, representing the piston, cylinder head and liner, respectively. These components are connected to the Combustion Chamber via the Heat Transfer Connection components. It is a transient component that, depending on the components connected to it, includes information about contact surfaces and allows defining the heat transfer coefficient. Similar to that, the Ports are also connected to 2 Solid Wall components (representing the masses of the port materials) through Heat Transfer Connection components. The whole engine is modelled with six cylinders representing all of same. As the engine is HSDI (high speed direct injection), Direct Injector element is connected directly with combustion chamber. With this component, mass flow of the fuel is defined while considering heat of evaporation. (AVL Cruise M documentation)

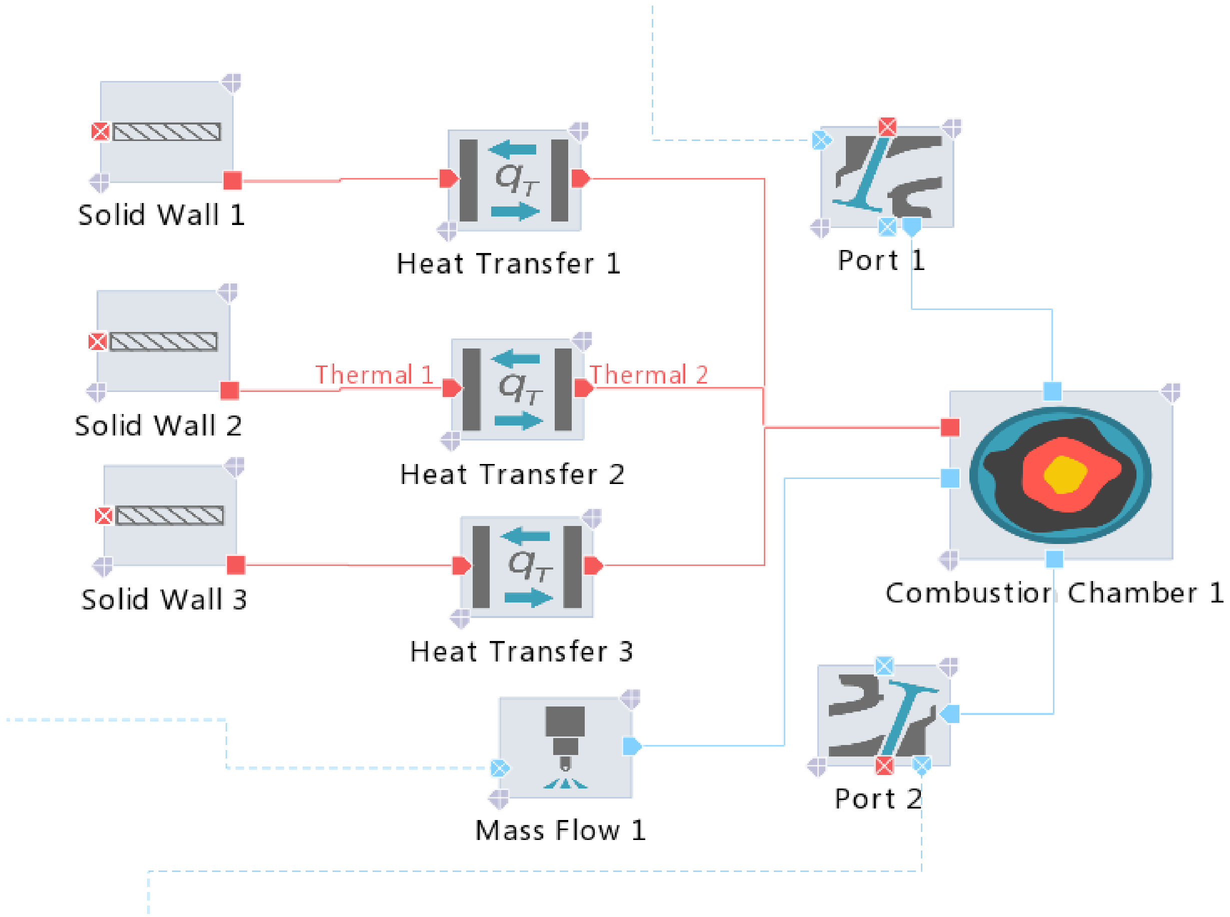

Figure 38: The cylinder system with transfer elements.

The exhaust system – Figure 39 represents the exhaust system, which is simply made of Plenum, Turbine and Ambient. Turbine transfers energy from the exhaust gasses to Compressor and Ambient boundary conditions are defined within the end of the exhaust system.

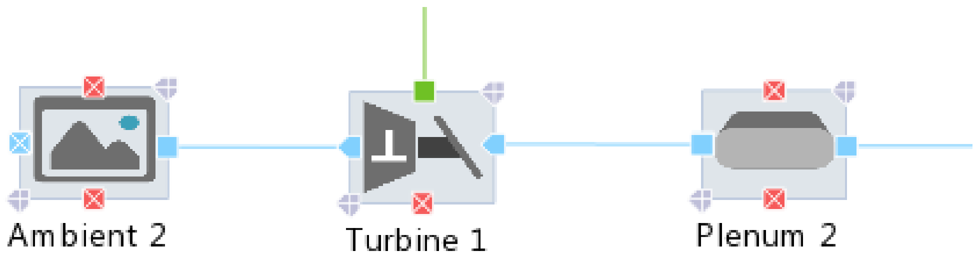

Figure 39: The exhaust system

4.2.2 Lepelletier automatic transmission

Different OEM's use different transmission systems. The most commonly used systems are Simpson planetary gearset and Ravigneaux gearset, which provide four speed and reverse transmission. The Lepelletier gearset principle is a combination of both Simpson and Ravigneaux, which transmits 6 speed and reverse gearbox, as shown below in Figure 40. Moreover, first to use such systems was ZF, who introduced „ZF6HP26"in 2001. [30]

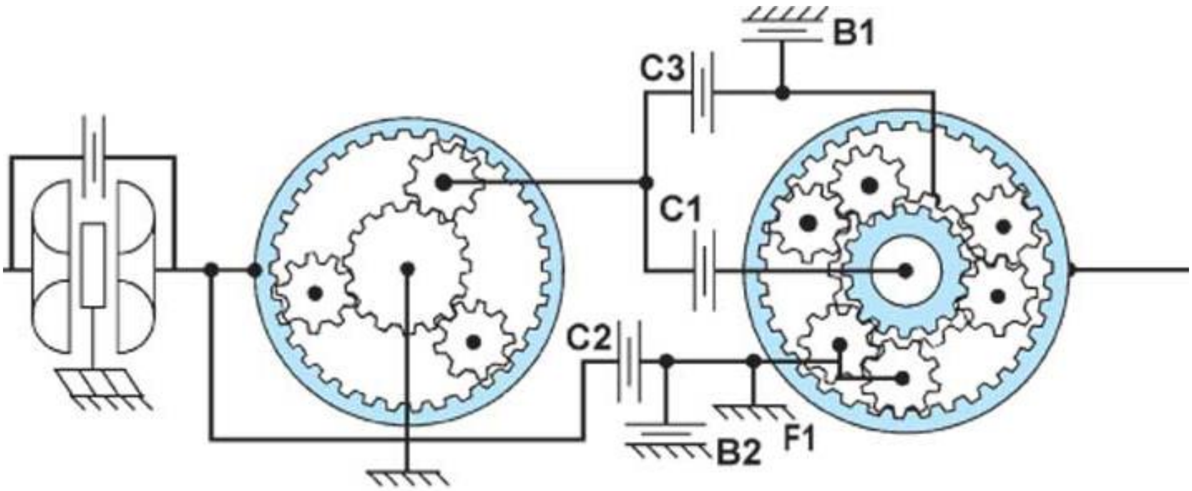

Figure 40: Representation of Lepelletier Gearset Principle (T.Birch & C. Rockwood, Automatic Transmission and Transaxles, 4th Edition)

The transmission arrangement represented in this assessment is based on 6-speed automatic transmission, whereby the clutch replaces the torque converter and requires no additional space for integration. This additional clutch is integrated between the combustion engine and electric motor. This ICE clutch (Figure 44) decouples the engine during pure electric driving. The model is further coupled with an electric machine (E – m/c), due to our motive for diesel hybrid application and to analyse stop and go scenario accordingly. The model demonstrates Lepelletier gear which combines: both Simpson gearset (PGS) and Ravigneaux gearset.

Simpson gearset: Simpson gearset (as shown in Figure 41) is defined as simple planetary gearset consists of sun gear, ring gear, planet carrier connected with planet pinion gears. To produce a powerflow, one part is the input, one part is held in reaction, and one part becomes the output.

Ravigneaux gearset: Ravigneaux gearset (as shown in Figure 41) is composed of two sun gears, one planet carrier that supports two sets of pinion gears and a single ring gear.

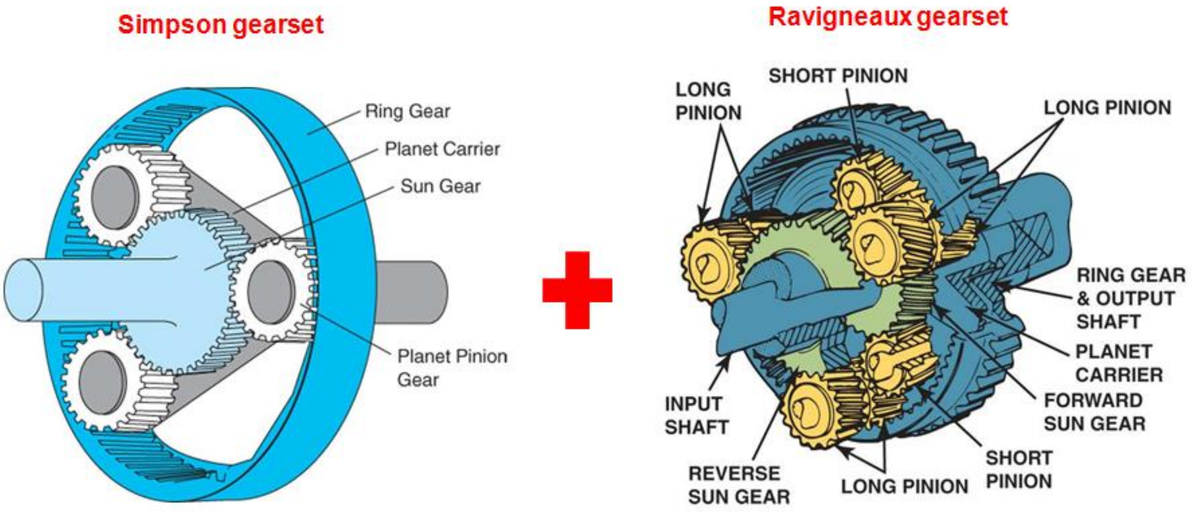

Figure 41: Lepelletier gearset: Combination of Simpson and Ravigneaux gearset (T.Birch & C. Rockwood, Automatic Transmission and Transaxles, 4th Edition)

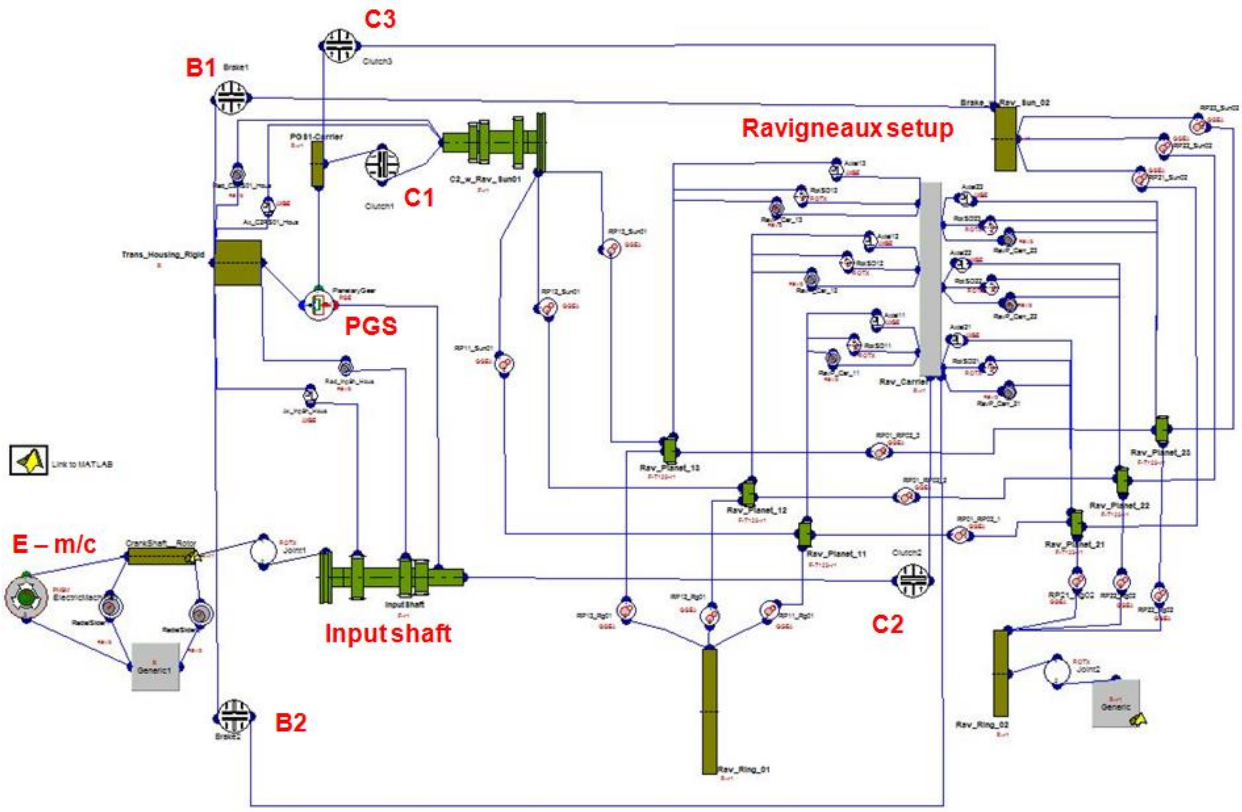

Figure 42: AVL Excite PU - Electric machine coupled with Lepelletier transmission system

Figure 42 shows the multi body dynamic model of a Lepelletier transmission consisting of flexible and rigid bodies, which are interlinked with non-linear force-based contact joints. The complete transmission model is defined with 6 Degrees of Freedoms (DoFs) for the shafts and gears in the model. The transmission Inputshaft further connected to ring gear of Simpson planetary gearseat (PGS) and Clutch C2 followed by Brake B2 towards Ravigneaux planet carrier. The sun gear of PGS is fixed with the transmission housing rigid body. The transmission flow of planet carrier body – PGS is further bifurcated: Clutch C1 to Ravigneaux smaller sun gear body and Clutch C3 of Ravigneaux larger sun gear body. Furthermore, Brake B1 is fixed with transmission housing rigid body and Ravigneaux larger sun gear body. Finally the ring gear of Ravigneaux gearset gives the output of transmission model, which is further connected to vehicle model.

The control of gear shift operation is done with elasto clutch and brake joints. These clutch and brake joints are engaged and disengaged via maps i.e. with help of shift operation table, as shown below in Figure 43, which gives the overview of 6 speed gears and reverse.

Lepelletier Gear Ranges							
Range	C1	C2	C3	B1	B2	F1	Ratio
1	X					X	4.15:1
Manual 1	X				X		4.15:1
2	X			X			2.37:1
3	X		X				1.56:1
4	X	X					1.15:1
5		X	X				0.86:1
6		X		X			0.69:1
Reverse			X		X		3.39:1

Figure 43: Shift strategy use for lepelletier transmission (T.Birch & C. Rockwood, Automatic Transmission and Transaxles, 4th Edition)

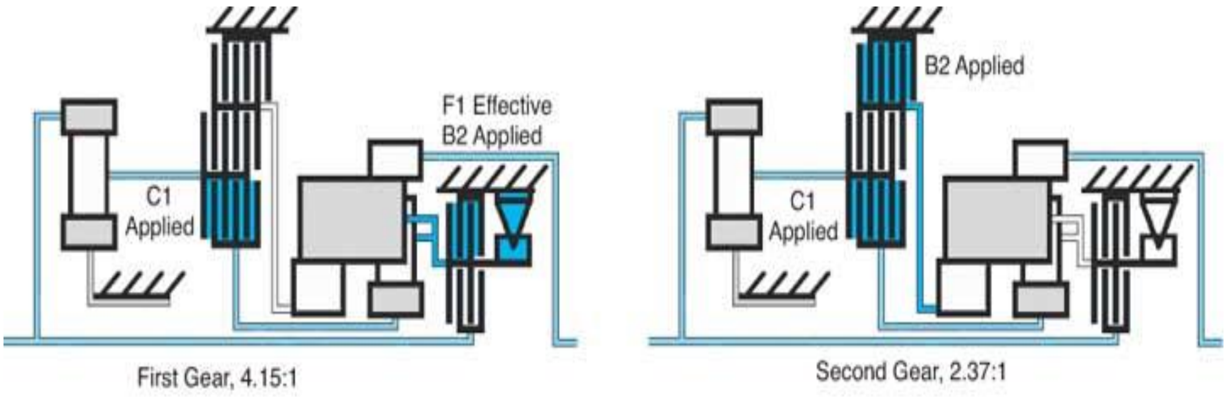

Figure 44: Powerflow of 1st gear and 2nd gear

Figure 44 details the powerflow of 1st gear and 2nd gear, as for current investigation only one gear shift is analyzed i.e. Upshift: Manual 1st gear → 2nd gear for low frequency effect.

4.2.3 Vehicle and driver model

Figure 45 represents the vehicle model. It consists of four sets of Wheels and Brakes, a Cockpit and the Vehicle (Manual RWD) component. The Vehicle contains general data of the vehicle, such as nominal dimensions and weights. Road resistances and dynamic wheel loads are calculated according to the profile and dynamometer runs, based on the vehicle dimensions and the load state. The wheel loads are calculated considering motion (e.g. from the effects of acceleration, aerodynamic drag, rolling resistance). The aerodynamic, rolling, climbing, acceleration and total resistance are calculated, too. The cockpit links to the driver and the vehicle. On one hand the driver(cockpit) gets information such as the vehicle velocity and the vehicle acceleration, on the other hand information from the driver such as the pedal positions are delivered to other components. The pedal positions (e.g. clutch pedal position) are transferred into corresponding indicators (e.g. clutch release) via the pedal characteristics (e.g. clutch pedal characteristic).The brake component is described by brake data and dimensions. The braking torque is computed considering the brake dimensions and the input brake pressure.

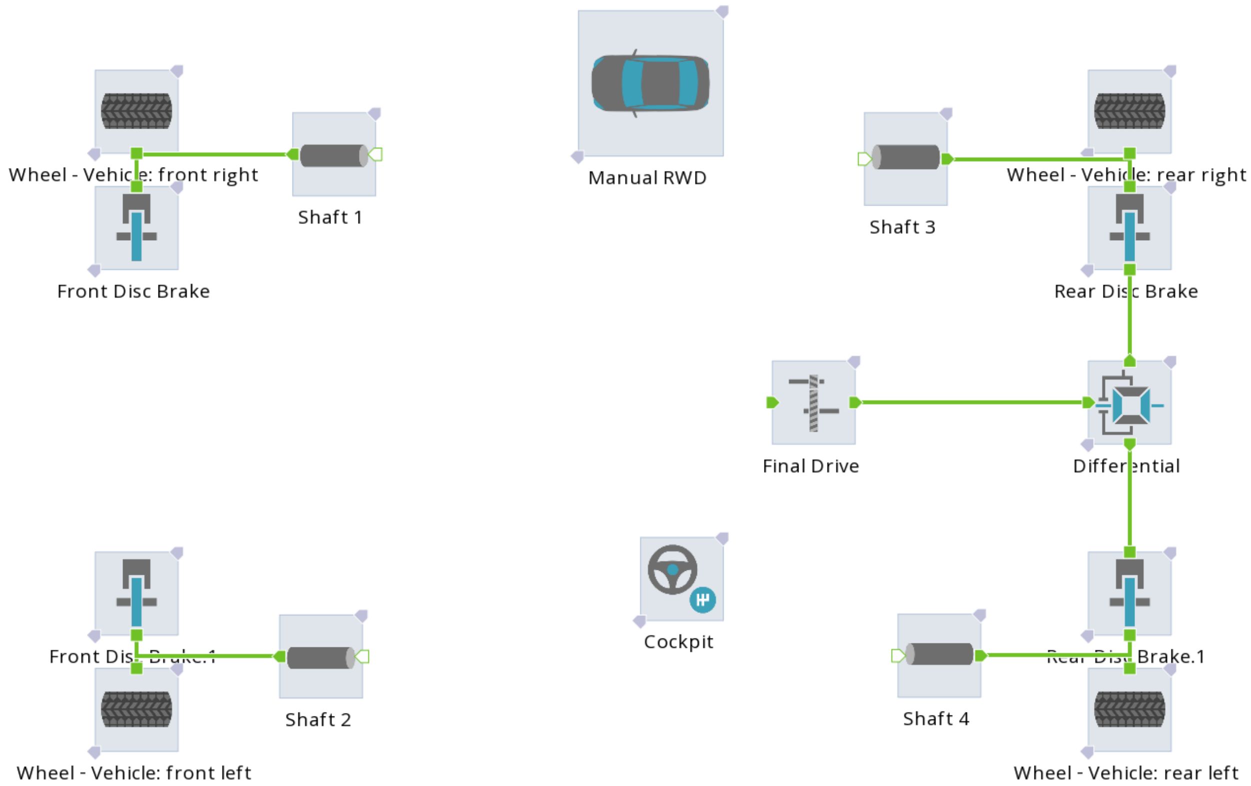

Figure 45: AVL Cruise M - Vehicle model.

4.3 **Model Panorama**

Figure 44, shows the complete co-simulation model overview evaluated for current assessment. The model starts with engine, which is further connected with ICE clutch controlled by map based input. The purpose of the integrated clutch is to couple and decouple the engine when required with the vehicle. Furthermore, the ICE clutch is connected with the Lepelletier transmission unit, with the electric machine placed between clutch and transmission. The electric machine is used to drive the vehicle, when ICE clutch is decoupled i.e. during the launching phase, where the vehicle has stopped and again starts to drive, according to driver’s profile. The output shaft of the transmission is further linked to the final drive and the complete vehicle.

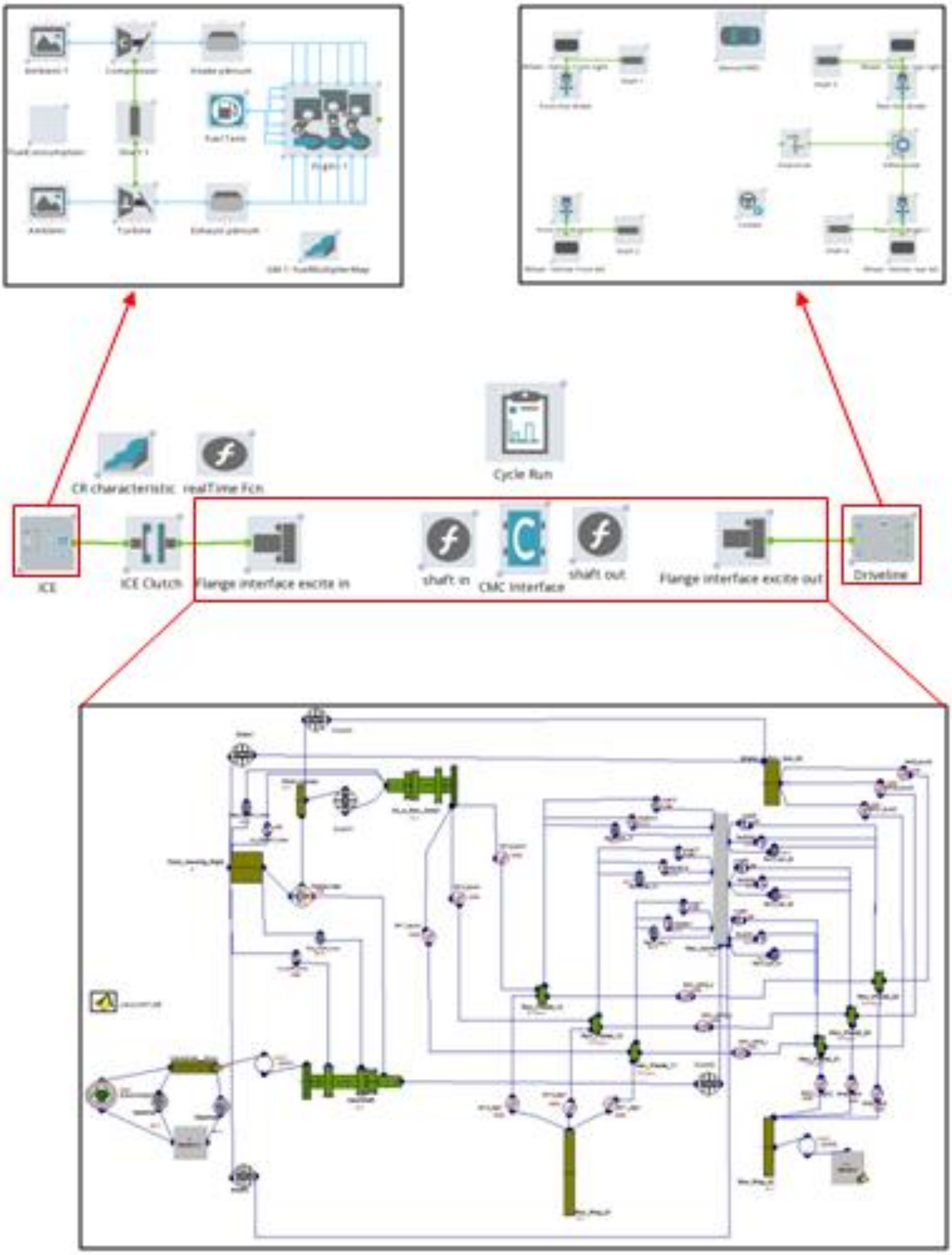

Figure 46: Complete co-simulation model

4.4 **Scenario’s and Results**

The results presented here illustrate a detail analysis on the complete vehicle model. However, the main focus of the simulation model is stop and go scenario, i.e. when driver brakes the vehicle and halts for certain time, and then again starts to move.

In this assessment three dominant phases are analyzed:

- First phase: Driver brakes the vehicle and halts for few seconds, during this phase the engine is at standstill position.
- Second phase: The vehicle starts moving after the stop. At this time only the electric machine drives the vehicle.
- Third phase: The engine ignites and its speed synchronizes with the speed of the electric machine . After this period, the ICE drives the vehicle and Upshift operation is considered.

Moreover to capture stop and go scenario the simulation runs for a period of 7 seconds, but the detail analysis are carried out only few seconds i.e. starting from 1 second to 5.5 seconds. Below Figure 47, represents the pre-defined driver’s desired velocity profile, starting from 15km/h. In order to firmly build and investigate this scenario: the vehicle begins with predefined velocity of 15km/h i.e. from 0 – 1 second, and in-between this period the Engine is ramped up till 15km/h to perfectly reach the predefined velocity. The reason this step is considered, so that the actual scenario can be interpreted for the later period i.e. from 1 – 5.5 seconds.

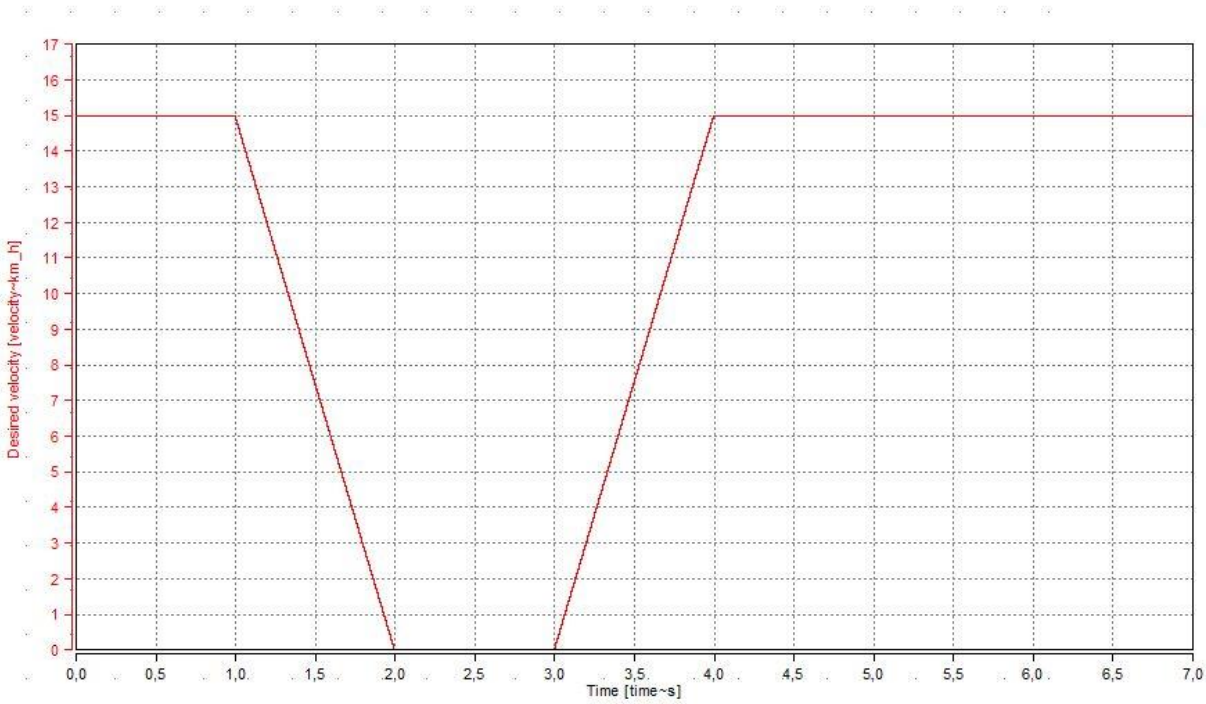

Figure 47: Desired velocity (Driver's profile)

Below Figure 48 showcases the driver's actual simulated profile, where:

- The engine which is driving the vehicle begins to brake at 1 second and reaches to halt position at 2 seconds.
- In-between 2 – 3 seconds the vehicle is standing still. During this period the engine is shut off and further decoupled from the vehicle with the help of ICE clutch.
- From 3 – 4.25 seconds the electric machine launches and drives the vehicle with its pre-defined torque.
- At 4.25 seconds the electric machine is deactivated, and on the other side the requisitely ramped up engine is coupled to drive the vehicle via ICE clutch i.e. starting from 4.25 – 5.5 seconds.
- From 5.25 – 5.4 seconds the Upshift is considered i.e. 1^{st} gear → 2^{nd} gear, where the engine powers the vehicle.

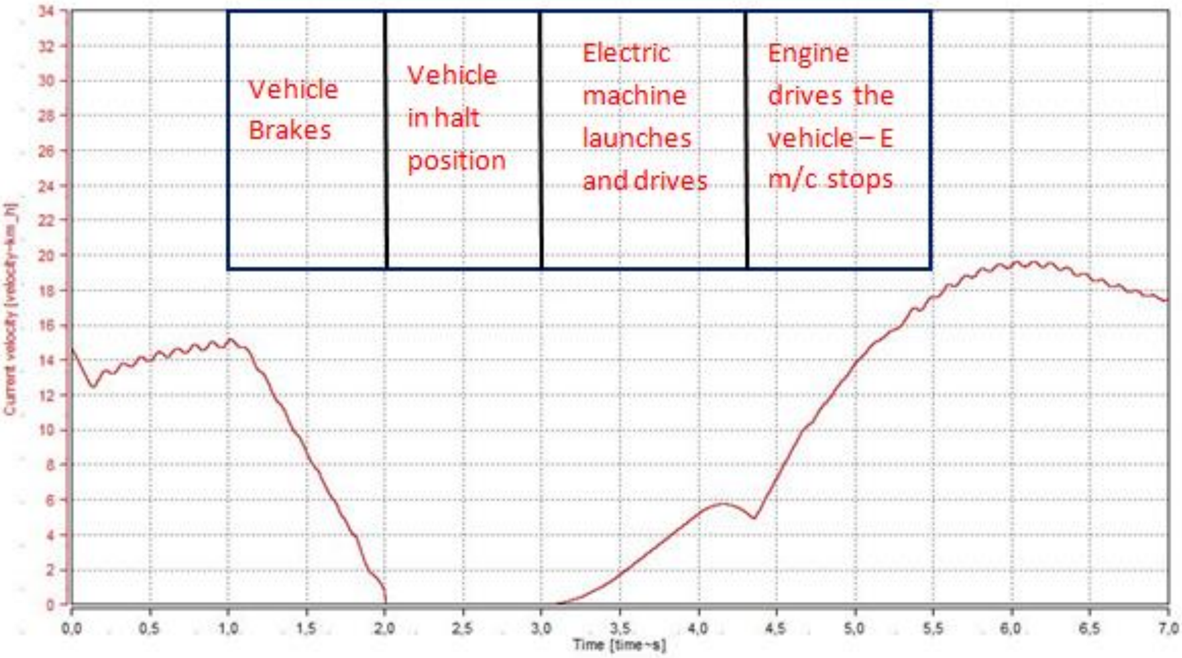

Figure 48: Actual velocity

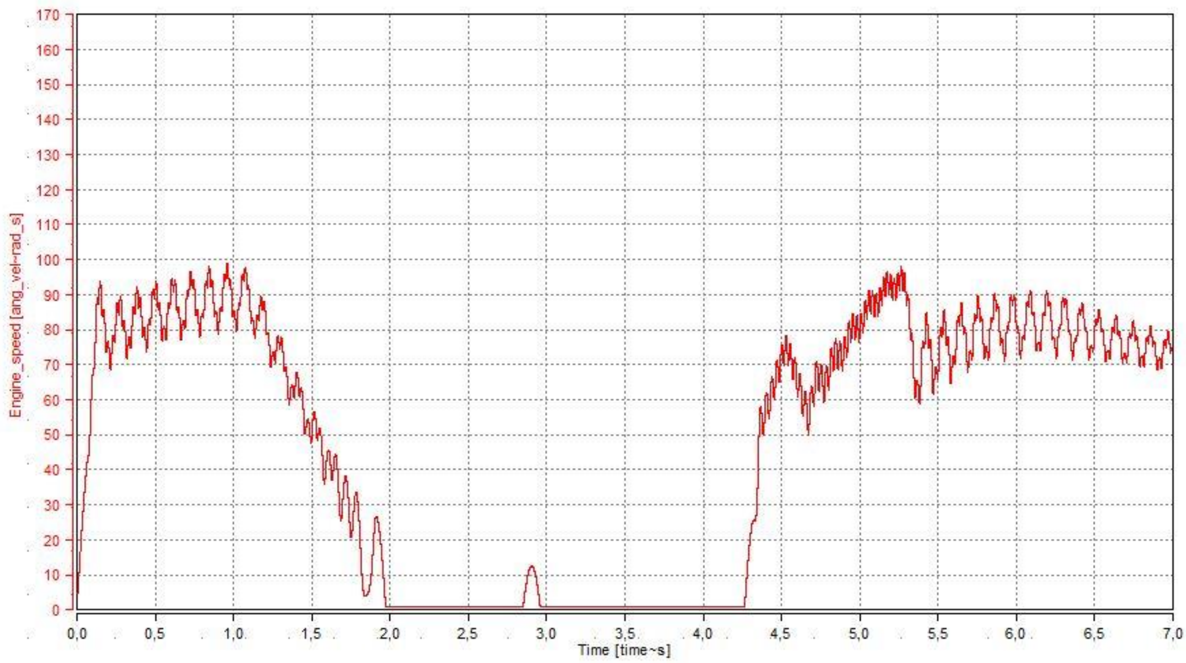

Figure 49: Engine speed

Figure 49, shown above represents the Engine speed, however as mentioned earlier, the reason engine speed starting from 0 (whereas the acutal velocity 15km/h), due to predefined velocity within the vehicle model. At 4.5 seconds, we see a sudden drop in Engine speed, which is due to complete disconnection of electric machine and connection of Engine. This is beacuse of lack of the engine power, for the desired profile and this sudden difference in speed is where jerk is felt within the system. Furthermore, we even see another drop at 5.25 seconds, which is due to gear shift within the transmission.

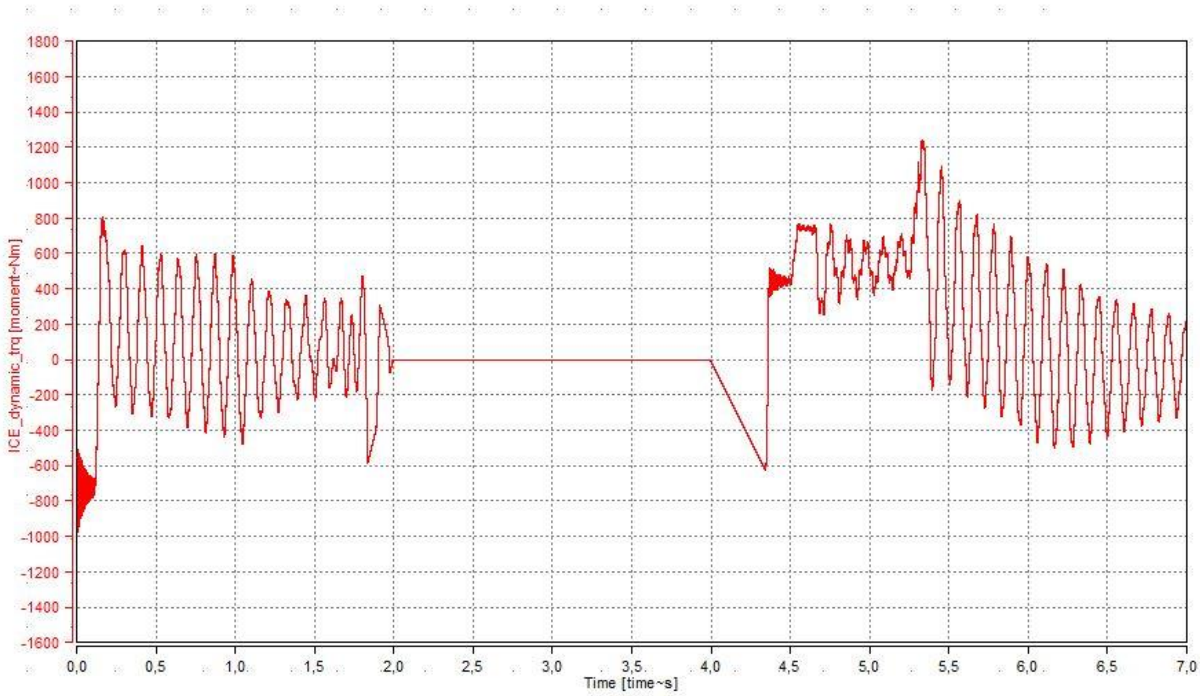

Figure 50: Engine Instantaneous dynamic torque

Figure 50, details Engine instantaneous dynamic torque, where we see the oscillation in the torque during the coupling phase and another during shifting phase. Around 4.3 – 4.6 seconds we see some transient behaviour that is due to sudden torque change, which influences low frequency vibration within the system.

Figure 51 below, showcases the ICE Clutch release, where decoupling (1.5 – 2.0 secs) and coupling (4.0 – 5.0 secs) of the Engine can be figured out.

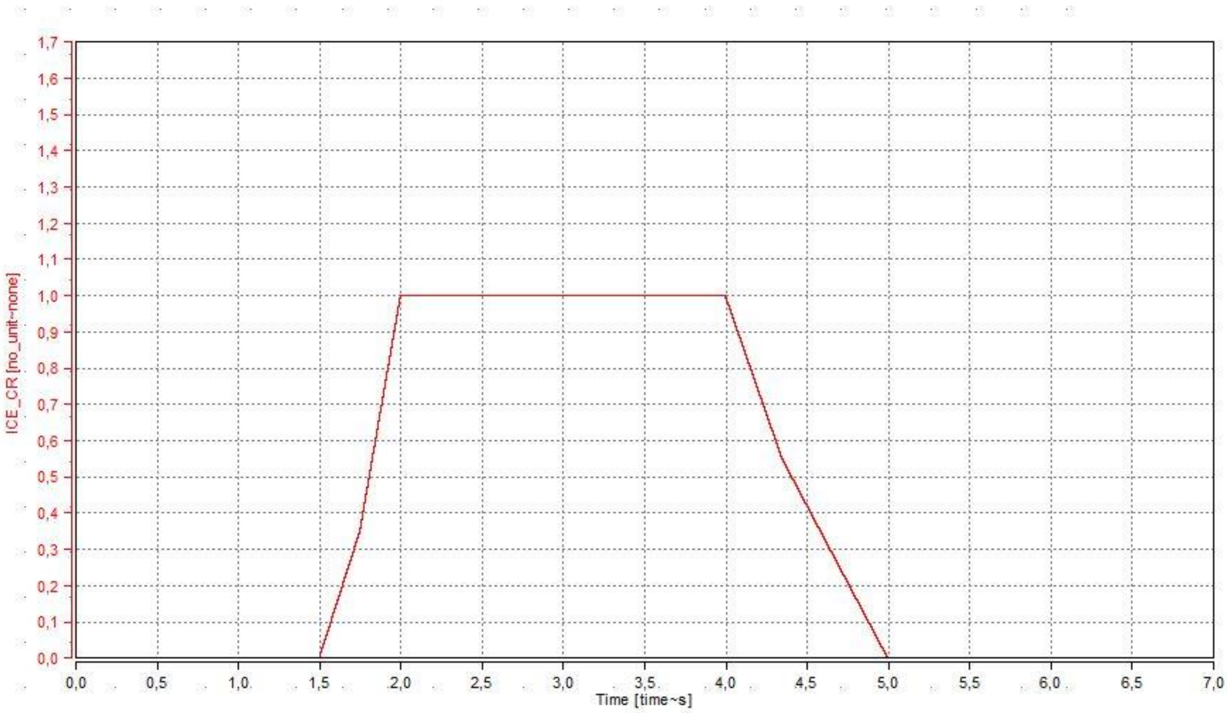

Figure 51: ICE Clutch release

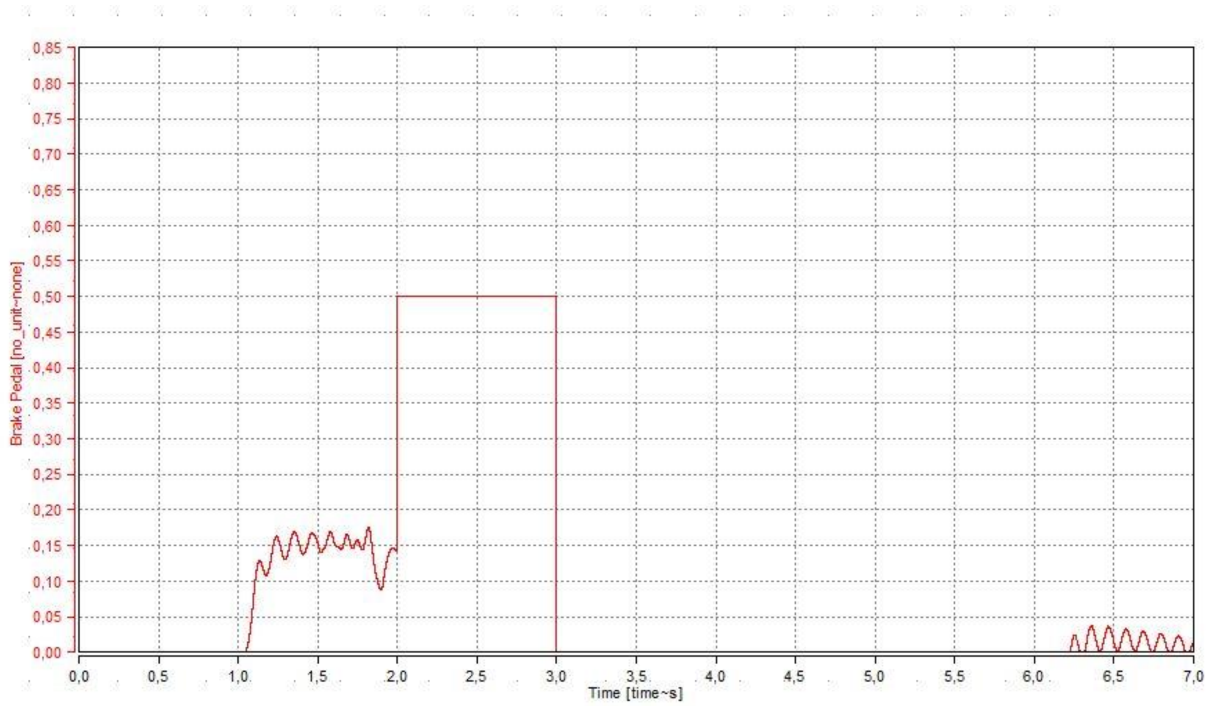

Figure 52: Driver Brake padel

Figure 52, represents Driver controlled brake pedal for the vehicle.

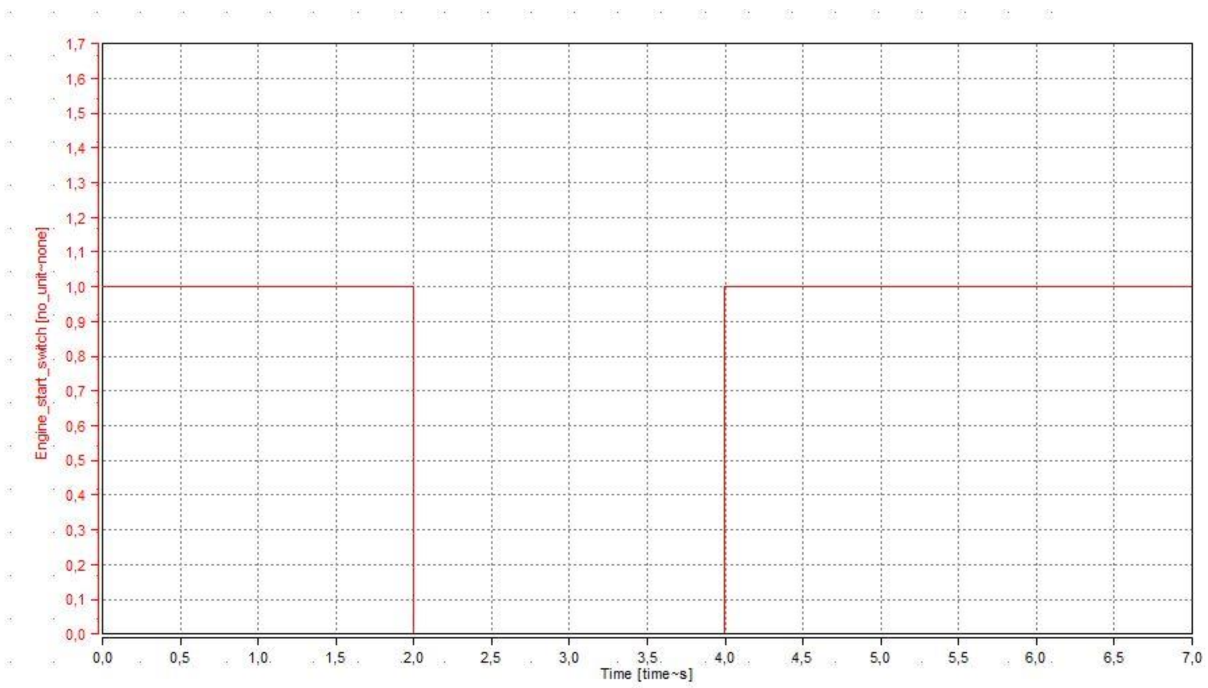

Figure 53: Engine start switch (0 – deactive, 1- active)

Figure 53 shows the Engine activation and de-activation signal.

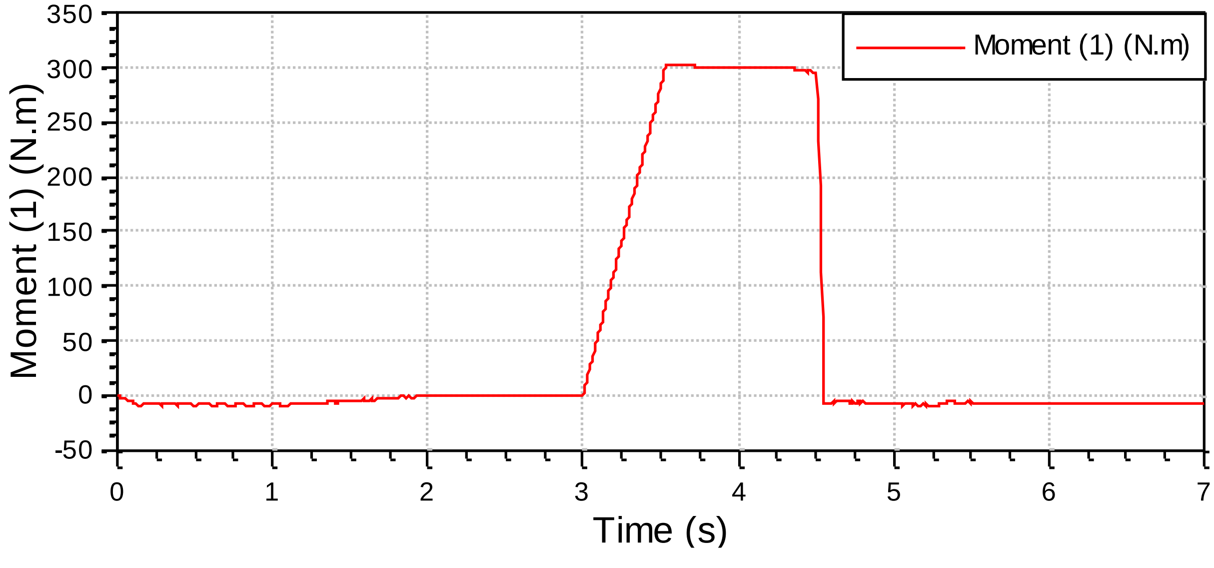

Figure 54: Electric machine torque

As mentioned earlier, the torque control strategy is defined for the e-machine. Figure 54 represents the pre-described e-machine torque input given to drive the vehicle i.e. after standstill of vehicle phase.

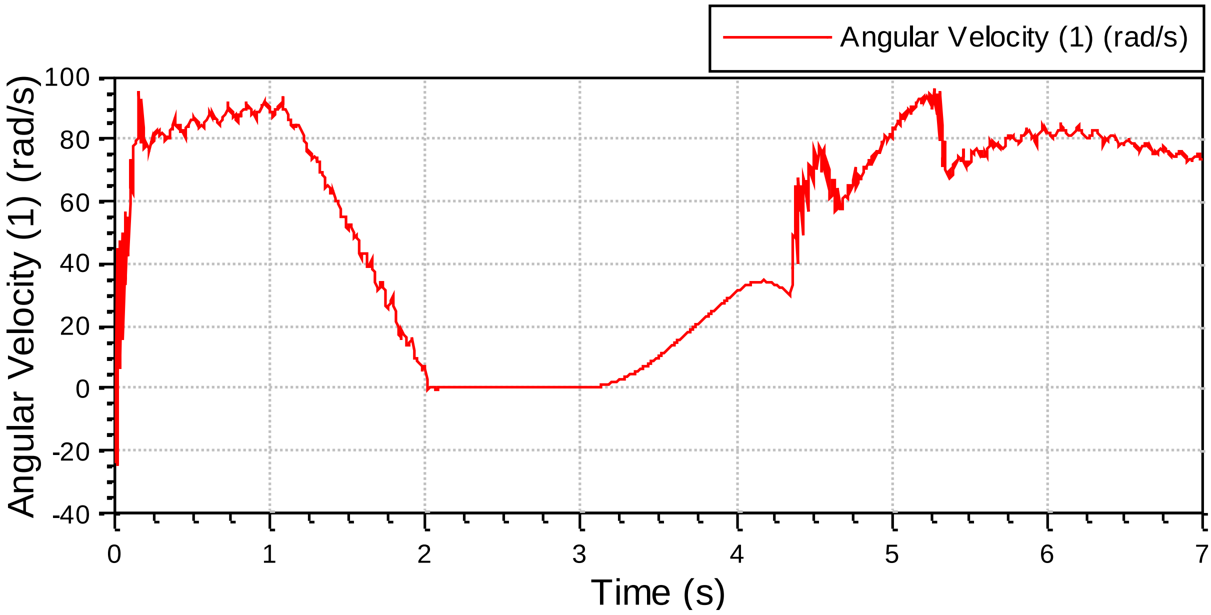

Figure 55: Angular velocity for transmission input shaft

Figure 55, represents the angular velocity of transmission input shaft, where we see the overall dynamic behaviour acting at different phase within the scenario. These non-stationary random oscillations are connected directly from the engine side, transmission side (PGS and clutch component) and from the dynamic effects of the electric machine. In-between 5.2 – 5.5seconds we again see torsional vibration, which is due to change in gear ratio. This abrupt change causes a torque reversal and leads to gear separation. The impact cause high frequency vibrations which transmit through the bearings and casing structure further then to ear as noise. The noise is likely to because of driveline clunk, which is heard in the final drive.

However, deeper investigation can further be carried out for each phase and its effect on each component can be evaluated. This oscillation at transient conditions will not only damage the connected components, but also affects the human sensitivity due to its frequent start/stop condition.

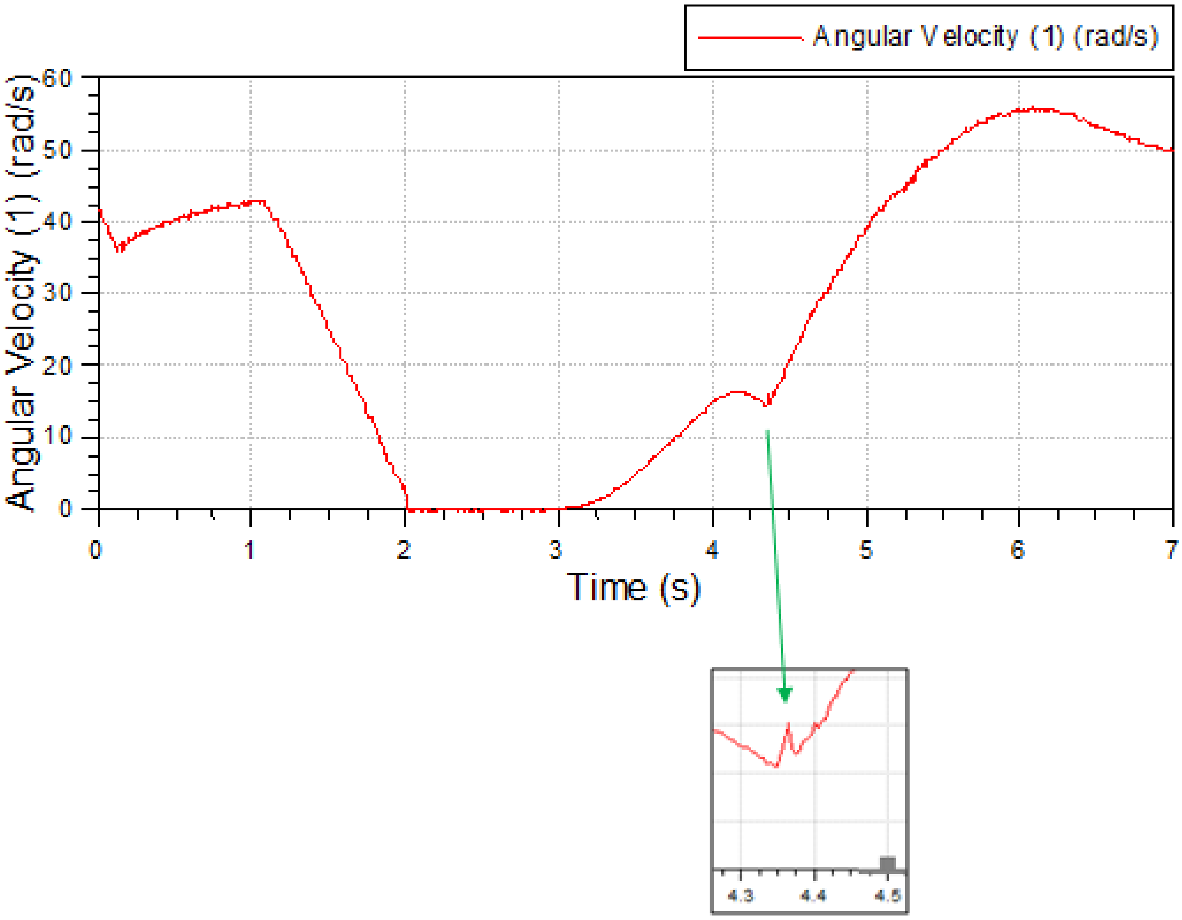

Figure 56: Angular velocity for transmission output shaft

Figure 56 represents the angular velocity of transmission output shaft, where a little disturbance is captured during the phase when the engine is coupled and electric machine is decoupled. This transient effect can introduce higher impact to the gear system and can even damage the coupling components (such as clutch), however to overcome this effects, torsional damper modelling is required.

4.5 Assessment conclusion

The work presented in this assessment showcases the research and analysis carried for stop and go scenario. In accordance with different phase operations following with predefined driving manoeuvre and its control via a driver model. As like other two assessments this model puts a special focus on the methodology of combining and coupling the different modelling domains within a comprehensive simulation framework and outlines results for vehicle oscillation. The model simulates the dynamic approach of diesel hybrid powertrain (which is rare, due to limited availability), where the influence of non-stationary oscillation are much higher when compared to other powertrains, due to its high torque demand.

The model can further be use to study the behaviour for longer time cycles, where more frequent stop and go scenario can be analyzed. This work can even be use for deeper investigation on single component and further analyze its effect for each phase, such as for instance driveline clunk i.e. from changing the damping to the lowest mode, the amplitude of the transmitted engine harmonics and the gear lash displacement. Evaluation based on vibration severity (VDV) of human body for complete cycle can be recommended for further investigation too. However, results shown above can further be verified with respect to measured data, in order to damped these effect and implement the control strategy for better drivability.

5 CONCLUSION & FUTURE WORK

Effective design of Hybrid/Electric vehicles is a challenge for involving multidisciplinary and specialized fields. And as mentioned earlier, GRESIMO project focuses on these interdisciplinary aspects within three disciplines which are NVH, Hybridization/Electrfication and Lightweight design. The research work presented here emphasis on this multidisciplinary field i.e. NVH and Hybridization/Electrification.

The main objective of this thesis is to elaborate the modelling approaches and methods for electro-hybrid powertrains and their incorporation in simulation software. The introduction of new methodology of combining all the domains (Engine, Transmission, Vehicle) for constructing one dynamic vehicle model has helped to simulate thermodynamic, strength, stiffness, dynamic and NVH responses of the drivetrain. Furthermore it has even helped to develop models/methods, which were appropriate for different types of hybrid applications, for optimization tasks and for use in simulation of transient operating conditions. However, all the HEV models presented have raised new noise and vibration quality related issues (e.g. harmonic issues between ICE and electric motor), which were not previously evident within the conventional ICE vehicles.

The discomfort produced by noise and vibration from these three assessment models can further be used to analyze and scale it's severity for the human body (by calculating VDV for all the non-stationary and transient effects). The reason for this investigation is because, as noise and vibration pollution tends to increase with increasing duration, which causes damage not only to the vehicle, but most importantly on human body too. Moreover, these effects can further be verified with the measured data and the oscillation behaviour can be reduced for betterment of technology.

Different opportunities can be identified for further research and development, as results of the methods developed and trailed during this work. Additionally, many more questions can be raised for deeper dive investigation on the issues mentioned within this report. Some important potential work which can be identified includes:

To design and reduce the mentioned noise and vibration, findings can be done on different damping concepts such as Dual mass flywheel (DMF), Centrifugal pendulum vibration absorber (CPVA) for torsional vibration from transmission and driveline, Internal crankshaft damper (ICD) for Torque ripple, and Balancershaft damper (BSD) for Engine related issues and furthermore materials for the components that can affect the response of different modes. These concepts can be introduced within the simulation environment and further can clarify the isolation of noise and vibration effects.

The proposal to use of control strategy that can focus on driver input and other important component of HEV model, rather than from fixed driving cycles for more dynamic approach. This approach is more cost effective and provides better robustness to the system, with the limitation to the increasing complexity.

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