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TNO report

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Final report of the Research Program (VTP1) on an Emissions and CO2 Test Procedure for Heavy Duty Hybrids (HDH)

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Summary

This report is the final report of the work by the Universities of Technology in Chalmers, Graz and Vienna and research institute TNO performed within the research program on an emissions and CO_2 test procedure for Heavy Duty Hybrids (HDH). This report specifically refers to Validation Test Program 1 (VTP1). The work is performed according to specific contract SI2.631381, titled "Developing the Methodology for Certifying Heavy-Duty Hybrids based on HILS", and sponsored by the European Commission. It also includes the report on the first work package within VTP1, which was sponsored by OICA.

Reduction of pollutants and greenhouse gas emissions as well as increased fuel efficiency are becoming even more important in the view of increased local pollution in some urban spots, global warming and higher fuel costs. Hybrid vehicles are therefore becoming more important not only for light-duty cars but also for heavy-duty vehicles such as city buses, distribution or refusal trucks. A test methodology using Hardware-In-the-Loop technology is applied in Japan to determine both CO_2 and pollutant emissions of HD hybrid vehicles. To support the Commission Services on the different activities of the emission legislation for heavy-duty hybrid and electric vehicles and especially in connection to the work of the UNECE GRPE informal group on Heavy-Duty Hybrid vehicles (HDH), the work described in this report covers specifically the research on following topics:

- Adaptation of the Japanese HILS simulator for serial hybrid
- Adaptation of the GTR-HILS simulator for parallel hybrid
- Report on HILS test procedure and adaptations towards adoption in GTR

These activities are defined as follow-up on previous research projects with regard to the development of a (globally harmonized) methodology for certifying heavy-duty hybrids based on HILS. A number of reports from past and current projects are to be combined in order to get the bigger picture of research results in this field.

Currently, the pollutant emissions of Heavy-Duty vehicles are regulated through various regulations. For Europe, this specifically is ECE R49, which is based on Global Technical Regulation No.4. Although CO_2 is measured as part of the regulation, it is not included as a regulated component yet and this certainly requires some modifications when this would need to be included in future versions of GTR No.4. For the moment CO_2 emissions will be covered by local regulations. Harmonization of the different local regulations into a GTR would require further development of, and gaining experience with, all newly introduced methodologies and tools.

This report specifically discusses the technical work on and results for procedures to measure pollutant and CO_2 emissions by investigating:

1   Defined HILS model library, its verification and user manual

2   Proposed test procedure based on the Japanese HILS procedure and indication of required modifications and discussion points with regard to adaption for a Global Technical Regulation

3   Continued investigation into the test cycles and issues originating from use of a
- vehicle cycle (speed and normalized power profile) leading to more vehicle specific test conditions
- engine cycle (normalized speed and torque profile)

HILS model library
Based on the Japanese HILS models, discussions with OEMs and stakeholders, an extensive HILS library has been defined in Matlab/Simulink (a commonly used development/engineering product by The Mathworks). This library covers all current and near-future expected hardware components for conventional and hybrid vehicles. HILS models for all currently foreseeable vehicle topologies can be constructed from this library and used in simulation and HILS environment. Within this project, actual validation in the HILS environment as part of a complete test procedure including modeling, verification/validation towards a HDH vehicle has not been possible (defined for the follow-up project VTP2). Therefore, completion of the HILS component model library cannot be confirmed until sufficient proof of valid HILS method results is generated and model changes are no longer required.

Proposed test procedure based on Japanese HILS regulation
The HDH working group has decided the Japanese HILS procedure to be a good starting point for defining a globally harmonized test procedure for Heavy-Duty Hybrids. All sections of the Japanese Kokujikan No.281 procedure have been reviewed and commented in order to identify required adaptations for adoption in a GTR. The current GTR No.4 (engine based testing) specifically targets the definition and implementation of a testing procedure for criteria/pollutants emissions for engine families, whereas the Japanese HILS regulation describes a procedure for certification of an individual hybrid vehicle for fuel consumption (calculated at charge sustaining condition) followed by engine testing to quantify its emissions.

Test cycles and procedures
When a vehicle cycle is selected, only defining vehicle speed over time (in combination with predefined road load conditions) is used as base input for determining the emission of a vehicle. It was concluded that the results delivered would not always be comparable with conventional vehicles which are tested based on a pure engine cycle. For that purpose, a method was developed to adjust the power to be delivered during the run of a certain speed cycle equal to the power which would have to be delivered for a conventional engine at a test on the engine test bench. Road gradients have been chosen to represent additional and lower loads to adjust the actual power demand for the vehicle. Investigations on the exact determination of applied road gradients, affected also by the final HDH certification methods, are still on-going, but a vehicle speed cycle with defined power demand seems to be a suitable solution for the needs of future hybrid drivetrain emission classification.

Based on the work carried out, a large number of topics that currently remain unresolved are included in the Open Issue List. A number of these topics are investigated in Validation Test Program 2 (VTP2). Other items may still need to be discussed or added.

Contents

1   Introduction

This report is the final report of the work by the Universities of Technology in Chalmers, Graz and Vienna and research institute TNO performed within the research program (Validation Test Program 1 – VTP1) on an emissions and CO_2 test procedure for Heavy Duty Hybrids (HDH). The work is performed according to Specific contract SI2.631381.

The project team structure and the relations with the contractors outside the scope of the contract are displayed in the figure below.
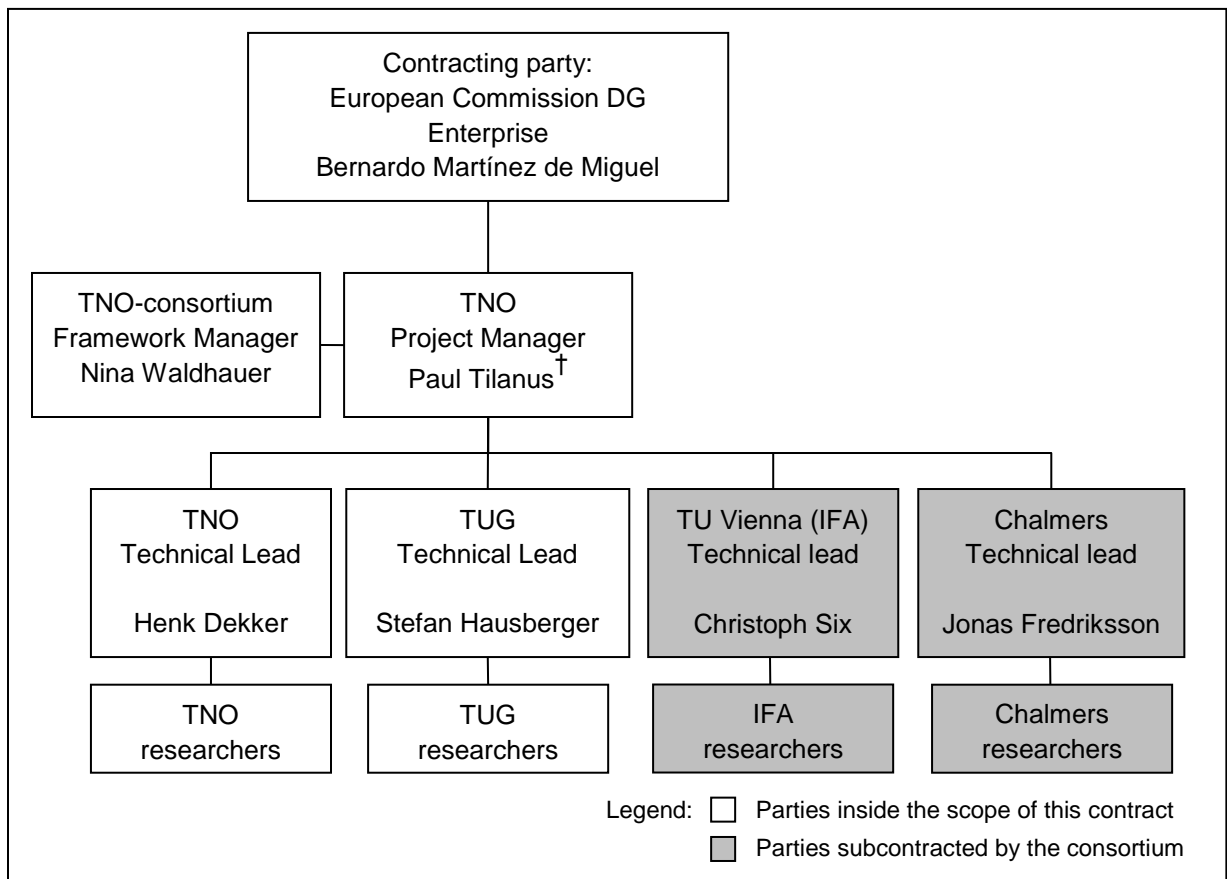
Figure 1: Project team structure ([†]Paul Tilanus replaced Martijn van Ras on 15-06-2013)

The overall HDH research program is executed by the Institute for Powertrains and Automotive Technology of the Vienna University of Technology, The Institute for Internal Combustion Engines and Thermodynamics of the Graz University of Technology and The Department of Signals and Systems of the Chalmers University of Technology and the department for Sustainable Transport & Logistics and Powertrains of TNO in The Netherlands. The project is sponsored by the European Commission.

The project was structured in three work packages plus an optional one:

Work package 1: Joint review, critical analysis and coordination.

Work package 2: Adaption of SILS for parallel HDH. This corresponds to Task 2 of the UNECE HDH informal working group.

Work package 3: Report on test procedure and adaptations. This corresponds to Task 3 of the UNECE HDH informal working group.

Option 1: Ad-hoc support.

Responsibilities for Work packages within the contract are displayed in the table below:

Table 1: WP overview according contract

WP	Description	Responsible party
1	*Project joint review, critical analysis and coordination*	
1.1	Framework and project management	TNO
1.2	Joint review and critical analysis of tasks 1, 2 and 3 *in which task 1 consists out of the work carried out by IFA, TUG and Chalmers in a project sponsored by OICA.**	TNO
2	*Adaptation of the GTR-HILS simulator for parallel hybrid*	
2.1	Meeting with OEM's and stakeholders	CH / TUG / IFA
2.2	Set up a data bus system in the model to allow various combinations of engines, gear boxes and storage systems.	CH / TUG / IFA / TNO
2.3	Adapt the software to simulate a parallel HDH.	CH / TUG / IFA
2.4	Simulation run and validation of basic functions, including the functions from task 1.	CH / TUG / IFA
3	*Report on test procedure and adaptations*	
3.1	Report on test procedure and user manual for software	CH / TUG / IFA / TNO
3.2	Provide the interface system for real ECU's	IFA / TUG
3.3	Adaptations and improvements on the methods for component testing, test-cycle definition and simulation method according to demands of industry and Commission.	CH / TUG / IFA / TNO

* Task 1 of the UNECE HDH informal working group, "Adaptation of the Japanese HILS Simulator for serial hybrid"*, is not part of the contract. This work is carried out by IFA, TUG and Chalmers in a project sponsored by OICA.*

In this final report, focus is given to technical investigation of issues and evaluation of results. WP1 from this project is therefore not explicitly included in this report directly, though indirectly covered in the other WPs. The results of Task 1 of the UNECE HDH informal working group, though not part of this project, are however, on request of the EC and OICA, included in this final report.

1.1   Reading guideline

This report is structured according to the project tasks as defined in the table on the previous page indicating the high level activities for Validation Test Program 1 (VTP1). Since the relations between the described issues mentioned in the different sections of the report are rather complex and not always presented in the most logical or convenient order, a reading guideline is established to provide the reader a suggestion for easier reading of the related topics:

- Section 1.2 for an introduction to type approval for Heavy-Duty hybrid vehicles
- Section 1.3 for an overview on the Japanese HILS test procedure
- Section 4.3.1 about test cycle related observations for HDH vehicles
- Sections 4.1, 4.2 and 4.3.2 on adjustments to Kokujikan No.281 based HILS methodology for adoption in a GTR
- Sections 2 and 3 for more background on the HILS library development and application
- Section 5 for an Open Issue List (with indication to specific sections of the report)
- Appendix A and B for detailed descriptions of component models in the GTR-HILS library

1.2   Type approval of HDH overview

In the current type approval process for heavy-duty vehicles only the combustion engine is certified for pollutant emissions. Engine testing may be sufficient for conventional vehicles, but is less representative for heavy-duty hybrid operation since the hybrid powertrain influences the way the engine is loaded during operation. Thus, the aim of the HDH-IG is to develop a type approval method that is more representative of HDH operation.

There are several potential ways to test the pollutant emissions of a HDH vehicle [2], but the mission of the research consortium in the HDH-IWG was to investigate the existing Japanese test procedure based on a HILS (Hardware-in-the-Loop Simulation) approach as basis for a future global regulation.

Following the existing Japanese HILS method the approach planned was to develop a procedure starting with a vehicle speed cycle as input. By using a simulation model (consisting of sub-models for the driving resistances, the different powertrain components and the driver) and the real vehicle control units as hardware the vehicle speed cycle should be transformed into a specific load cycle for the combustion engine.

Due to two independently operated energy converters the load cycle of the combustion engine of a HDH depends highly on the control strategy. For conventional vehicles, which need to provide the required propulsion power only through the combustion engine, the load cycle of the combustion engine is directly linked to the required propulsion power. By including the real vehicle control units in the transformation process, the distribution of the required propulsion power is handled by the actual operating strategy like in the real vehicle.

The specific engine cycle generated out of the vehicle speed cycle by usage of the HILS system is then used for testing the pollutant emissions on the engine test bench in the same way as it is done for a conventional engine.

The final, worldwide established HILS test method should be - as far as reasonable - in agreement with the test procedure for conventional engines specified in GTR No.4. Therefore, the WHVC (Worldwide Harmonized Vehicle Cycle), a representative vehicle speed cycle used as intermediate step in the generation of the WHTC (Worldwide Harmonized Transient Cycle), was chosen as promising basis of the test procedure [8]. A vehicle speed cycle is a very stable reference basis and does not change much with evolution or new development of drivetrain technologies.

The following chapter 1.3 gives a more detailed insight to the existing Japanese HILS test method.

1.3   Japanese HDH HILS test procedure

This chapter describes the Japanese HILS (Hardware-in-the-Loop Simulation) method defined in Kokujikan No.281 [1] and gives an overview of the different steps of the certification process.

The main goal of the HILS procedure is to transfer a vehicle speed cycle into an engine test cycle that is representative for the specific hybrid control strategy. With HILS, it is possible to simulate a hybrid vehicle driving a transient vehicle speed cycle. During this simulation, engine operation is recorded, thus creating a vehicle specific engine cycle. This engine cycle can then be used to test the engine's emissions on a conventional engine dynamometer.
The operation of the engine in a hybrid vehicle is highly dependent on the manufacturers' proprietary hybrid control strategies. These strategies are encased in the hybrid electronic control unit (hybrid ECU). It is undesirable to disclose the proprietary software inside the hybrid ECU. To be able to include these control strategies in the simulation loop, the hybrid ECU is kept as hardware and is connected to the simulation, which is run in real-time. This process is called 'hardware in the loop simulation'.

By the means of the simulation model (consisting of sub-models for the driving resistances, the different powertrain components and the driver) corresponding to the real vehicle and the real vehicle control units as hardware the vehicle speed cycle is transformed into a specific load cycle for the combustion engine. Operating the HILS system reduces the effort for varying the vehicle parameters as well as the starting conditions compared to testing of the real vehicle. Figure 1.1 shows the basic approach of the Japanese HILS method.
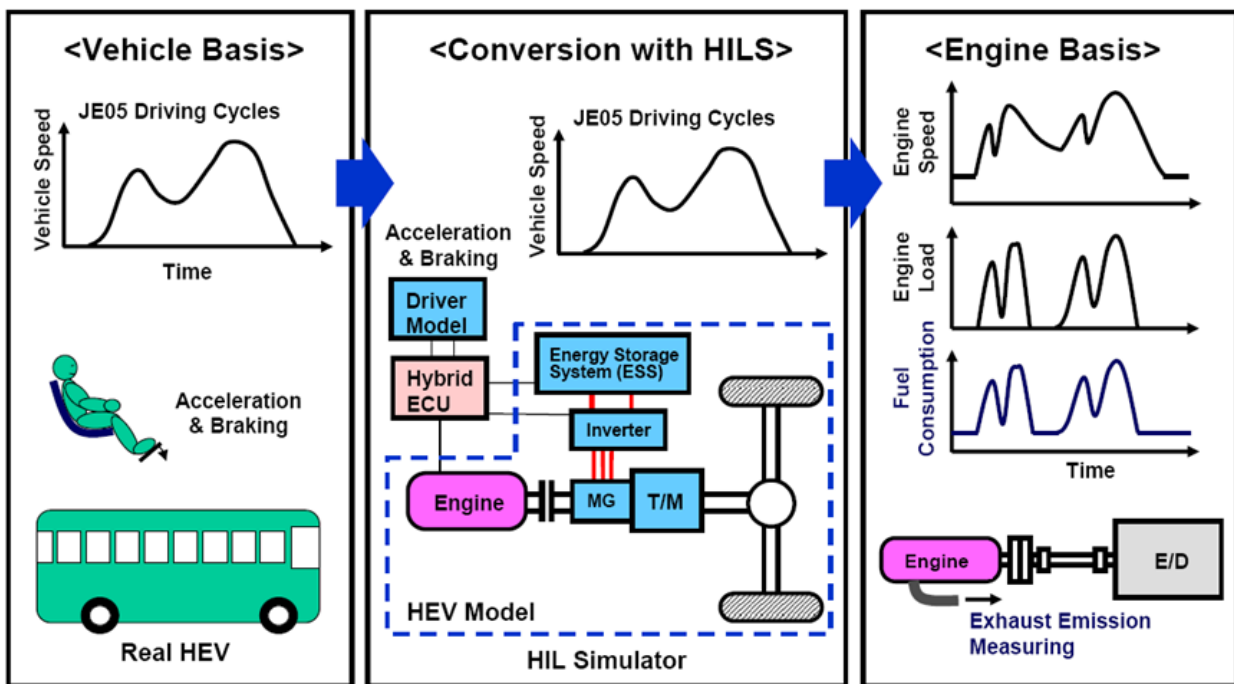
Figure 1.1:   Basic principle of the Japanese HILS method [10]

In general, the HILS method in use in Japan can be divided into several steps, which are structured in the flowchart below (Figure 1.2) and explained as follows:

1. Investigation of the hybrid drivetrain topology of the vehicle to be tested.

2. Creation of the vehicle model according to the topology defined in step 1.

3. In order to test the hardware that runs the vehicle model, a model verification with generic component data and a generic control strategy (predefined operation strategy) is done as a pre-check.

4. Before the certification method can be performed, conformity between real vehicle and simulation model has to be ensured. Therefore measured data of real vehicle operation (either from chassis dynamometer or from a pre-transmission powertrain test bench) is compared to simulation results. If the output from the simulation meets the defined tolerances, the HILS model can be used for the certification process.
    If the same powertrain topology has been certified before, no new model verification is necessary and step 5 follows. If a new model verification is necessary, the component parameters of the verification vehicle are put into the simulation model (determined according to the procedures in step 5). Then a HIL simulation run is performed and the simulation results are compared with measured data.

5. Component specific data for engine, electric machines and energy storage according to the defined test procedures including vehicle mass, inertias, transmissions and gear ratios are generated and input into the HILS model.

6. A simulation run of the vehicle model is performed with the defined vehicle speed cycle as input.

7. It is checked, if the simulation properly follows reference speed (defined vehicle speed cycle input). If not, the parameters of the driver model have to be adjusted and the HIL-simulation has to be repeated.

8. It is checked, if the change in state of charge of the electric storage (ΔSOC) between cycle start and end is within the allowable limit. If not, the initial state of charge has to be adjusted and the HIL-simulation has to be repeated.

9. The engine operation points recorded during successful HIL simulation are used to calculate fuel consumption from a stationary fuel consumption map and to measure emissions on an engine test bench like it is done for conventional vehicles.
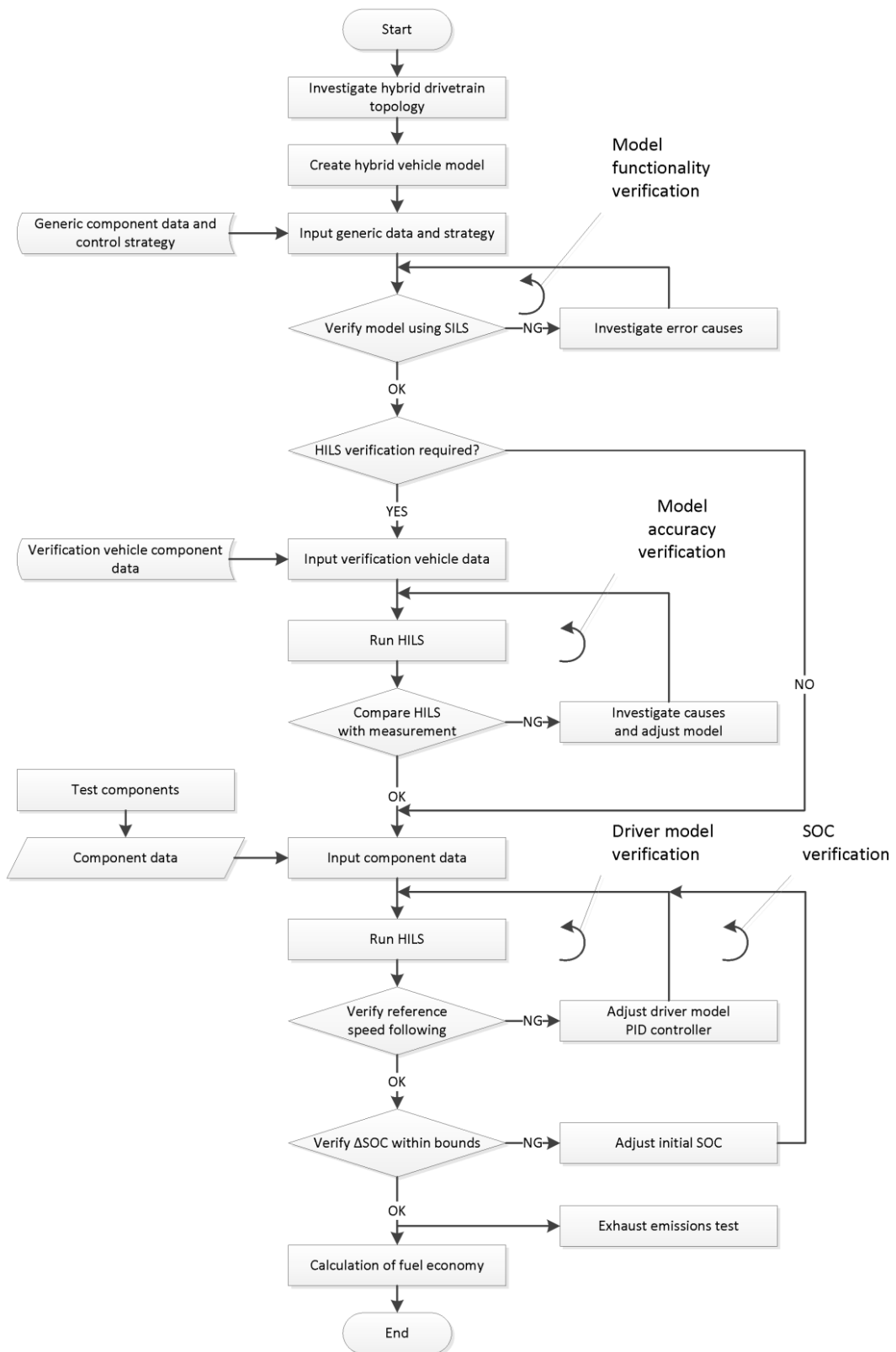
Figure 1.2:   Flow chart of certification process with the Japanese HILS method   [11]

2 Task 1 - Adaptation of the Japanese HILS Simulator for serial hybrid

Task 1, as defined by the UNECE HDH informal working group, consists of the following activities:

1.1 Set up a serial HDH in the Simulator with the ECU as software in the loop as basis for further programming and software development

1.2 Add a software tool ("driver model") which allows to run the Simulator with test cycles consisting of power and rpm at the wheel hub and at the powertrain shaft as basis for the "GTR-HILS"

1.3 Extend the Simulator with a library for non-electric components

1.4 Meetings with OEM's and stakeholders to discuss relevant components to be included in a first version of the "GTR-HILS" models as basis for tasks 1.5 and 1.6

1.5 Extend the "GTR-HILS" Simulator with a library for powertrain components not yet included in the Japanese HILS model (e.g. planetary gear box and power split, others if relevant and possible)

1.6 Extend the "GTR-HILS" Simulator with thermal models for exhaust gas aftertreatment components, coolant, lube oil, battery and electric motor where relevant according to task 1.4

1.7 Simulation runs and validation of basic functions

The results of these activities are reported in the subsections of this chapter.

The reporting in this chapter is under complete responsibility of Chalmers University, TU Vienna and TU Graz since this is the report of the earlier OICA sponsored project.

2.1 Task 1.1 - Serial HDH with ECU as SIL

Since it turned out later in the project that there is a need to restructure the available Japanese vehicle models, and the restructuring was confirmed within the HDH group the work processed with the Japanese models will, since it is mostly no longer relevant for a GTR adoption, only be roughly outlined in this report. Only the work on the Japanese models in task 1 which affects the new models for the GTR will be reported in detail. A detailed description of the new GTR models which are also able to deal with the requirements of task 1 (except the driver model for power cycles – see section 2.2) will be bundled in section 3.2.

The requirement of task 1.1 was to extend the existing version of the Japanese HILS model with a simple module, which simulates the ECU of a serial hybrid. This module should allow running the software without a hardware ECU. The functions include the monitoring of the battery SOC and a resulting on/off function of the ICE for generating electricity at three freely adjustable load points. The aim was to run the serial hybrid model in any vehicle velocity cycle inputted. Since the Japanese model did not contain a driver model this had to be developed in addition to the ECU model. Figure 2.1 gives an overview of the serial hybrid model based on the Japanese structure.
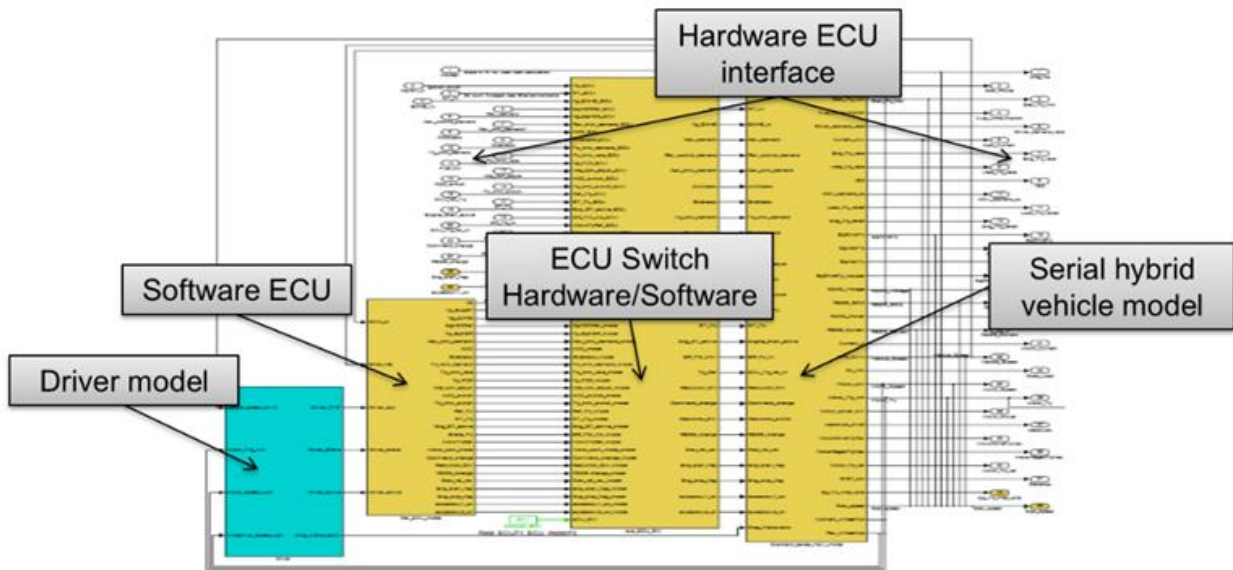
Figure 2.1:   Serial hybrid model base on the Japanese model structure

The ECU model was designed in a way that the vehicle could be run either by providing time based signals for the accelerator and brake pedal or by means of a driver model which was able to actuate accelerator and brake pedal in order to meet the given velocity cycle. The actual SOC of the energy storage system was used to trigger the operation of the ICE. Below a specified level the ICE was turned on to produce energy until a certain SOC level was reached again. The driver model was designed using a common known PI controller whose output is dependent of the deviation between actual and desired vehicle speed. Partially it could be transferred into the new model structure where it is described in Appendix A - Component models. A closer description of the model itself is not foreseen here since it was changed for a GTR adoption extensively (see section 3.2). Nevertheless simulation test runs were performed and results are presented in section 2.7 for completeness.

2.2   Task 1.2 - Driver model tool

In phase one of the project the replacement of a vehicle speed cycle as input by a WHTC-based torque + rpm cycle at the wheel hubs or alternatively at the shaft of the HDH powertrain was recommended to provide similar load conditions for hybrid propulsion systems and for conventional ICE's. To handle torque and rpm control instead of vehicle speed control, an alternative driver model had to be elaborated. The idea itself seemed to be smart in order to be able to compare conventional and hybrid vehicles in terms of emissions but during the implementation a lot of challenges were faced.

Section 4.3.1 will report in detail, representatives are enumerated below:

- How to denormalize a WHTC for a hybrid powertrain
- How to deal with stipulated gear shift patterns included in the WHTC
- How to deal with different powertrain layouts (pre-, post-transmission,…)
- Since the WHTC is a pure engine cycle, how can rotational speeds be defined if post-transmission powertrains are tested
- ….

Nevertheless the operation principles of a driver model able to run power test cycles will be explained in this section.

If you want to track a desired vehicle speed a common driver model can be used to actuate accelerator and brake pedal. The rotational speed of the engine (ω) which corresponds to the vehicle speed is basically a function of generated powertrain torque (T_{PP}) and acting load torque derived from the actual road load (T_{RL}). The inertia (θ) represents the vehicle mass as well as the inertia from the rotating sections of the vehicle.

$$\omega = \int \frac{T_{PP} - T_{RL}}{\theta}$$

If you neglect deviations of the rotating inertia caused by different gear ratios, the inertia can be considered as a constant value which means that the rotational speed is a pure function of resulting torque on the drivetrain. A conventional driver model is requesting a certain amount of torque to track the desired vehicle speed. Torque is your control parameter, speed is your system result.
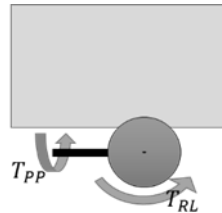
Figure 2.2:   schematic vehicle and acting loads

If you want to run a vehicle model following a desired power curve (propulsion power vs. time) one single control signal like the driver torque request will not be sufficient. To track the desired power you will have to track a specific torque and a specific speed. This indicates that you will need one controller for the torque and one for the speed tracking. Controlling the torque is more or less similar to a conventional driver model. The torque controller is actuating the accelerator pedal to adjust the desired driving torque (T_{PP}). To control the resulting speed you need the second controller, which controls the load torque on the system (T_{RL}) in order to track the desired speed. The chassis model, which is usually representing the road load for the system, is therefore replaced by a controller which controls the road load for this configuration. Both controllers are designed as common known PID controllers. The fact that they are influencing each other makes it harder to tune them accurately but at least for the serial hybrid model based on the Japanese structure the controller tuning could be handled well for both, the wheel hub referenced and the powertrain output shaft referenced driver. Figure 2.3 illustrates the principal structure containing demanded speed and torque as well as permitted tolerances, a basic PID controller and a watchdog system to detect deviations.
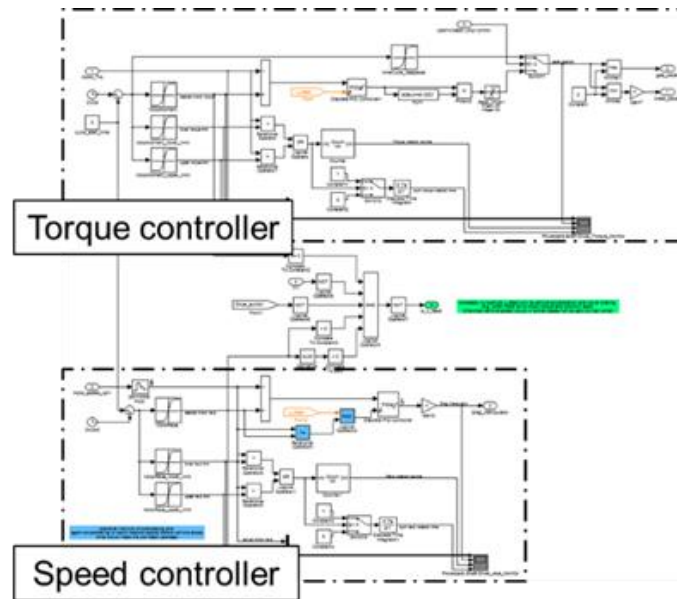
Figure 2.3:   schematic of driver model running power referenced input cycles

A successful run of the Japanese structure based SILS vehicle model with power cycles consisting of torque and speed, which were derived from the WHTC test cycle, could be performed but faced some serious problems other than the developed driver model. Section 4.3.1 will in detail report on these drive cycle related issues which make the usage of a driver model tracking torque and speed derived from the WHTC doubtful. For that reason this driver model was in a first step not transferred into the new model structure, but replaced by a model for a combined vehicle speed and road gradient cycle. In this cycle the road gradient is variable and thus ensures a good correlation between WHTC cycle work and the cycle work from the combined vehicle speed and road gradient cycle (see 4.3.1.2).

2.3   Task 1.3 - Non-electric components library

In this subsection, an analysis on what kind of non-electric hybrid systems/components that are available or can be foreseen in the future on the market will be presented, in order to identify which components that need to be modelled and included in the library.

A hybrid vehicle is a vehicle that has two or more ways to propel the vehicle. A hybrid system needs i.e. a secondary energy storage and a secondary energy converter that can propel the vehicle. In a hybrid electric vehicle, the conventional fuel tank and internal combustion engine is complemented with an electric energy storage system, like a battery or a super capacitor, and an electric machine to propel the vehicle.

There are several ways to store energy in a non-electric way, in a flywheel kinetic energy can be stored, in a spring potential energy can be stored, and potential energy can also be stored in an accumulator. From a vehicle point of view, storing energy in a spring is not such a good way since it requires a very large spring, but using a flywheel or an accumulator are technical viable solutions. On the market

today, two specific alternatives exist for storing non-electric energy for hybrid vehicles, namely hydraulic systems and flywheel systems. Therefore these two different ways are identified as the most likely to incorporate in a simulation based method. The operating principle for a pneumatic system is very similar to the operating principle of a hydraulic system, so introducing hydraulic systems also covers pneumatic systems.

The second important part of a hybrid system is the energy converter. This is the part that converts the stored energy into mechanical, kinetic energy for propelling the vehicle. It is, of course, preferable if the same component can be used for converting mechanical kinetic energy to energy that can be stored in the energy storage system, i.e. the reverse operation. For a flywheel system, either the kinetic energy is transferred directly via a mechanical transmission, or transferred via an electrical transmission. An electrical transmission requires conversion of kinetic energy to electric energy and back to kinetic energy, and is usually done using an electric generator/motor configuration. For hydraulic and pneumatic systems, the potential energy needs to be converted to kinetic energy; this is usually done using a hydraulic (pneumatic) motor. In reverse operation, the motor is called a hydraulic (pneumatic) pump.

The components in a hybrid vehicle can be arranged in several different ways. The ways the components are arranged are often referred to topology or vehicle topology. For hybrid electric vehicles three different topologies are usually used, series, parallel and split, see e.g. [4]. In the series powertrain topology, there is no mechanical connection between the combustion engine and the propelling wheels. The propulsion is made by the electric motor and the combustion engine is primarily used to assure that there is enough energy in the energy storage. In the parallel powertrain topology, both the combustion engine and the electric motor can propel the vehicle. The energy for propulsion can be "added" together before (pre-) or after (post-) the transmission, the gearbox. The split powertrain topology is a more or less a combination of the series topology and the post-transmission parallel topology. It is sometimes referred to as series-parallel powertrain topology. In Kokujikan No.281 [1], at the moment, only series and parallel powertrain topologies are considered.

2.3.1   *Flywheel system*
A flywheel hybrid system consists of, as mentioned earlier, a flywheel energy storage and a transmission, mechanical or electrical. For a flywheel system with a mechanical transmission, often a Continuous Variable Transmission (CVT) is used. Such a system can be categorized as either a series topology or a parallel topology, depending on how the flywheel system is controlled. An alternative approach is to connect the flywheel to an electric machine, the device then works as a "mechanical battery". The "mechanical battery" can than be used in a the same way as a normal battery in a hybrid electric vehicle. Flywheel systems are typically used in a series topology.

2.3.2   *Hydraulic/Pneumatic system.*
The most common non-electric hybrid vehicle combination is to use a hydraulic system. Numerous examples of this type of solution exist on the market today, for example Eaton's Hydraulic Launch Assist, Bosch Rexroth's HRB system, Parker's Runwise system and Poclain's ADDIDRIVE Assist. All of these systems can be categorized into a series or a parallel topology. There exist

also variants that can be categorized as a split hybrid topology. Pneumatic hybrid systems work in the same way and can solely be categorized in the same way as hydraulic hybrid powertrain systems.

2.3.3   *Summary*

Non-electric hybrid powertrain topologies (concepts) fit well into the same categories as for electric hybrid powertrains. Of the identified different powertrain concepts that are available or can be expected in the foreseeable future on the market, the following non-electric propulsion systems are considered interesting: CVT and flywheel, motor/generator and flywheel and pump/motor and accumulator (hydraulic and pneumatic). Models enabling the simulation of the identified non-electric hybrid powertrains have been developed. For details of all models, see Appendix A - Component models.

2.4   **Task 1.4 - Meetings with OEMs and stakeholders**

Insights, conclusions and demands from all OEM meetings during the entire validation test program 1, covering the work packages in Task 1 and Task 2 are bundled in section 3.1.

2.5   **Task 1.5 - Additional powertrain components library**

2.5.1   *Planetary gear set in old model structure*

According to the project time schedule task 1.5 started before the first OEM meeting could be held. During the offering process it was assumed that a planetary gearbox model will be needed and therefore modelling was started. To avoid several switches for each mounting condition in which the planetary gearbox could be operated (vehicles propeller shaft connected to the sun gear, the ring gear or the planetary gear carrier), in general three different models have been developed for each condition. The respective remaining two gears/shafts can be connected freely to any rotating machine or can be locked. Since the models have been developed from the white scratch different complex models are available. A very simple planetary gear set for each mounting condition, which does not consider efficiencies and inertias, is available in the library. Although there will be no demand if a gear set should be modeled accurately it can be used to check the principal function of a vehicle model with a planetary gear set. A more developed model considers inertias of all rotating sections but still does not take efficiencies into account. Modelling a gear set which considers inertias and efficiencies as well was started and one sub model where the propeller shaft is connected to the planetary gear carrier was already finished. All mentioned models are well validated with a different software from Gamma Technologies Inc. (GT-Suite). Due to the upcoming workload regarding the development of a HDH drive cycle, the need of restructuring the vehicle models for the GTR and since there was no demand yet for a planetary gear set the models have not been further developed and the manpower was used, according to the contract, to manage the resulting tight planning. The implementation of the new structured vehicle models for later GTR adoption made the developed models for planetary gear boxes incompatible for operation. An adjaptation represents a significant effort and was not performed yet.

2.5.2 *Components in new model structure*

A detailed description of additionally modeled components available in the model library can be found in Appendix A - Component models. Specific component test procedures for the new developed models, which were not considered in the original Japanese model, are not available yet nor required for all models. The enumeration below gives an overview.

* Mechanical connection (summation gearbox)
* Electric Auxiliary System
* Mechanical Auxiliary System
* DCDC converter
* Retarder
* Torque converter
* Battery (RC model)

2.6 Task 1.6 - OEM/stakeholder requested simulator extensions

In phase one of the project it was concluded, that HDHs will have to undergo a cold start test similar to conventional internal combustion engines in the GTR No. 4 [9]. The ECUs of HDHs will need plausible information on the temperature levels of all relevant components to select the correct running strategies. To provide reasonable temperature signals, relatively simple thermal models were developed and integrated into the HILS simulator.

2.6.1 *Engine coolant and lube oil*

As a first step an EURO V engine with SCR system was measured on the engine test bench at the Institute for Internal Combustion Engines and Thermodynamics at TU Graz to gather data for the temperature behaviour during warm up of the engine. The temperatures of the engine lube oil as well as temperatures of engine coolant at inlet and outlet of the inner cooling circuit were measured during three different warm up cycles (WHTC, ETC, stationary operation point).

In order to generate a simple model for the engine warm up and keep the effort to generate input data to a minimum the following method was used to calculate the oil and coolant temperatures during warm up by usage of lookup tables:

1. From the measured instantaneous fuel consumption and the net calorific value of the fuel the introduced energy to the combustion chamber per time unit (i.e. calorific power) is calculated.
2. From this introduced calorific power the mechanical power delivered by the engine is subtracted, giving the total heat loss power.
3. This total heat loss power is multiplied by a constant factor of 0.5, giving an approximation of heat loss power that is warming up engine oil and coolant (i.e. heat loss power to coolant and oil).
4. The heat loss power to coolant and oil is accumulated and the respective temperatures of engine oil and coolant are plotted as a function of this accumulated heat loss power to engine block.
5. These function plots look very similar for the different warm up cycles. Thus, a polynomial function that fits the three different warm up curves best is used to represent the warm up behaviour.
6. These polynomial functions, one for engine oil and one for coolant, are then implemented as lookup tables in the simulation model (Figure 2.6).

7   After reaching the normal operating temperature, the temperature is set to a
constant value (i.e. the cooling system starts controlling the temperature and
keeping it relatively constant).
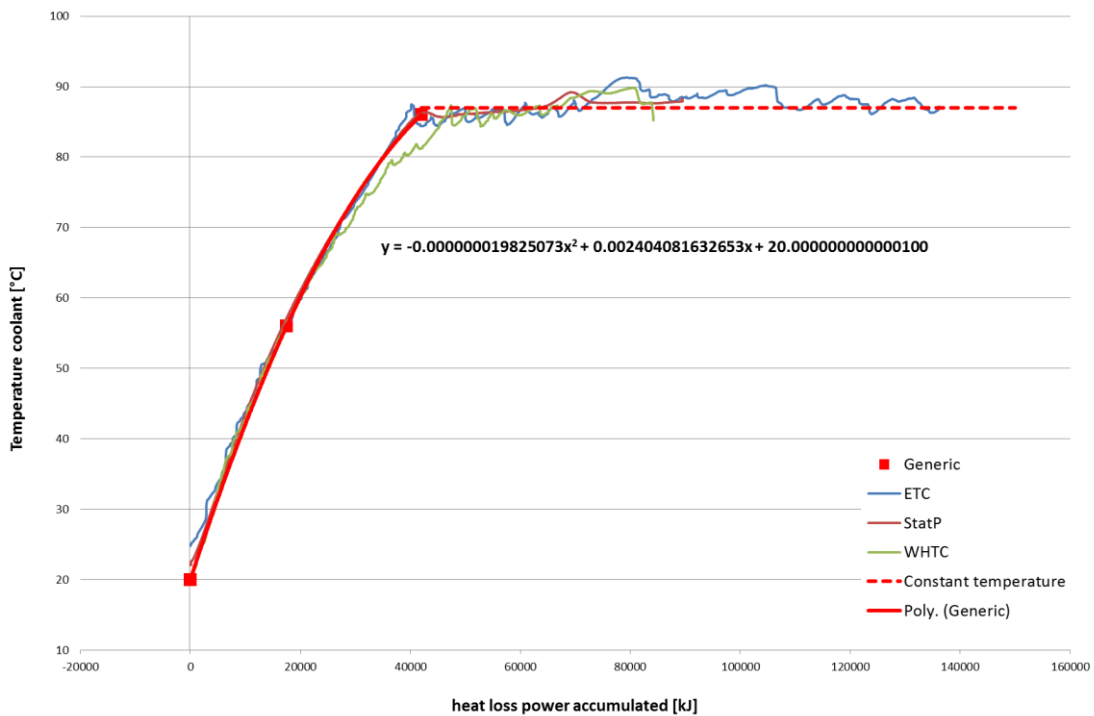
Figure 2.4:   Warm up behaviour of engine coolant
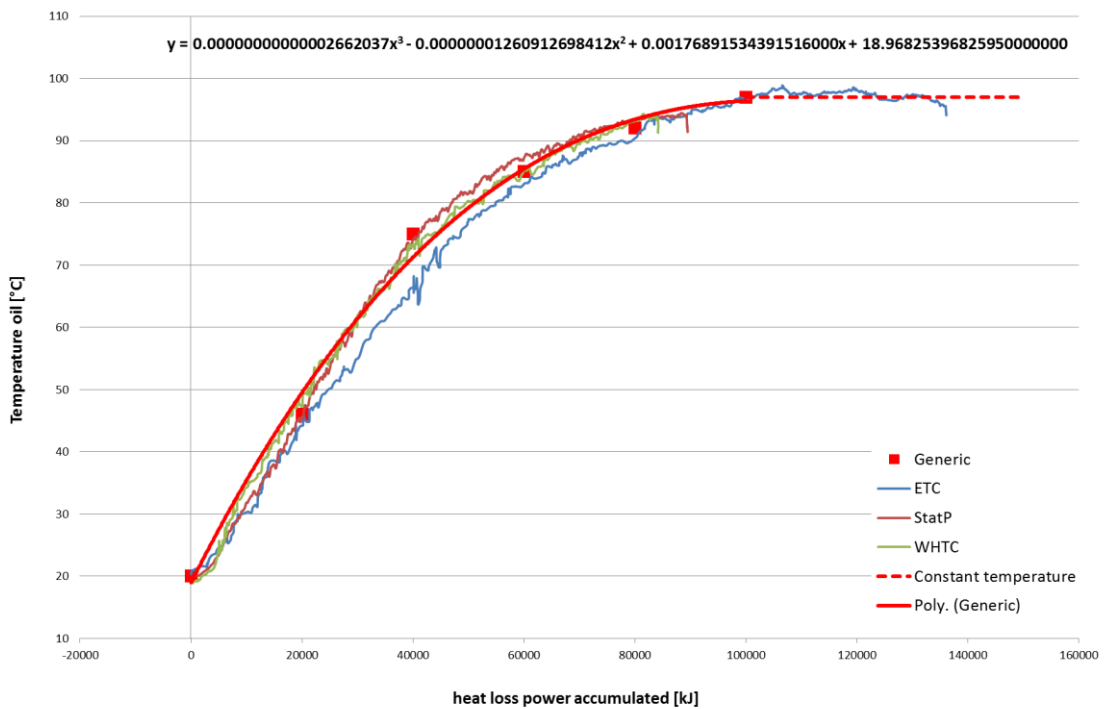
Figure 2.5:   Warm up behaviour of engine lubricant

The corresponding simulation model is shown in Figure 2.6**.**
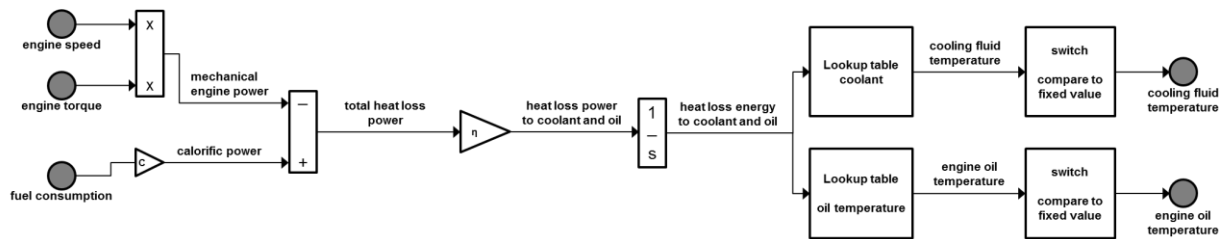
Figure 2.6:   Block scheme for engine coolant and oil thermal model

In order to generate generic lookup tables for the warm up behaviour the values of the accumulated heat loss to coolant and oil are normalized to engine capacity. The engine capacity is one input parameter in the engine parameter file and used to denormalize the engine warm up lookup tables for a specific engine.

If the generic warm up behaviour should prove not to be accurate enough for some engines the generic data could easily be replaced by OEMs. Therefore a few cold start measurements according to the described method and generation of a warm up lookup table for the specific engine out of the measured fuel consumption and temperature data are needed.

In addition also the parameter η (eta) can be used to adapt the generic warm up behaviour to one specific engine. As standard setting η (eta) should have the dimensionless value 0.5.

2.6.2 Exhaust gas aftertreatment systems
The simulation of the temperatures of the exhaust system is based on a zero-dimensional model of a series of heat capacities representing typical sections of the system, such as pipes, after-treatments or the manifold and the turbocharger. The simulation includes the convective heat transfer between the exhaust gas and the heat capacities as well as the convective heat transfer and radiation between the heat capacity and the environment. The heat input from exothermic reactions in the after-treatments and the thermal behaviour of the thermocouples are also included. Heat conduction between the modules is neglected because its effect is smaller than the zero-dimensional model accuracy.

Based on the current driving condition (engine speed, engine power and velocity of the vehicle) the model calculates the temperatures for up to four different positions in the exhaust system. Figure 2.7 shows the basic structure of the thermal model for the exhaust gas aftertreatment system.

maps for temperature exh. out, lambda, massflow

maps for emissions

Calculations in MATLAB code

maps for conversion rates of catalyst
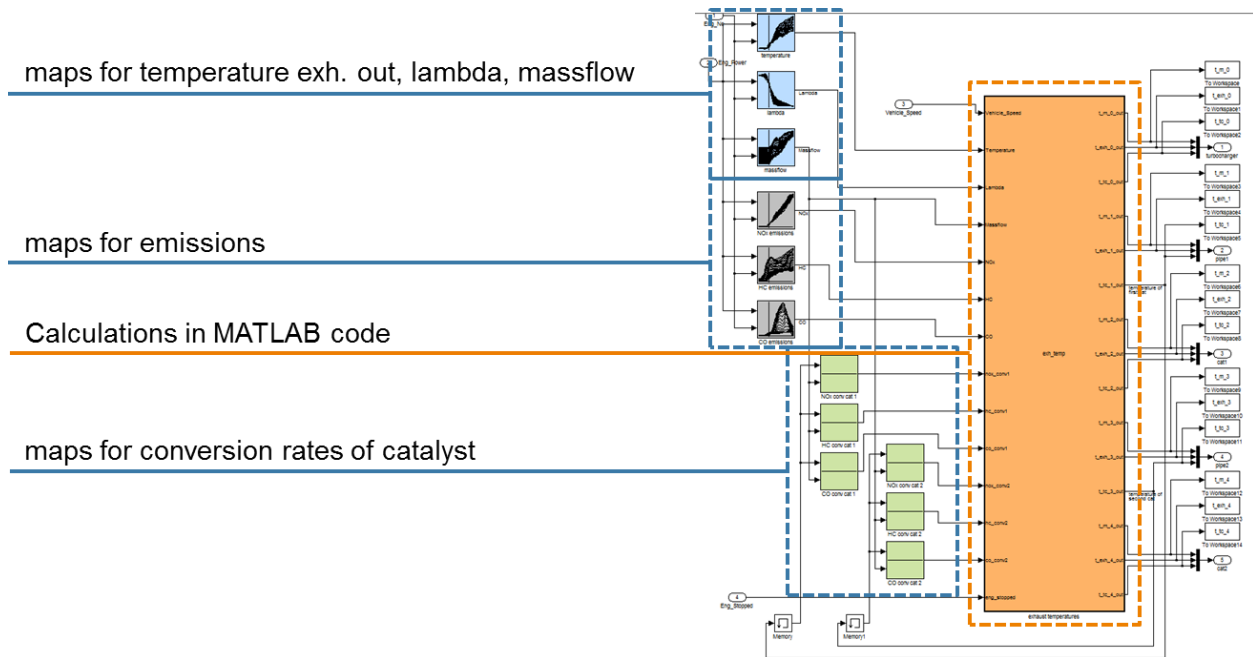
Figure 2.7:   Structure of thermal model for the exhaust gas aftertreatment system

The main calculation is done inside the embedded MATLAB function block "exhaust temperatures" which uses the values of lookup tables and some values of the current operating condition.

All the parameters and maps are set as generic and normalized values from existing average measurement data. It is suggested to use the model with the existing generic settings since parameterizing would require a lot of measurement effort for temperatures in the exhaust system, data evaluation and generation of maps as well as a defined standard operating procedure for parameterizing the model. Exact simulation of the vehicle specific temperatures is not intended, the model should only deliver reasonable values for the temperatures of the aftertreatment system during a cold start operation of the hybrid system as input for the hybrid control units to perform temperature dependent operating strategies.

If the generic parameters and maps are used the only required input parameters are engine rated power, engine rated speed and engine idle speed.

The exhaust system is defined by five different modules numbered from 0 to 4 and are presented in the following sections:

    Module 0: Turbocharger and multifold
    Module 1: First pipe section
    Module 2: First after-treatment
    Module 3: Second pipe section
    Module 4: Second after-treatment

The model calculates the following output values, where X (value from 0 to 4) corresponds to the respective module (i.e. component) of the exhaust system:

    t_m_X_out   mass temperature of module X, [K]
    t_exh_X_out   exhaust gas temperature of module X, [K]
    t_tc_X_out   thermocouple temperature of module X, [K]

This output values are available on the data bus inside the vehicle model and can be used as inputs for the respective control units of the hybrid vehicle.

2.6.3 Electric components

The temperatures of the electric storage and the electric motor are calculated by an energy balance of the energy losses heating up the respective component and energy dissipated by a liquid cooling flow. The equations used for calculating the component temperatures are the same for both components, the calculation of the power losses is explained below in separate sections.

For the calculation of the heating of a component a simple point-mass system combined with a liquid cooling is used.

The heat flow from the point-mass to the cooling liquid is calculated as:

$$P_{cooling} = \frac{T - T_{out}}{R_{th}}$$

For numerical reasons $P_{cooling}$ depends only on T_{out} (and not on T_{in}), else unreasonable values would occur, when the mass flow of the liquid $\dot{m}_{fluid}$ becomes zero.

Due to the supplied heat, the cooling liquid is heated as:

$$T_{out} - T_{in} = \frac{P_{cooling}}{c_{fluid} \cdot \dot{m}_{fluid}}$$

Combining the last two equations leads to:

$$T_{out} = \frac{T + T_{in} \cdot R_{th} \cdot c_{fluid} \cdot \dot{m}_{fluid}}{1 + R_{th} \cdot c_{fluid} \cdot \dot{m}_{fluid}}$$

This equation simplifies to $T_{out} = T$ in case the cooling is switched off ($\dot{m}_{fluid} = 0$).

The temperature of the point-mass is calculated as:

$$T = \frac{1}{C} \int (P_{loss} - P_{cooling}) \, dt$$

with:

T	temperature of the point-mass, [K]
T_{in}; T_{out}	cooling fluid inlet and outlet temperature, [K]
P_{loss}	power loss of the system, [W]
$P_{cooling}$	heat flow from the point-mass to the cooling liquid, [W]
C	heat capacity of the system, $[\frac{J}{K}]$
C_{fluid}	specific heat capacity of the cooling liquid, $[\frac{J}{kg \cdot K}]$
$\dot{m}_{fluid}$	mass flow of the cooling liquid, $[\frac{kg}{s}]$
R_{th}	absolute thermal resistance, $[\frac{K}{W}]$

The estimation of the cooling parameters for the component (C and R_{th}) is given in the following chapters.

The corresponding simulation model is shown in Figure 2.8.
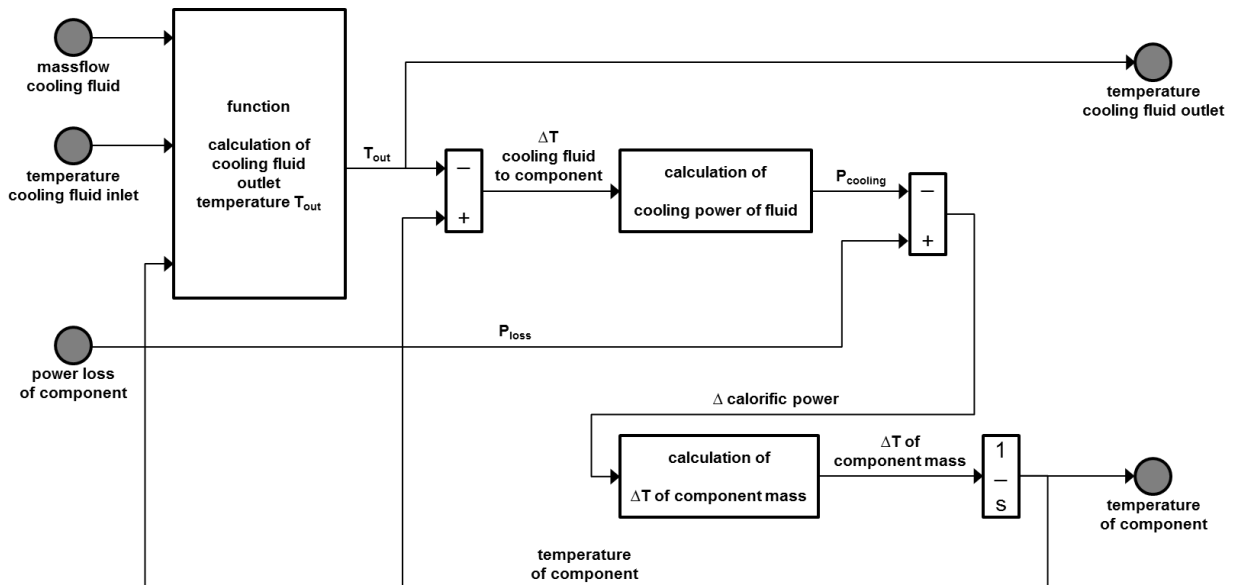
Figure 2.8:   Block scheme for component coolant model

The input signal of the cooling fluid mass flow ($\dot{m}_{fluid}$) has to be provided by the OEM specific interface model since there has to be some logics implemented depending on several actual temperatures (e.g. component temperature, cooling fluid temperature, etc.).

For the input signal of the cooling fluid inlet temperature (T_{in}) it is suggested in a first stage to set it to a constant value, since modeling of a heat exchanger to the ambient air would be a vehicle specific solution and require a lot of effort to parameterize the model. This would correspond to operating the hybrid system on a test bench with an external conditioning unit for the cooling fluid. Also, the approach should be in line with the procedure for powertrain testing of the hybrid system according to the part of the GTR regulation drafted with the input of EPA. The final solution will be decided based on the procedure for powertrain testing as well as on the outcome of the vehicle simulation according to the HILS procedure in VTP2.

2.6.3.1   Electric storage

For the electric storage the existing model had to be extended to a more complex model in order to get a more accurate temperature behaviour. The extended model provides a better and more realistic description of a time-dependent current-voltage behaviour through the additional RC-circuit. In the original model according to Kokujikan No.281, consisting only of a resistive part, a rectangular current signal leads to a rectangular voltage behaviour. Whereas the extended model with additional RC-circuit leads to a time-dependent voltage behaviour with the same rectangular current signal as input (see Figure 2.9).

This time dependent current-voltage behaviour allows a more accurate calculation of the power loss in the electric storage which is needed for temperature calculations.
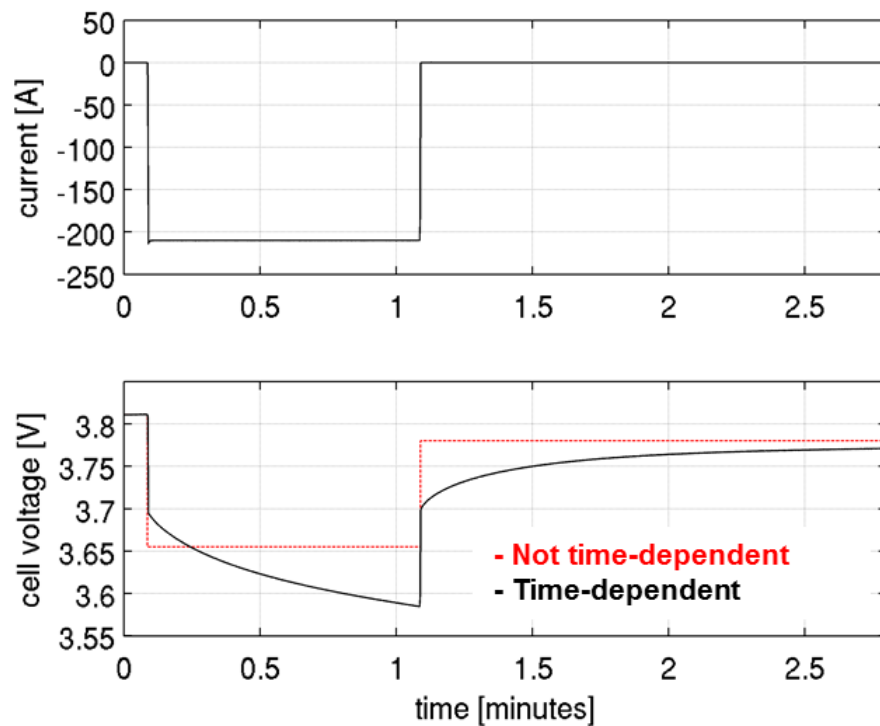
Figure 2.9:   Current voltage behaviour of different electric storage models

The losses are calculated as resistive losses in the parts R and R0 of the model. If the thermal behaviour of the electric storage needs to be simulated this more complex model has to be used. See chapter 4.1.2.2 sub item 4 for the necessary changes in component testing and appendix A7.2 for a detailed explanation of the extended component model.

To measure the heat capacity of the storage the temperature rise during the injection of a known heating power is measured. The cooling system must be turned off for this measurement.
A possibility to inject the needed heating power is to drive the storage with a symmetric rectangular current signal I (period duration T) at maximum allowed current amplitude. Since it is symmetric the state of charge is not changed over time:

$$\Delta SOC = \int_0^T I\,dt = 0$$

But due to the conversion efficiency some amount of energy is converted to heat, which can be directly calculated from the measurement values:

$$P_{loss} = \frac{1}{T}\int_0^T V \cdot I\,dt$$

Given two temperature measurements T_1 and T_2 at a representative location for a temperature sensor, with time Δt in between, the thermal capacity C of the electric storage can be calculated as:

$$C = \frac{P_{loss} \cdot \Delta t}{T_2 - T_1}$$

For the determination of the thermal resistance the cooling system has to be turned on (nominal flow rate) and after sufficient time, so that the system is settled, the storage temperature $T_{storage}$ at a representative location for a temperature sensor and fluid outlet temperature T_{out} is measured, leading to:

$$R_{th} = \frac{T_{storage} - T_{out}}{P_{loss}}$$

2.6.3.2   Electric motor

For the electric motor the losses are calculated as the absolute value of the difference between mechanical power and electrical power:

$$P_{loss} = |P_{mechanical} - P_{electrical}|$$

In order to simplify further calculations, it is assumed that the complete power loss is dissipated via the liquid cooling circuit. In order to get a more accurate estimation of the power loss the accuracy of the measurement devices has to be higher than specified in Kokujikan No.281. The torque measurement shall be better than 0.5 % of the actual reading, the measuring of the revolution speed better than 0.1 % of the actual reading. The measurement of the input voltage of the controller shall have an accuracy better than 0.1 % of the displayed reading and the input current shall be measured with an accuracy better than 0.3 % of the displayed reading. The resolution of the thermometer shall be better than 0.1 K to be able to measure a small warming.

To measure the heat capacity of the electric machine the temperature rise during the injection of a known heating power is measured. The cooling system must be turned off for this measurement.

A possibility to inject the needed heating power is to drive the machine (stator winding) with a constant current signal I_{stator} at maximum allowed current amplitude. By additionally measuring the stator winding voltage drop V_{stator} the power loss can be calculated as:

$$P_{loss} = V_{stator} \cdot I_{stator}$$

Given two temperature measurements T_1 and T_2 at a representative location for a temperature sensor, with time Δt in between, the thermal capacity C of the machine can be calculated as:

$$C = \frac{P_{loss} \cdot \Delta t}{T_2 - T_1}$$

For the determination of the thermal resistance the cooling system has to be turned on (nominal flow rate) and after sufficient time, so that the system is settled, the machine temperature $T_{machine}$ at a representative location for a temperature sensor and fluid outlet temperature T_{out} is measured, leading to:

$$R_{th} = \frac{T_{machine} - T_{out}}{P_{loss}}$$

2.7   Task 1.7 - Simulation runs and validation of basic functions

This tasks main goal is to check the basic functionality of the developed models based on the Japanese model structure, which means it is to show that they mainly run stable and deliver physically reasonable results. The headline may indicate something else, but this has nothing in common with the validation process mentioned in Kokujikan No.281 where simulation models are compared with actual measurement data in order to proof their validity.

Due to the model restructuring in a later project phase the simulation runs with the Japanese model structure based serial hybrid model which have been carried out have lost their importance. For completeness of the report they are mentioned in an abbreviated form.

Several simulation runs were done to test the operational suitability and robustness of the developed Japanese structure based serial hybrid model containing

* Thermal models for electric motor, generator, energy storage system, ICE oil, ICE coolant water and the ICE exhaust system
* A redesigned energy storage system
* Driver models for vehicle speed and propulsion power

As a key constraint the desired vehicle speed or the desired power demand always had to be tracked within the given limits for each test run. Different generic vehicle data for a fixed final drive ratio was used to run the model with either the vehicle speed or the power cycle driver model. Figure 2.10 illustrates the model output representative for the 10.4 ton vehicle for the first 500 seconds of the WHVC using the driver model to follow a given vehicle speed without any violation of the specified limits (± 2 km/h, 1 sec.). Although the temperature behaviour looks very linear this is reasonable because the ICE is operated in stationary conditions. The right upper chart describes the temperatures for the ICE. The right lower chart illustrates the temperatures for one specific point in the exhaust system where a thermocouple is located.

	Vehicle mass [t]	Tire radius [m]	Rated power [kW]
Vehicle 1	8,2	0,32	165
Vehicle 2	10,4	0,47	210
Vehicle 3	12	0,32	210
Vehicle 4	40	0,47	315
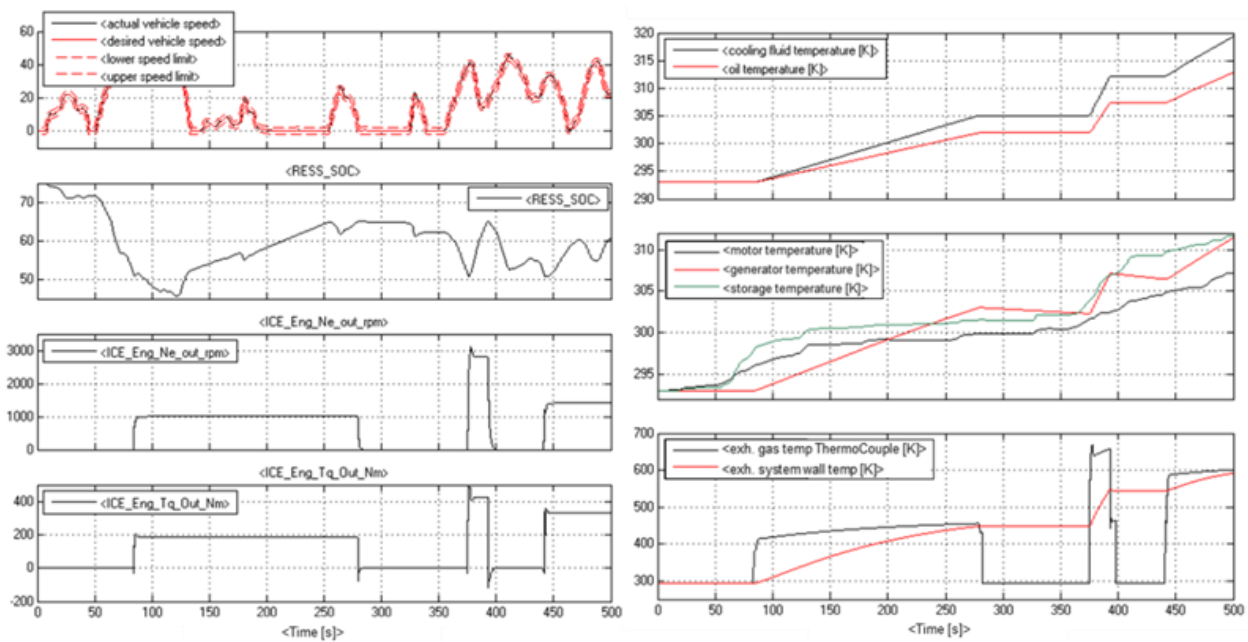
Figure 2.10:   Example of a Japanese structure based serial hybrid model output using the speed driver model, 10.4 tons, 201kW

Results for the identical vehicle but using the driver model which is able to follow a given power time curve are plotted in Figure 2.11. The upper 4 charts belong to the torque controller, the lower 4 charts to the speed controller. It is shown that the driver model is able to follow the given rotational speed and the given torque within the tolerances. The drag manipulator is the speed controllers control variable and additionally applies a road load in order to meet the given rotational speed at the respective load torque. The high fluctuations indicate a high fluctuation of the power cycle which is mainly caused by gear shift events in the WHTC. Section 4.3.1 will in detail deal with that issues.

Detailed simulation results for the new structured serial and parallel hybrid vehicle models will be bundled in section 3.4.

2.8   Summary

Main objectives of Task 1 have been:
- The preparation of a serial hybrid model using SIL simulation
- Providing additional powertrain components/models in order to meet stakeholder demands and ensure the establishment of a comprehensive model library
- And providing different driver models in order to be able to perform model test runs, investigate the model behaviour and the impacts of different test cycles

With regard to the previous bullet points the achievements can be summarized as follows:
- A basic serial hybrid model provided by our Japanese colleagues could be extended and model test runs could successfully be performed with new components, different driver models and different vehicle parameters

- New powertrain components have been developed and already transferred into the later introduced new model structure (except planetary gear set)
- The implementation of a driver model capable of running a test cycle referenced to a certain power time curve could be successfully tested but faced some serious weak points related to the test cycle itself. In contrast to the conventional driver model (tracking the vehicle speed) it was therefore not transferred into the new model structure until now

Figure 2.11:   operating conditions of a serial hybrid at the powertrain output shaft using the power driver model to run a WHDHC, 10.4 tons, 201kW

3 Task 2 - Adaptation of the GTR-HILS simulator for parallel hybrid

Task 2, as defined by the UNECE HDH informal working group, consists of the following activities:
2.1 Meeting with OEMs and stakeholders
2.2 Set up a data bus system in the model to allow various combinations of engines, gear boxes and storage systems.
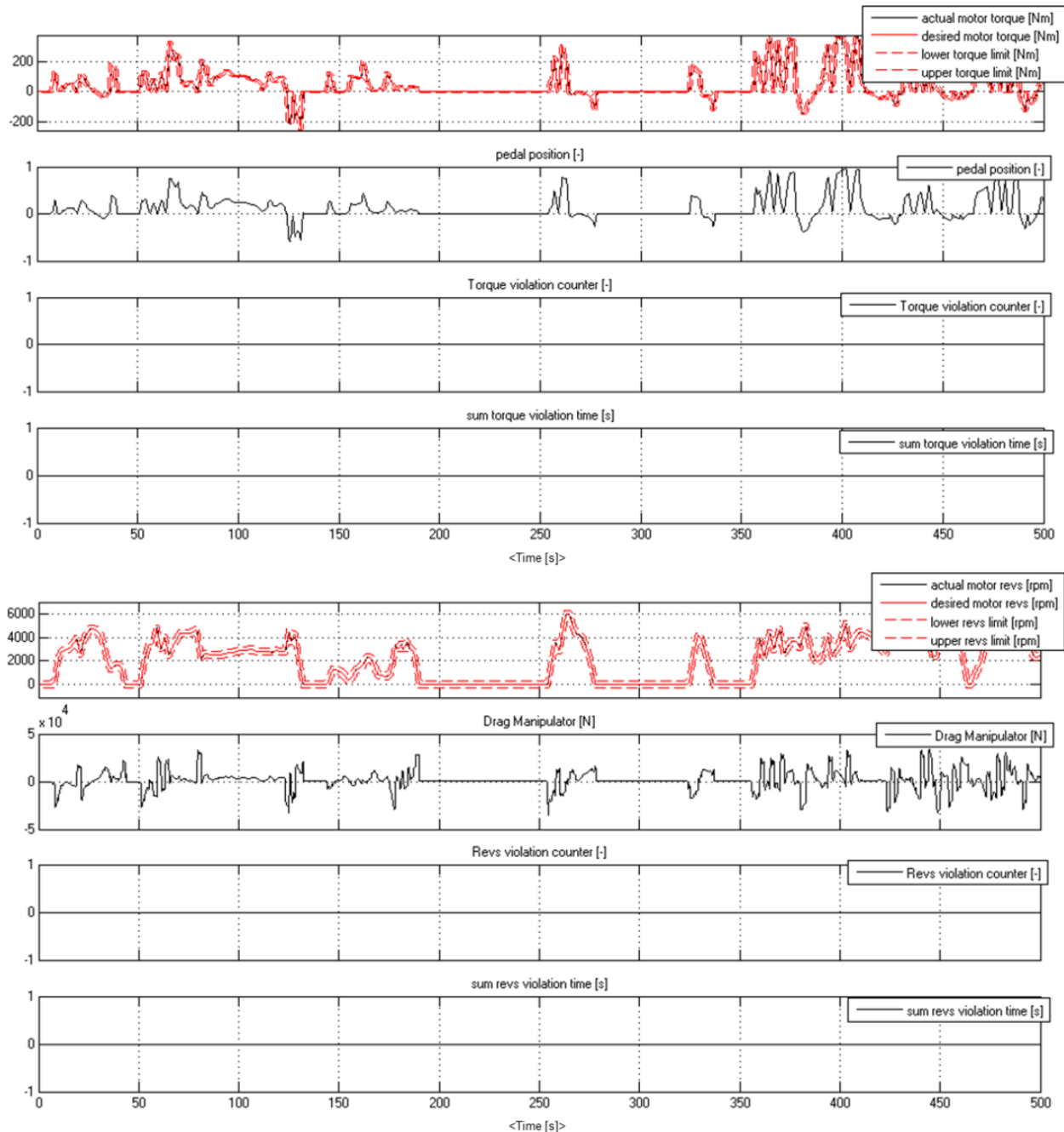2.3 Adapt the software to simulate a parallel HDH.
2.4 Simulation run and validation of basic functions, including the functions from task 1.

The results of these activities are reported in the subsections of this chapter.

3.1 Task 2.1 - Meetings with OEMs and stakeholders

During the entire project, meetings with several OEMs, by name Daimler, Volvo, Scania, MAN and DAF, have been held and a close and fruitful cooperation could be established. The meetings were primarily used to present the Japanese HDH certification procedure to HDH experts not familiar in detail with the HDH meeting documents and to discuss open issues bilaterally. This section reports the most important outcomes of meetings and resulting phone conferences held and is therefore subdivided in different topics.

3.1.1 *HILS model interface and interface model*
An initial aim of this task was to define a standardized HILS model interface, which fits for each manufacturer and allows to run the HILS model with actual hardware ECUs. The discussions were based on the existing Japanese model interface where certain model in- and output signals are specified. Soon it turned out that defining a fixed interface suitable for each manufacturer and different vehicle concepts will probably not be feasible. The need of restructuring the HILS models in consequence solved this task by the introduction of a flexible signal bus where every user is allowed to add or remove signals needed to run the HILS model respectively the actual ECUs connected. Default signals that are needed to run the provided component models from the model library are specified in Appendix B - Interface signals. If a component model is replaced by a manufacturer specific component model which fits to the proposed model structure the interface signals needed can easily be changed.

A second issue was the interface model itself which is placed between the interface of the vehicle model and the actual hardware ECU in order to allow signal level tuning etc. (see Figure 3.4). This interface model is once validated within the whole model validation process. For certifying different vehicles with a validated model the need of an adaptation of the interface model will raise depending on the variety of used ECUs. The Japanese legislation described in the Kokujikan papers specifies that minor changes (e.g. additional signals for anti-lock braking system), which are not emission relevant, are allowed but have to be agreed by the authorities. For major changes in the interface model a new overall model verification would be needed. Due to a different ECU topology between vehicles in the Japanese market

and European ones experience has to be gathered regarding the need and extent of adaptations for different European vehicles (see also section 4.1.2.5-1 and OIL/V2).

3.1.2   *HDH drive cycle*

During the meeting with Daimler a good contact could be established and especially the work on a valid drive cycle for HDHs was intensified. In an earlier project phase the Japanese open source model was remodeled to roughly represent a conventional vehicle from Daimler. A gearshift strategy was implemented in a way that the resulting engine operation pattern looks similar to the WHTC for the vehicle running at the WTVC. The focus of investigations was laid on the modelling depth and the impact of traction force interruptions during gearshift events on emissions. Therefore rotational speed and torque patterns for the ICE were generated with and without modeled traction force interruptions during gearshifts to be proofed on the engine test bed. The respective ICE operation patterns were provided in 1Hz, 10Hz and 50Hz to proof also the impact of the cycle resolution on emissions. Detailed results have not been presented yet but the institutes recommend a resolution of the ICE test cycle at 10Hz. This would be in line with the recommendations for the command values of the WHTC in GTR No.4. A frequency of 50Hz has not been possible at all and should be discarded from the considerations (see OIL/H3 and H4).

During the development process of a vehicle speed based drive cycle, which has similar work demand than a respective WHTC several telephone, conferences were held. Currently a very insightful investigation is planned where a very accurate OEM intern conventional vehicle model should be used to run a WTVC with road gradients. The resulting ICE operation pattern should be proofed on the engine test bed and the emissions should be compared to the emissions released during a WHTC for the same engine. This will give more insights on the general comparability of the test method for conventional HD vehicles and the planned test method for HDHs (see OIL/D2).

Especially for low volume and niche HDH vehicles there was the request of several OEMs that the WHTC should remain valid as alternative type approval test because the HILS method would be very high effort. The HILS method would only be applied if the usage would be beneficial for the OEM in a certain way. This of course has to be discussed in the HDH investigation group.

3.1.3   *HILS model and demands (see OIL/C7)*

When the first new structured models were lined up the feedback from involved parties was in general very positive. Useful comments on general and specific needs were given and will be implemented in a new model release. Especially for the need of different brake systems for different OEMs in the driver model a flexible solution has to be found. It also turned out that the models will not be able to represent each vehicle tested in VTP2 with the existing components. The MAN bus requires additional components to be modeled which have been identified as a DC/DC - Converter to run electric components on different voltage levels, a braking resistor to dissipate energy and control energy flows and an automatic transmission gearbox with torque converter. Although only 3 weeks of modelling and validating new components was planned originally all components except the braking resistor could already be provided until now.

Different demands were identified regarding the capabilities and the exchangeability of the driver model. The basic idea is to keep it as simple as possible. Therefore a generic driver model is provided which should be sufficient for a general use. Nevertheless replacing should be allowed otherwise the generic driver model will get very complex and will in fact never be able to meet all demands for future vehicles when thinking about e.g. actuating vehicle specific levers / functions. Flexibility is needed here but it might be considered that a specific driver model is then only valid for the certification of one specific vehicle or for a powertrain that enables the same possibilities regardless the vehicle it is mounted in.

3.1.4   Test procedure

Regarding the HILS test procedure it was intensively discussed how to handle multiple ECUs within a HILS test environment. It is expected that at least 3 to 4 ECUs are needed so the common opinion was to avoid multiple ECUs at the test rig in order to reduce effort, costs and complexity. One approach is to define a representative hardware ECU which should be chosen for the test. The surrounding ECUs should be represented as SILS solution in the interface model. Regardless how multiple ECUs are handled finally a pure SILS solution is clearly preferred by the automotive manufacturers (see also section 4.1.2.5 - ECU and multi-ECUs).

Discussions about actual ECUs at the test rig also raised the question of how to deal with dummy signals needed in order to avoid failure modes in the ECUs. This can be handled either by the interface model or specific test modes are foreseen in the ECUs. Of course it is than questionable if the software is still representative for the in vehicle behaviour (see also section 4.3.2.5).

Discussing the cold start issue at 20°C it was reported that the performance of several hybrid components (battery, inverter, motor,…) will not be temperature dependent at this temperature level. This is convenient since then there is no need for costly temperature dependent component tests.

Issues mentioned above in this section are also stated in the OIL in chapter 5.

Regarding the real world behaviour it was stimulated that auxiliaries are disabled in the HILS certification test run although they affect the control strategy of a hybrid system. Since the WHTC for conventional vehicles does not consider auxiliaries that approach was also followed for the HILS method and it was decided to reject PTOs for a GTR adoption. This is reasonable because thinking of a constant additional power take off, the ICE has to deliver the additional work in the end as the SOC for the certification run has to be balanced. In fact the total mass of emissions would be increased but since they are divided by the work done there should be a minor impact. The hybrid system would only allow to apply the power more flexible.

Just a remark to be mentioned is the fact that it is crucial to provide vehicle speed and to offer the possibility to estimate the vehicle's mass/inertia when considering a complete vehicle independent certification approach. This is important and needed for ECU logics (e.g. gear shifting) and indicates that also for the post transmission powertrain testing a vehicle model needs to be run in background where specific data has to be provided. For a pre transmission powertrain testing a conversion program from WTVC to ICE operation pattern would be needed.

3.1.5   *CO$_2$ legislation*

Since CO$_2$ is in the mandate this issue was also discussed with OEMs but is considered as problematic within the HILS process. The first proposal where a power time curve from the CO$_2$ tool for conventional vehicles (VECTO [12]) will be handed over to the HILS tool to derive the ICE operation pattern has still some open tasks to be solved. In fact a HILS model run would be needed for each derivate of a vehicle (e.g. with spoiler, without spoiler,...) to depict its CO$_2$ value. This seems not feasible. Different approaches (e.g. comparative factors,...) need to be found. It cannot be reported here in detail because the investigations are still on-going.

3.1.6   *Summary*

Identified comments / issues that needed to be addressed based on OEM feedback are:

- Changes allowed in interface model
- Consideration of traction force interruption at the HILS model run
- Test cycle command frequency for an engine emission test
- Possibility of certifying HDHs using a WHTC engine test
- Multiple ECU handling
- Dummy signal handling / ECU test modes

All issues are also summarized in the OIL in chapter 5.

3.2   **Task 2.2 - Set up a data bus system in the model**

To set up a simulation tool, which allows a well-defined selection and combination of the components included in the library in the HILS simulator, the structure of a data flow shall be adapted. The structure shall follow a bus system or similar with defined interactions of each module of the library. The design shall simplify adaptations of the HILS simulator to different hybrid systems in the future type approval applications.

This task has partly changed from the original description with as the main reasons:

- Components are represented in different ways in the two vehicle models (the series and the parallel).
- Components are lumped in different ways in the two vehicle models (the series and the parallel).

Consequently, it is difficult in the open-source models to set up a data bus, there is a need to restructure the models. Chalmers has started this work with finance from the Swedish Road Administration. The work will focus on defining and standardizing model structures that can be proposed and/or used in a GTR.

3.2.1   *Model structures and interface signals*

For complete vehicle simulation it is preferable when the component models can be connected together in a straightforward manner to form a complete vehicle model. In Figure 3.1 an idea of a HILS/SILS simulation model structure is presented.
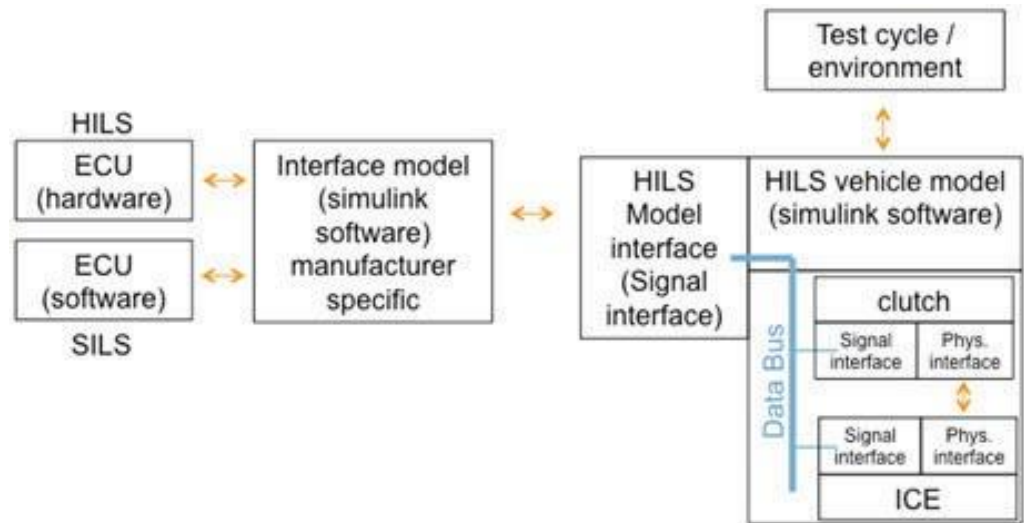
Figure 3.1:   HILS certification model

The modeling philosophy that is suitable for HILS/SILS applications is called forwarding, which means that the powertrain is described by models described by differential equations.   This makes it possible to take into account dynamic effects such as engine speed-up and vehicle inertia etc. The other alternative, called back warding, is usually based on quasi-static models. Such descriptions can be simulated much faster, but the result does not describe transient effect. Furthermore, in back warding feedback control cannot be used.
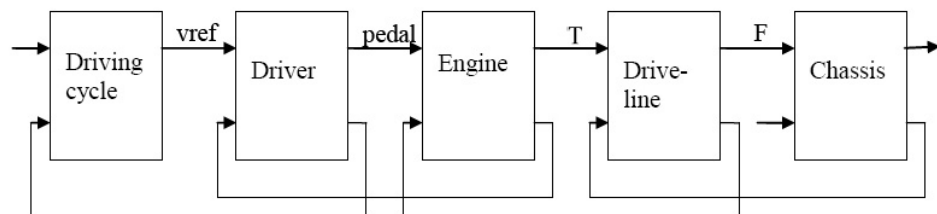
Figure 3.2:   Model structure for a powertrain model using forwarding

Dynamic simulation or forwarding is outlined in Figure 3.2, this idea is also used, more or less, in the open-source models. The name forwarding comes from the fact that the current subsystem is using information determined in subsystems in front of the current subsystem. The idea is to use a driving cycle, to set the desired vehicle velocity for the driver. The driver utilizes the desired velocity and the current velocity in order to command the vehicle by using the pedals, very similar to what the driver does in a chassis dynamometer setup in reality. The driver is usually represented with some sort of control system. In turn, the engine uses command signals from the driver and a control system and feedback signals from the driveline in order to calculate the current engine states, and so on. In order to achieve this, the model interfaces between the powertrain components needs to be determined.

Two types of interfaces are needed:

- The *physical interface* is related to how different components are connected together physically
- The *signal interface* is related to control/sensor signals needed to control the components for an ECU

It is important to define good interfaces capturing all necessary information shared by the different objects. The idea is to use a port-based modelling paradigm. The communication signals between the different components are physical signals, like electric wires, mechanical joints etc. The interfaces or connectors (*physical interfaces*) are based on energy flow to and from the component or through a port. A port is characterized by an across and a through variable, also known as flow and effort variables in Bond Graph modeling. The interfaces are a key to exchangeability of component models.

For automotive powertrains, four (five) different physical interfaces are necessary, the interfaces are: electrical, mechanical (rotational and translational), chemical and fluid. The table below shows a proposal for physical interface signals.

	Electrical	Mechanical (rotational, translational)	Chemical	Fluid
Flow	Voltage [V]	Torque [Nm], Force [N]	Specific energy [J/kg]	Pressure [Pa]
Effort	Current [A]	Speed [rad/s], Velocity [m/s]	Mass flow [kg/s]	Flow [m^3/s]

The physical interface proposed is based on best-practice from a number of vehicle powertrain simulation tools, Autonomi, ADVANCE, Dymola (Powertrain library), CAPSim, VSIM, TruckSim.

The port-based modelling paradigm is complemented with a signal interface, for making it possible to control each component.

3.2.2 Naming convention

The following naming convention for the physical interface signals are used:

- *Physical interface:* `phys_description_Unit`

Where `phys` is fixed to indicate that it is a physical signal, `description` is a description of the signal, e.g. torque `torque`, voltage `voltage` and `Unit` is the unit of the signal in SI-units, e.g. Nm, V, A etc. An example: `phys_torque_Nm`, which is the physical torque in a component model.

For the signal interface, the naming convention follows the AUTOSAR, [5], standard as far as possible:

- *Signal interface:* `Component_description_Unit`

where `Component` is the component short name, e.g. Clu, Engine, ElecMac etc., `description` is a description of the signal, e.g. actual torque `tqAct`, voltage `u`

and `Unit` is the unit of the signal in SI-units, e.g. Nm, V, A, rad/s etc. An example: `ElecMac_nAct_radps`, which is the actual rotational speed of an electric machine, the speed is in rad/s.

The physical interface and the signal interface for all powertrain component models are available in Appendix B.

3.2.3   Component model structure in Simulink

The following model structure is proposed, see Figure 3.3. The model structure has been presented at several HDH meetings and it has been accepted for use.
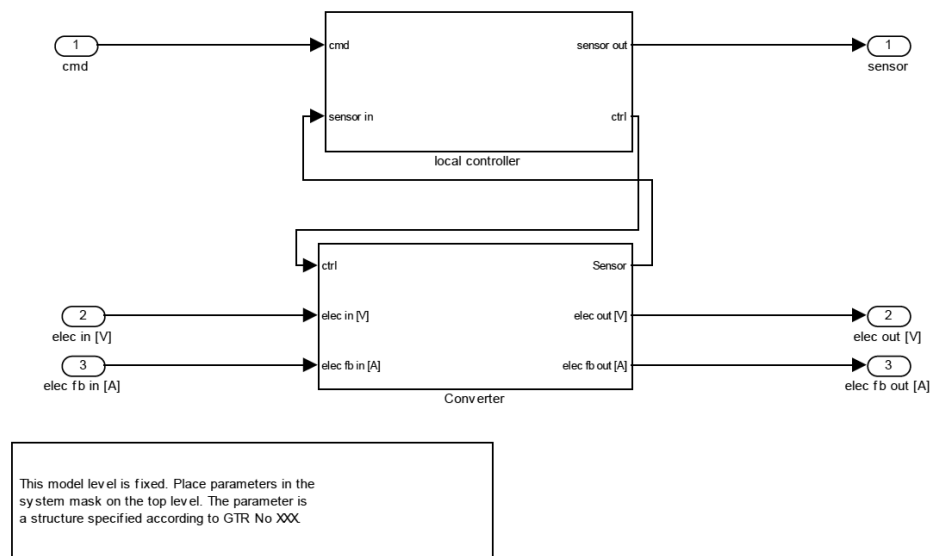
Figure 3.3:   Model structure (example)

All component models except the driver use the proposed model structure. The model structure is divided into two parts, the physical model and the local controller. Every model includes a local controller, which converts the control signals from the control system (if existing) into local control signals, the block also sends sensor signal values to the control system, i.e. it handles the communication between the control system (ECU) and the physical model. The physical model block should include the implementation of the model equations.

In the Simulink implementation of the physical interfaces for the mechanical components, the inertia or the mass of the component is also transferred between the components in the 'torque' interface, see the table below.

Node	Name	Variable name	Description	Unit
Electrical	elec in [V]	phys_voltage_V	voltage	V
	elec fb in [A]	phys_current_A	current	A
Mechanical	mech in [Nm]	phys_torque_Nm	torque	Nm
		phys_inertia_kgm2	inertia	kgm^2
	mech fb in [rad/s]	phys_speed_radps	speed	rad/s
Mechanical	mech in [N]	phys_force_N	force	N
		phys_mass_kg	mass	kg
	mech fb in [m/s]	phys_velocity_mps	velocity	m/s
Chemical	chem in [J/kg]	phys_specenergy_Jpkg	specific energy	J/kg
	chem fb in [kg/s]	phys_massflow_kgps	mass flow	kg/s
Fluid	fluid in [Pa]	phys_pressure_Pa	pressure	J/kg
	fuid fb in [m3/s]	phys_flow_m3ps	volume flow	m^3/s

As forwarding is used, feedback signals that go in to a block comes from the block in front of the current component block. This means that from an energy perspective the energy that goes into a component block is given as the product of the input signal and the feedback output signal. Similarly, the energy that goes out from a component block is given as the product of the output signal and the feedback input signal. As an illustrative example, consider the model in Figure 3.3. The incoming energy (energy $flow$ = power) is determined as P_{in} = elec in [V] $\times$ elec fb out [A] and the outgoing energy is given as P_{out} = elec out [V] $\times$ elec fb in [A].

3.2.4   Vehicle top level model structure

The top level for all vehicle topologies looks the same. It includes a driver model, an ECU model block and its corresponding input/output interface block for converting ECU signals into the proposed signal interface and the powertrain block, see Figure 3.4.

Figure 3.4:   Vehicle top level model

The ECU block is replaced by the real ECU when performing a HILS simulation. The input interface block is modified in order to convert HILS model signals into desired/needed ECU signals in order to be able to run the ECU. The output

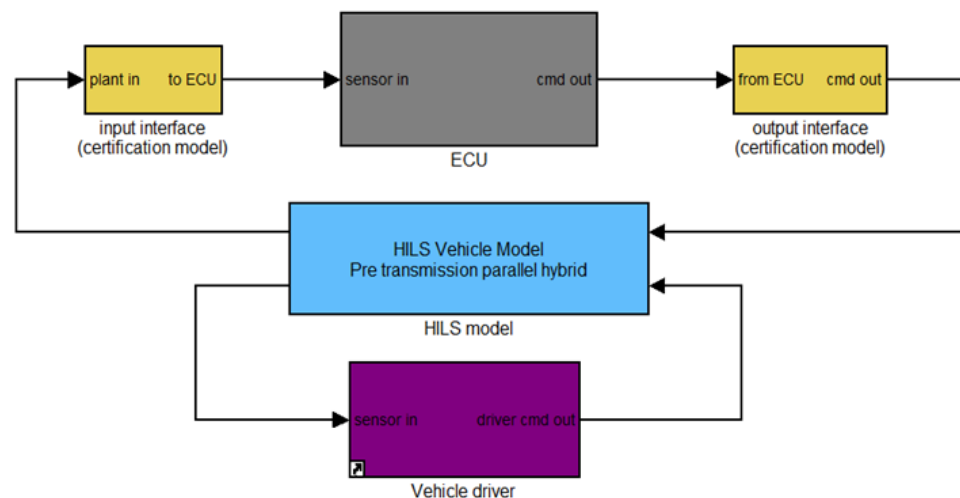interface block is modified in order to convert the ECU signals into signals required by the HILS model in order to be able to run. See Appendix B for signals available from the HILS model and signals required for the HILS model.

3.2.5   Model library

Based on the new proposed model structure, which offers flexibility and exchangeability, the open source models are remodeled as separate component models and implemented into a model library, the documentation of the component models is available in Appendix A. The component models are categorized into different categories. The models are categorized into the following categories:

- Auxiliary system
- Chassis
- Driver
- Electrical components
- Energy converters
- Mechanical components
- Rechargeable energy storage systems

Each category contains component models related to that specific category. The model library is part of a toolbox. The toolbox is organized as shown below:

```
HILS_GTR............................ The main folder
   |--> Documentation................. Model documentation is located here
   |--> DrivingCycles................. Data of the different driving cycles
   |--> Library....................... The model library is located here
        |--> ParameterFiles........... Template parameter files for all
                                        models (copy if used)
   |--> Misc.......................... All additional files for the
                                        HILS model library are stored here
   |--> Vehicles...................... Vehicle models are stored here
        |--> Parallel................. Models for parallel hybrid vehicles
             |--> PostTransmission..... Post transmission hybrids powertrains
                  |--> ParameterData... Data of the different component models
                  |--> SimResults...... Simulation results
             |--> PreTransmission...... Pre transmission hybrid powertrains
                  |--> ParameterData... Data of the different component models
                  |--> SimResults...... Simulation results
        |--> Series................... Series hybrid powertrains
             |--> ParameterData....... Data of the different component models
                  |--> SimResults...... Simulation results
```

The different directories contain files important for the toolbox and library to work. The library is developed for MATLAB 2012a.

3.2.6   Documentation

A directory containing the documentation of the different component models, the modeling philosophy and examples.

3.2.7   Driving Cycles

A directory containing different driving cycles that can be used in the toolbox. The driving cycles are implemented as a vehicle velocity profile as function of time. The driving cycles are saved as mat-files and can be loaded into MATLAB's workspace using the load function.

3.2.8 Library

The model library is located in this directory. The component models are categorized into different categories as mention earlier. In each category, different component models are available. The main reason for using a model library is that modifications can easily be broadcast to all powertrain models using the library, this secures that all powertrain models are up-to-date.

3.2.9 Parameter files

For each component model in the library there is a corresponding parameter file associated to the model. The parameter file contains all parameters that need to be inputted in order to simulate the model. If a component model is included in a powertrain model, it is a good idea to copy the corresponding parameter file and modify the parameters according to the component modeled.

3.2.10 Misc

This folder contains functions used for pre or post processing of data and/or simulation results.

3.2.11 Vehicles

This folder contains example powertrain models; modeled using the model library. It contains one series hybrid powertrain model and two parallel hybrid powertrain models, one pre-transmission parallel hybrid powertrain model and one post-transmission parallel powertrain model.

3.2.12 Summary

In order to fulfill the task, a new model structure has been proposed and implemented. Also a new model library has been developed based on the model structure. Component models build up the model library. Using a component-based modeling philosophy offers flexibility for different hybrid systems and makes it easy to include new or future hybrid systems. The component models in the library are based on the Japanese component models presented in Kokujikan No.281, with modifications to fit the proposed model structure.

3.3 Task 2.3 - Adapt the software to simulate a parallel HDH

The software package with ECU-functions implemented as software (SILS) shall be tested also for parallel hybrid systems. For this work software for ECU functions of a parallel hybrid has to be developed/adapted.

3.3.1 Open source models

The open source parallel hybrid model that was provided by JARI has been complemented with an ECU control strategy and a driver model, see Figure 3.5. The driver model is the same as developed in Task 1 (out of scope of this contract).
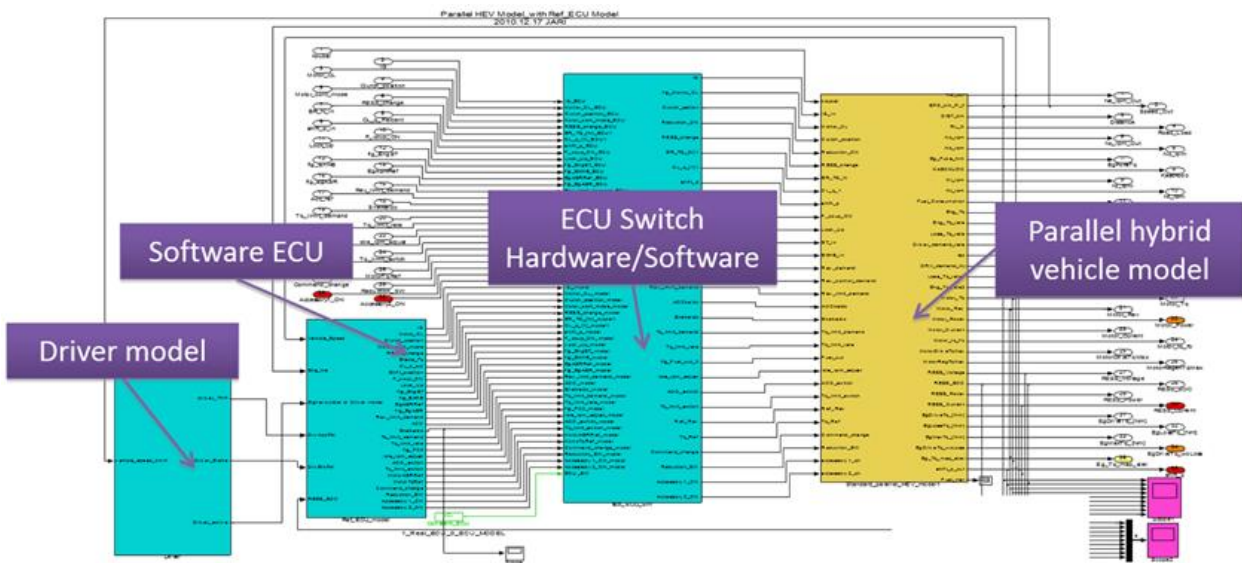
Figure 3.5:   Parallel hybrid vehicle in Simulink

The following control strategy has been implemented:

"The control strategy is to use the electric machine below a certain vehicle speed and the combustion engine above that. If the energy level stored in the accumulator is lower than a certain value, the electric machine is used as generator and is then driven either by the engine or purely by the kinetic energy of the vehicle. The electric machine is used for braking the vehicle when possible, if the brake torque is not sufficient then the mechanical brakes are used as well. The electric machine is also used for power assist when the desired torque, interpreted from the accelerator pedal position, is larger than the combustion engine can deliver."

The open source model can be simulated, using the above presented control strategy.

3.3.2   *New restructured model*
The same control strategy has been implemented also in the new restructured model and produces similar results as for the open source model. Simulation results will be provided in the next subsection.

3.4   **Task 2.4 - Simulation runs and validation of basic functions**

This tasks main goal is to check the basic functionality of the developed models based on the new proposed model structure, which means it is to show that they mainly run stable and deliver physically reasonable results. Validation in this context means a check on basic functionality and should not be confused with the validation process mentioned in Kokujikan No.281 where simulation models are compared with actual measurement data in order to proof their validity.

In this subsection, a number of the component models that are presented are connected together with a control system for powertrain simulation. The idea is to do a SIL simulation test, in order to get more familiar with the new restructured

powertrain models and to verify that the proposed models work properly. It should be noted that this case study is pure simulation and does not include any attempts for HILS.

3.4.1   Series hybrid vehicle

A series hybrid powertrain model is built using the component models in the library. The vehicle modeled corresponds to a 10 tons vehicle, powered by a 170kW electric motor and a 25kWh electrical energy storage. The Simulink model of the vehicle is shown in Figure 3.6, it is also available as one of the example model in the model library, `hilsmodel_series_ example.mdl`.

Figure 3.6 Series hybrid powertrain in Simulink

3.4.1.1   Reference ECU model for SILS

A simple control strategy has been developed and implemented in order to be able to simulate the vehicle. The control strategy is an on-off relay control strategy:

"The electric machine is propelling the vehicle according to the driver's input, accelerator and brake pedal positions, using energy in the energy storage. When the energy level in the energy storage drops below a certain level, the combustion engine is turned on and operated at a constant operating point until the energy level reached another specified level. When this level is reached, the engine is turned off again."

3.4.1.2 *Simulation results*

Two simulations are presented in this report; the first simulation is the complete WTVC driving cycle and the second simulation is a short simulation using the first 140s of the WTVC driving cycle.

The first simulation is to present the complete simulation over a complete driving cycle. The aim of the simulation is, as mentioned earlier, to see that the models are numerically stable and produce realistic results. The simulation results are presented in Figure 3.7.

Figure 3.7   Simulation results from simulation of the WTVC driving cycle for the series hybrid powertrain.

The second simulation can be used for HILS setup approval, similar to Kokujikan No.281, section 8, page 11, Operation Check of HEV Model for Approval. Instead of using the first 120s of the JE05 driving cycle, it is proposed here to use the first 140 seconds of the WTVC driving cycle without slope. The simulation results can be used for comparison with HILS setup simulation results later, to verify that the HILS system numerically produce the same results. The same criterion as proposed in Kokujikan No.281 can be used. The simulation results from this simulation are available in the SimResults folder for the series hybrid vehicle, saved as a MATLAB mat-file. The simulation results are presented in Figure 3.8.

Figure 3.8   Simulation results for the first 140 s of the WTVC driving cycle. The engine torque and speed is omitted due to that the engine is either off or working at a constant operating point.

3.4.2   *Parallel hybrid vehicle*

A parallel pre-transmission hybrid powertrain model is built using the component models in the library. The vehicle modeled corresponds to a 6 tonnes vehicle, powered by a 192kW engine and 52 kW electric machine and a small 2kWh electrical energy storage. The Simulink model of the vehicle is shown in Figure 3.9, it is also available as one of the example model in the model library, `hilsmodel_parallel_pretrans_example.mdl`.
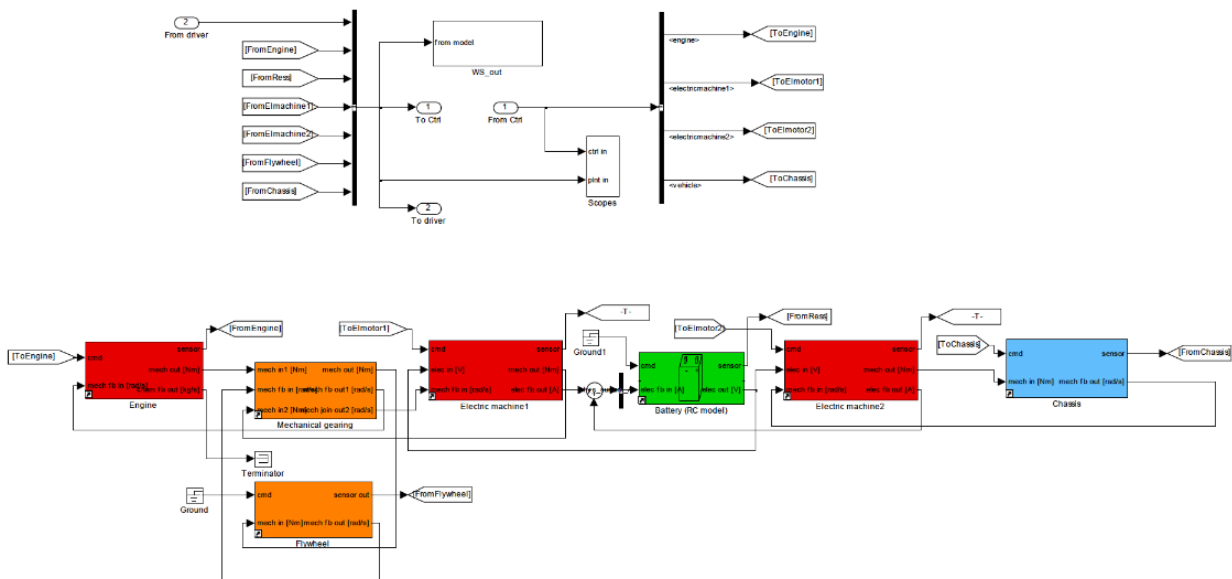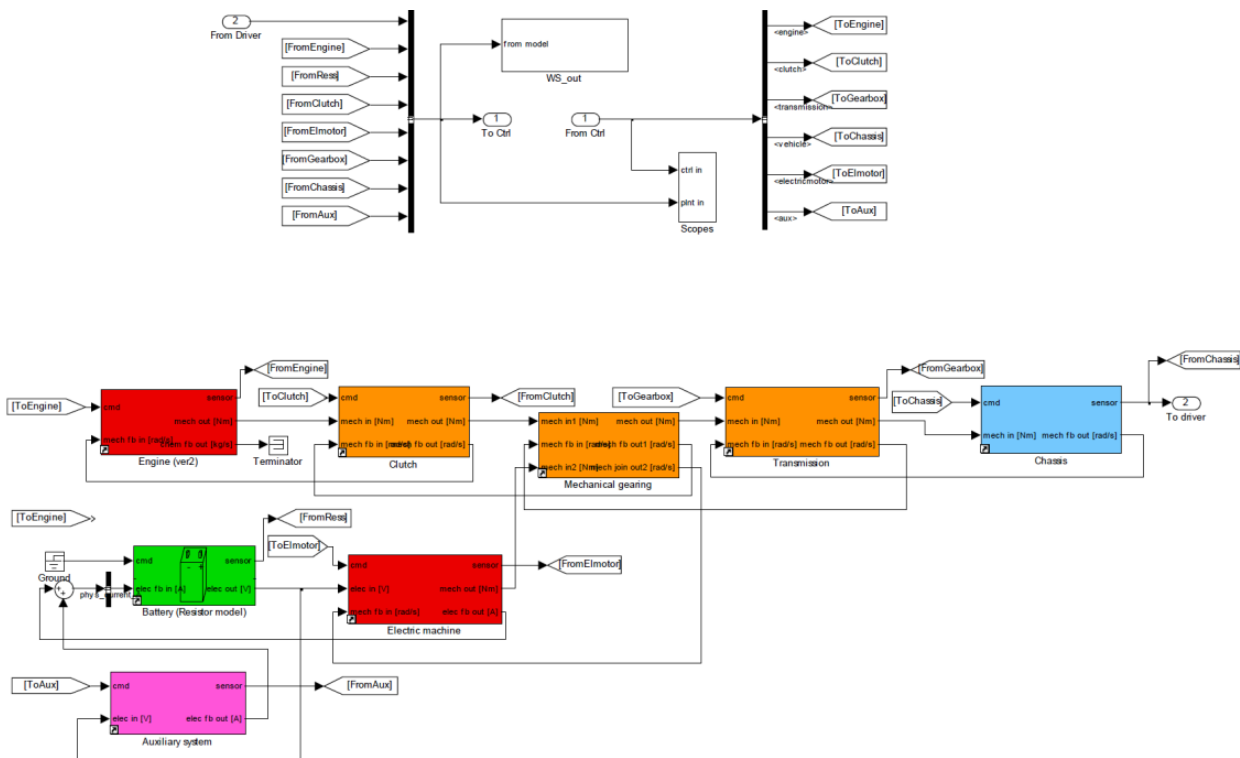
Figure 3.9 Parallel hybrid powertrain in Simulink

3.4.2.1   *Reference ECU model for SILS*

A simple control strategy has been developed and implemented in order to be able to simulate the vehicle, it is the same as mentioned in the previous subsection, i.e:

"The control strategy is to use the electric machine below a certain vehicle speed and the combustion engine above that. If the energy level stored in the accumulator is lower than a certain value, the electric machine is used as generator and is then driven either by the engine or purely by the kinetic energy of the vehicle. The electric machine is used for braking the vehicle when possible, if the brake torque is not sufficient then the mechanical brakes are used as well. The electric machine is also used for power assist when the desired torque, interpreted from the accelerator pedal position, is larger than the combustion engine can deliver."

3.4.2.2   *Simulation results*

Also for the parallel HDH model, two simulations are presented in this report; the first simulation is the complete WTVC driving cycle and the second simulation is a short simulation using the first 140s of the WTVC driving cycle.

The first simulation is to present the complete simulation over a complete driving cycle. The aim of the simulation is, as mentioned before, to see that the models are numerically stable and produce realistic results. The simulation results are presented in Figure 3.10.
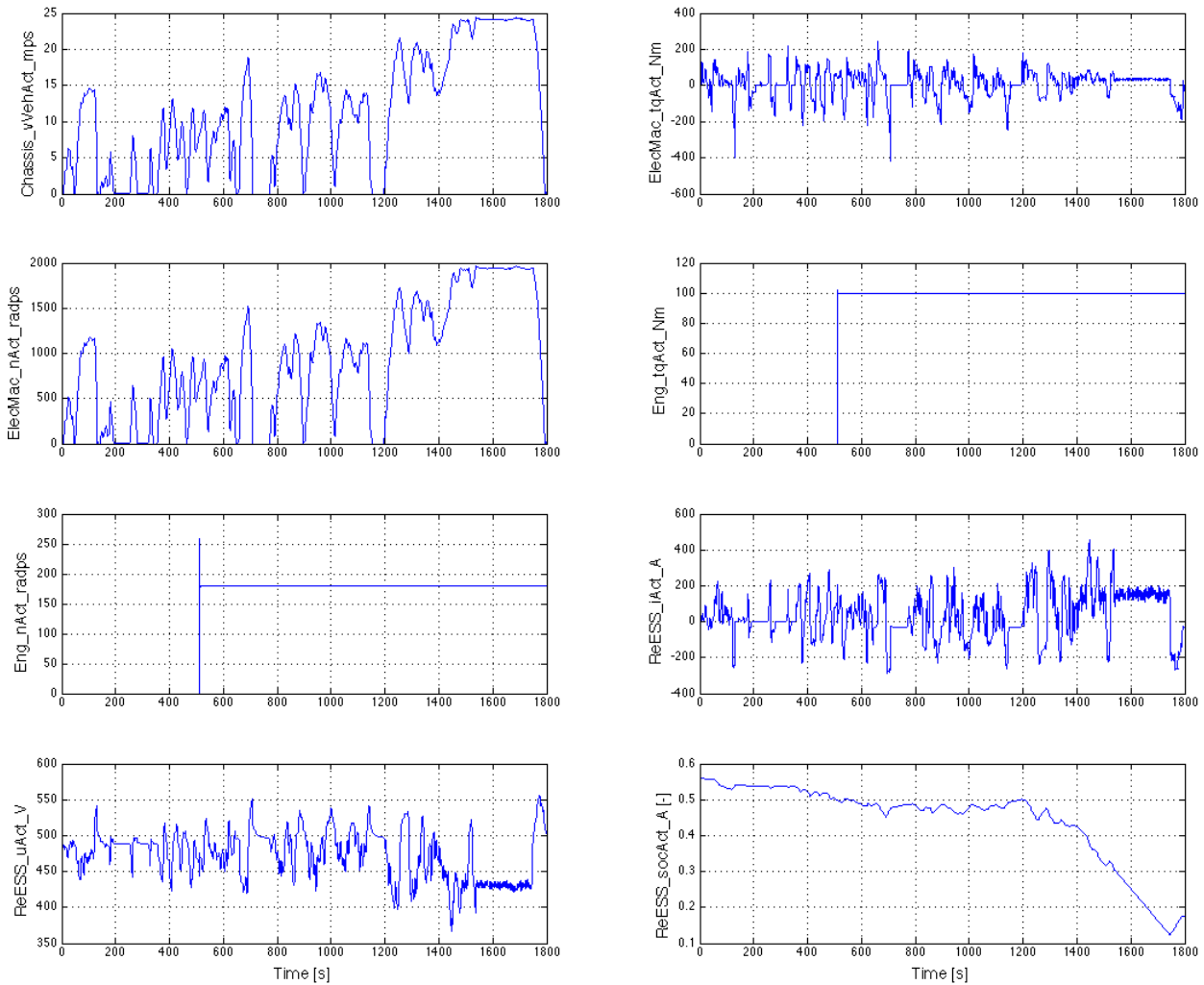
The second simulation can be used for HILS setup approval, similar to Kokujikan No.281, section 8, page 11, Operation Check of HEV Model for Approval. Instead of using the first 120s of the JE05 driving cycle, it is proposed here to use the first 140 seconds of the WTVC driving cycle without slope. The simulation results can be used for comparison with HILS setup simulation results later, to verify that the HILS system numerically produce the same results. The same criterion as proposed in Kokujikan No.281 can be used. The simulation results from this simulation are available in the SimResults folder for the series hybrid vehicle, saved as a MATLAB mat-file. The simulation results are presented in Figure 3.11.
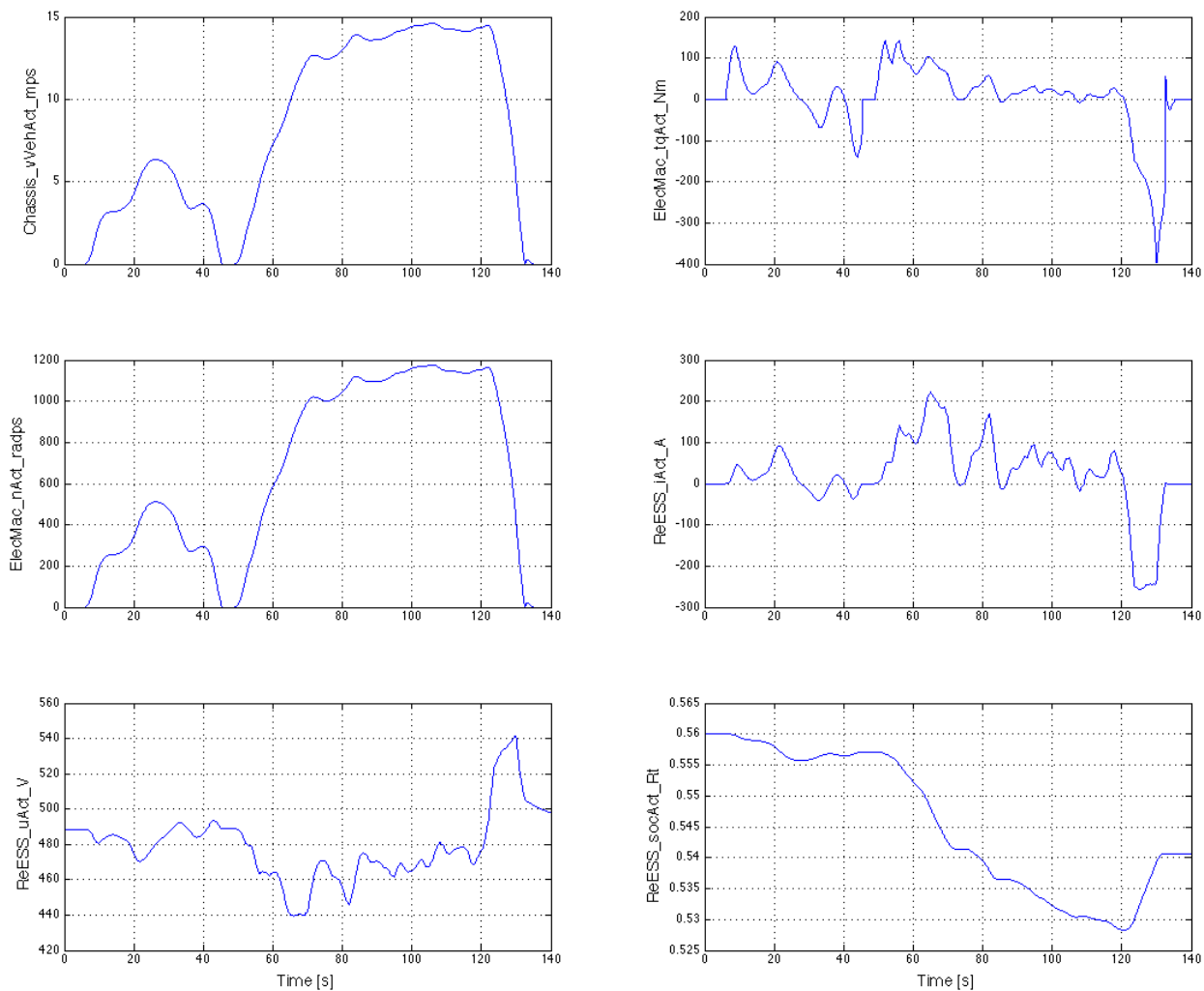
Figure 3.10 Simulation results from simulation of the WTVC driving cycle for the parallel hybrid powertrain.
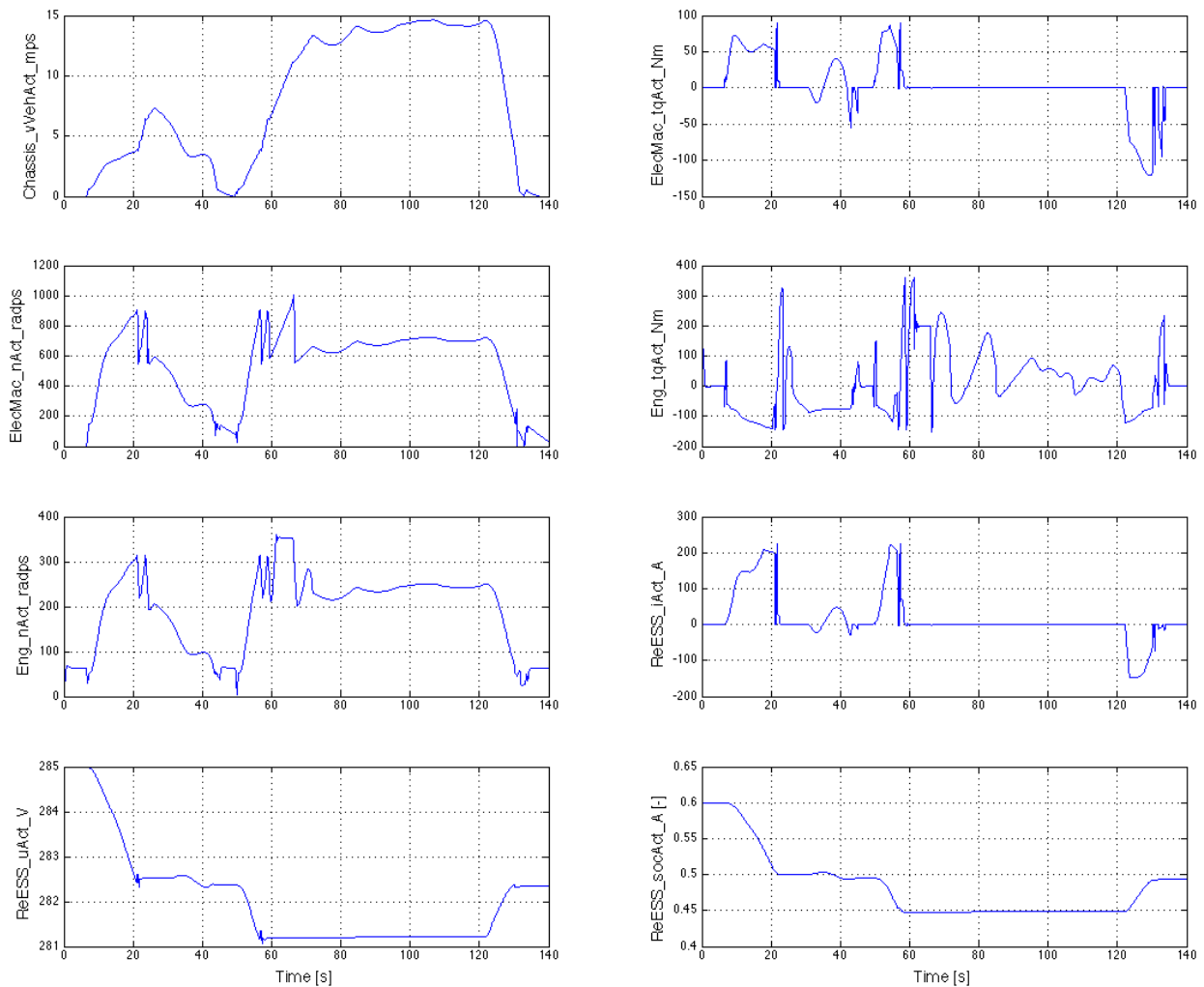
Figure 3.11 Simulation results for the first 140 s of the WTVC driving cycle.

3.5   Summary

A new model library has been developed based on the open-source models presented in Kokujikan No.281 and new models presented in this project or previous project by the same research consortium. The model library offers a data bus structure, component based models and the possibility to construct hybrid powertrain models, both conventional of today like series and parallel hybrid powertrains and more futuristic like flywheel based hybrid powertrains. The models run numerically stable and deliver physically reasonable results, but complete vehicle model validation has not been performed (future work).

4   Task 3 - Report on test procedure and adaptations

Task 3, as defined by the UNECE HDH informal working group, consists of the following activities:

3.1   Report on test procedure and user manual for software
3.2   Provide the interface system for real ECU's
3.3   Adaptations and improvements on the methods for component testing, test cycle definition and simulation method according to demands of Industry and Commission

4.1   Task 3.1 - Report on test procedure and user manual for software

The aim of this task was to review the procedures for component testing, for application of the HILS simulator and for the validation of the HILS set up and additionally describe where necessary in a report as basis for the text of the GTR.

4.1.1   HILS simulator application
In contrast to the Japanese HILS procedure where basically four parallel and one serial HEV topology models are available, the specific HEV model for approval should be created here, based on the component models available in the component library and described in Appendix A, in a way that the actual test vehicle is represented appropriately. Examples for a serial and a parallel vehicle model are available in the library. This opens a wide field of application and enables the usage also for future hybrid topologies. Furthermore it is proposed that the available component models are exchangeable if they don't meet the demands of the respective user. Exchanged models have to fit the modeling guidelines and structure described in section 3.2.

4.1.2   Adaptations and remarks to Kokujikan No.281
In this section the execution of the Kokujikan No.281, which includes the procedures for component testing and the validation of the HILS setup, is reviewed and comments are given on respective sections according to the practical knowledge gained during the validation test programs yet. For a better traceability the numbering of the following subsections corresponds to the numbering in Kokujikan No.281. Sections, paragraphs and items of Kokujikan No.281 not mentioned in section 4.1.2 are considered as valid also for a GTR adoption and are therefore not explicitly mentioned.

4.1.2.1   Chapter 1- HILS system definition

2. Software to be used
According to the new structured HEV models it will not be possible to define the respective HEV model for approval prior to the test. Only the components and the guideline how to use them can be specified. Furthermore there is no need for a fuel efficiency calculation-assisting program since the GTR aims to regulate emissions and not fuel consumption for HEVs at first. The Hermite interpolation program as it was kindly provided by JARI remains valid.

6. HEV model for approval

Since the models have been changed for a GTR adoption this section need adaptation. Appendix A describes the new component models entirely.

7. Reference ECU model for SILS and 8. Operation check of HEV model

The method described for the purpose to check the operability of the HEV model used for the approval is due to the changed models not feasible any longer. In any case this provisions will and cannot be applied. Although this is compatible with the existing Kokujikan provisions this tasks could be rejected. Nevertheless a vehicle model - which could be one of the example models already available in the library - could be used to check if the simulation environment for the type approval test is configured correctly. Model in and output data would have to be specified for that purpose (see OIL/H5).

9. Construction of interface

This provisions are in general valid but due to the established signal bus specific interface signals cannot be prescribed fix. A list of all available signals for the generic models is bundled in Appendix B.

10. Input parameters

10-5 Rolling and air resistance coefficient

The values for rolling and air resistance are calculated by specified equations. For route buses or general buses only the value obtained for the air resistance has to be multiplied by 0.68. The English translation of Kokujikan No.281 states the multiplication of air and rolling resistance by 0.68 which is not correct and caused by a translation error. Nevertheless the obtained values using the specified equations have not been proofed to be representative for a real vehicle behaviour yet. This is a scheduled task for VTP2 and mentioned in the OIL in chapter 5 (see OIL/P2).

10-6 Inertia moment of rotating sections

Pursuant to (1): Settings for inertia moment of rotating sections refers to Kokujikan No.280 which was actually established for conventional vehicles. Since the acting inertias affect the work to be delivered by the propulsion system during the test cycle a mandatory calculation method for the cycle work (considering the inertias) is specified in Kokujikan No.280 for chassis dyno measurements. For HDH vehicles this calculation method is not necessary since Kokujikan No.281 allows to use CAN signals from the vehicles control system to calculate the work delivered.

Pursuant to (2): 7% of the vehicle kerb mass should be set as inertia moment for all rotating sections of the vehicle representing sections from the driven side of the transmission to the tire. This approach is reasonable but the wording should be changed from "may become 7%" to "become 7%". However the value of 7% should be discussed and proofed by random samples of real vehicles and available hybrid vehicles which could be different to conventional ones (see OIL/P3).

10-8 Torsional stiffness and attenuation coefficient

This section can be rejected since these are no model parameters any longer (see Working Paper No. HDH-03-03e [6])

10-9 Engine model response delay block

To represent the transient engine behaviour different torque built-up models are included in the engine models. In the simple model just a time constant has to be specified which means the engine dynamics are independent of rotational speed. This is probably sufficient for natural aspirated engines but unlikely for engines equipped with turbo systems. Therefore a more accurate model represents the transient torque behaviour depending on the current rotational speed. Parameters have to be set to represent the real engine behaviour.

11. Gear change mode
For HDH vehicles equipped with a manual transmission the gearshift provisions of the simultaneously developed European fuel consumption tool for conventional HD vehicles (VECTO) are basically planned to be applied [12]. This was agreed in the 13th HDH meeting in March 2013 and is currently in process (see OIL/H2).

4.1.2.2   *Chapter 2 - Component test procedures*
Basically treating the component test procedures as guideline would be desirable. Available OEM and supplier data should be used as much as possible in order to avoid additional test effort. For an initial HILS model the accuracy of the input data is anyway checked during the HILS verification run. Note that for changes made in the component maps of a validated HILS model, for the certification of a vehicle variant, the input data has to be checked somehow (see OIL/T1 and T2). The modified component library requires adaptations of the described component test methods.

In this section it will be checked whether the described component test procedures fit to the new models in the component library.

3. Test procedure for engine
Since the HILS method for HDH vehicles will be attached to GTR No.4 the measurement procedures for internal combustion engines in the GTR No.4 are supposed to be taken as long as they fit for the HILS method. Definitions from paragraph 1 to 3-5-1, chapter 2, Kokujikan No.281 are supposed to do so. Because the provisions in GTR No.4 only describe the engine mapping procedure for engine full load additional provisions for HDH vehicles have to define the mapping of the entire engine torque characteristics, the engine friction loss, the auxiliary brake and the fuel economy map. Paragraphs 3-5-2, 3-6-2, 3-7-2, chapter 2, Kokujikan No.281 are in principle valid to do so.

A remaining issue is the handling of a cold start requirement for the HILS method. A temperature dependent mapping of the maps mentioned above is due to the high test effort not feasible. It is supposed that there is only a minor impact on the torque characteristics of the engine. For the friction and the fuel consumption map the engine could be mapped at a different than warm condition and the values for different temperatures could be generated by using Hermite interpolation. This is just a first draft idea and has to be discussed in the HDH investigation group first but the question in general is if this is needed at all (see OIL/S2).

4. Test procedure for electric motor

Basically the test procedure for electric machines is considered as reasonable, only two remarks will be outlined.

The current measurement accuracy in paragraph 4-2-(3) must be changed from [m/s] to [%] or absolute current values.

The measurement of the coolant temperature in paragraph 4-4-2-(6) seems to indicate that a kind of pre-condition state before the test should be defined. However, no conditions are specified here and in terms of reproducibility of measurements it may be useful if they are stated like they are stated for example in ECE R85 paragraph 5.3.1.1.

Since there are no restrictions regarding the performance of the electric machine for a cold start at 20°C, the component test procedure for warm conditions should be sufficient.

5. Test procedure for electric storage device

With the extension of the HILS model with a model for thermal behaviour of the electric storage there are two model versions for the electric storage device.

    a) a simple resistor based model

        For this component model there is no thermal model available since the losses covered by the component model and needed for temperature calculations are too inaccurate to describe realistic thermal behaviour. So the component test procedure defined in Kokujikan No.281 is valid for this simple resistor based model.

    b) a more complex RC-circuit based model

        If the thermal behaviour of the electric storage needs to be simulated this more complex model has to be used. This model provides a better and more realistic description of the time dependent current-voltage behaviour through the additional RC-circuit. Thus, allowing a more accurate calculation of the power loss in the electric storage which is needed for temperature calculations.

        The component test procedure is basically the same as defined in Kokujikan No.281, only the data analysis is different. In the following section the differences to Kokujikan No.281 will be described:

        The accuracy of the measurement devices has to be higher, to obtain accurate values for the calculation of losses. Hence the accuracy of the voltmeter shall be better than 0.1 % of the displayed reading and the accuracy of the ammeter shall be better than 0.3 % of the displayed reading. Moreover, the resolution of the voltmeter must be large enough to measure the impressed voltage during the smallest current pulse. The resolution of the thermometer shall be better than 0.1 K to be able to measure a small warming.

        The test sequence shall be performed similar to Kokujikan No.281, chapter 5, Fig. 3, but with altered amperage. The highest charge $\hat{I}_{charge}$ and discharge pulse amplitudes $\hat{I}_{discharge}$ shall be the maximum pulse amplitudes of the in-vehicle use of the storage. The smaller pulses shall be calculated from this maximum values by successively dividing it by a factor of three for three times (e.g. $\hat{I}_{charge}$ = 27A gives a sequence for the charge current pulses of 1, 3, 9 and 27A).

        Chapter 5, paragraph 5-1-5, "Calculation of direct-current internal resistance and open voltage", from Kokujikan No.281 is replaced with the

following procedure:

For each pulse with the pulse current I_{pulse} measure the idle voltage before the pulse (V_{start} in Figure 4.1), and the voltage at 1, 5 and 9 seconds after the pulse has started (V1, V5 and V9).
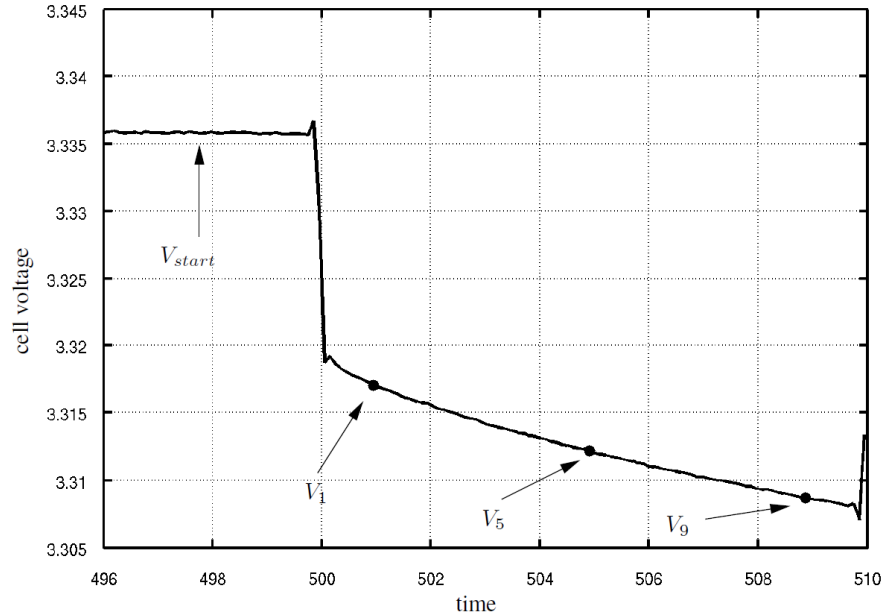
Figure 4.1: Example for a single voltage pulse during a discharge pulse

From this calculate:

$$V_\infty = \frac{V_1 \cdot V_9 - V_5^2}{V_1 - 2 \cdot V_5 + V_9}, \quad \tau = \frac{-4}{\ln(1 - (V_9 - V_5)/(V_\infty - V_5))}$$

additionally for a charge pulse

$$K = -\tau \cdot \ln(1 - \frac{V_1}{V_\infty}), \quad V_0 = V_\infty \cdot (1 - e^{(1-K)/\tau})$$

or a discharge pulse

$$V_0 = \frac{V_1 - V_\infty}{e^{-1/\tau}} + V_\infty$$

Now the values for $R_{0,pulse}$, R_{pulse} and C_{pulse} for a single pulse can be calculated as:

$$R_{0,pulse} = \frac{V_0 - V_{start}}{I_{pulse}}, \quad R_{pulse} = \frac{V_\infty - V_0}{I_{pulse}}, \quad C_{pulse} = \frac{\tau}{R}$$

Taking the mean values for all pulses leads to the desired values for R_0, R and C for the actual state of charge. The measurements shall be repeated for different values of the state of charge according to chapter 5, paragraph 5-1-4, sub item (1).

Since the cold start operating temperatures are not below 20°C the performance of the electric storage is not affected (see chapter 3.1.4). Thus, no component tests at lower temperatures are needed.

4.1.2.3   *Chapter 3 – Procedure for fuel consumption rate*
Since vehicle fuel consumption is not directly determined by the use of the HILS method the provisions in chapter 3 are not relevant for a GTR adoption of the HILS method.

4.1.2.4   *Chapter 4 – Procedure for exhaust emissions*
The following sections should be replaced by the respective existing chapters in GTR No. 4:
- 4. Test engines
- 5. Test fuel
- 6. Measuring devices
- 7. Test Room and Atmospheric Conditions Related to Test
- 9. Test Procedure for Exhaust Emissions from Heavy-Duty Hybrid Electric Vehicles
  with minor changes to:
    o 9-3-1 Time correction of engine revolution speed and shaft torque
      It has to be added that for the HILS method a time correction for the actual measured data of engine revolution speed and torque can be performed in relation to the reference data of the exhaust gas measurement cycle obtained according to paragraph 8-1-3
    o 9-3-2 Calculation of integrated engine shaft output, etc.
      Definitions could be amended by the formulas to calculate the integrated engine shaft output during measurement driving as well as the integrated reference engine output during the exhaust gas measurement cycle
- 10. Measurement of Emission Mass of CO, CO_2 and so on as well as PM

3. Test Method for Exhaust Emissions from Heavy-Duty Hybrid Electric Motor Vehicles

The JE05-mode test cycle has to be replaced by the respective test cycle that is used for the GTR and still needs to be defined in validation test program 2 (VTP2).

8. Creation of Exhaust Gas Measurement Cycle

8-1-1 Operation check of HILS system

The operation check of the HILS system by means of a SILS reference ECU and reference parameters cannot be applied due to the new, more flexible model structure where there is no basic hybrid model with defined output values available for comparison (also see 4.1.2.1, sub item 7). Nevertheless it is possible to define a new dataset for operation check purposes (see OIL/H5).

8-1-2 Construction of HILS system and verification of compatibility

This subchapter should just refer to the provisions of chapter 1 (see 4.1.2.1) where the guidelines for the construction of the HILS system according to the layout of the vehicle to be tested are explained. Chapter 1 should also explain that the HILS system consists out of actual ECU, driver model, unique interface model and vehicle model with input parameters and maps according to the provisions in chapter 2 (see 4.1.2.2). In accordance with chapter 5 (see 4.1.2.5) the correct operation and accuracy of the HILS model should be confirmed.

8-1-3 Calculation of exhaust gas measurement cycle by means of HILS system simulated running

The simulated running of the HILS system should be performed using the respective test cycle defined under sub item 3 of chapter 4 (see 4.1.2.4).

The effect of the command frequency used for the operation set points (engine revolution speed and torque) on the engine test bench on resulting engine emissions is still under investigation in VTP2 and may be defined with a higher value than 1Hz based on the outcome of the investigations (10Hz recommended). According to GTR No. 4 at least 5Hz have to be used for test bench command values, 10Hz are also recommended here.

If the data during the gear change period, i.e. the drop of torque due to clutch disengagement, may be replaced by the values before the gear shift event is also still under investigation. This provision was once established in Kokujikan No.281 because concerns raised that a high transient speed and torque operation could not be covered by the engine test bed and mainly because conventional vehicles are allowed to do the data manipulation due to the functioning of the conversion program for vehicle speed to ICE speed/torque for conv. HD vehicles. Based on the outcome of VTP2 this provisions shall be adapted (see OIL/H3 and H4).

The allowable errors in speed and time during the simulated running of the HILS model can be valid also for the adaption of the GTR.

8-1-4 Range of electricity balance for HILS system simulated running

Basically this subchapter shall stay valid. The initial state of charge is adjusted by limiting the ratio of the energy conversion value of the electricity balance to the integrated shaft output of the engine. Both values are obtained by simulated running of the HILS system.

The value of the fixed limit for comparison depends on the outcome of VTP2.

There seems to be an error in the units used in the calculation formula. The energy conversion value of the electricity balance ΔE is stated in kWh but the formula results in Wh by multiplying Ah and V.

Also a definition of the integrated shaft output of the engine is missing. It has to be defined somewhere in this chapter simply by adapting the formula given in chapter 9-3-2, Kokujikan No.281.

8-2 Replacement of test torque value at time of motoring

This paragraph shall be valid. For hybrid vehicle testing, different than for conventional engines in GTR No. 4 paragraph 7.4.7 sub item (b), only the values from the engine friction torque curve shall be used if the torque of exhaust gas measurement cycle obtained according to paragraph 8-1-3 becomes negative.

11. Calculation of Integrated System Shaft Output

The usage and hence the underlying definition of the integrated shaft output of the hybrid system for calculating the emission mass of exhaust gas per unit work done in the test cycle has been discussed in the GRPE Informal HDH Working Group. But it still has to be decided what should be the reference value for calculating specific emissions. This is still an open issue and marked in the OIL in chapter 5 (see OIL/C5).

Basically, there are two options for the reference value for calculating specific emissions (i.e. emissions per unit work done):

a) Refer to the total delivered work needed for vehicle propulsion in the test cycle.
    In this case the reference value would be the integrated system shaft power (i.e. the sum of power delivered by combustion engine and electric motor) according to Kokujikan No.281.
    From a complete vehicle point of view, this approach would limit the emissions per unit vehicle propulsion work done to the same level as for conventional vehicles. That means a hybrid powertrain is allowed to generate as much emissions per unit vehicle propulsion work done as a conventional combustion engine. From an engine point of view, this approach would allow a combustion engine in a hybrid powertrain to generate more emissions per unit engine work done than a conventional combustion engine due to the recuperation of work performed by a HDH.
    For this approach also a method for taking into account the deviations between simulated reference work and actual measured work of the combustion engine, similar to the method proposed in Kokujikan No.281, has to be defined.
b) Refer to the delivered work of the combustion engine in the test cycle.
    In this case the reference value would be the integrated combustion engine power measured in the emission test run on the engine test bench.
    From an engine point of view, this approach would limit the emissions of a combustion engine in a hybrid powertrain to the same level of emissions per unit engine work done as a conventional combustion engine. From a complete vehicle point of view, this approach could limit the emissions per unit vehicle propulsion work done to a lower level as for conventional vehicles as it would not account for possible additional EM work available due to brake energy recuperation.

4.1.2.5   *Chapter 5 – HILS verification test procedure*
           It was agreed that the model validations in VTP2 will be performed according to the provisions in Kokujikan No.281 as long as they are valid for the new proposed model structure. The main purpose is to gain knowledge and identify gaps for a GTR implementation. Since the model validations have not been finished yet it cannot be reported fully but expertise so far which was gathered during tests and

discussions with our Japanese colleagues, with very special thanks to Mr. Nobuya Osaki, will be outlined. The following numbering does not correspond to the numbering of chapter 5, Kokujikan No.281.

1. ECU and multi-ECUs
In chapter 1 of Kokujikan No.281 the "hybrid ECU" is defined as one part of the HILS system to be tested. For the addendum to the GTR a definition which control units of the real vehicle have to be used in the HILS system is necessary. A Japanese study (referenced in [7]) concerning the usage of multiple ECUs in the HILS procedure concludes that some functionalities will have to be included as software part in the OEM-specific, unique interface model in order to minimize the effort for the certification process. In this case just the control units with major hybrid control functionalities would be included as hardware parts. Therefore a definition of certain functionalities instead of hardware parts that have to be included seems to be a viable approach since architecture of control units and distribution of hybrid operation strategy will be very OEM-specific. If some ECU algorithm is included as software part in the OEM-specific, unique interface model the interface model is a crucial part in the verification process where real life vehicle operation is compared to the simulated operation of the HILS model. Therefore changes in the OEM-specific, unique interface model affecting hybrid control has to lead to a mandatory repetition of the model verification (see OIL/V2).

2. Data measurement for HILS verification
In the practical application of the original Japanese HILS regulation Kokujikan No.281 there are several additional definitions, clarifications and amendments necessary which are available in additional Japanese documents but are not available in an English version. This subsection lists these additional topics in the verification process identified so far in discussion with the Japanese representatives in the GRPE HILS informal working group. For the amendment of the GTR these topics need to be included to describe the HILS verification procedure properly.

Torque values in HILS verification

In order to get the actual measured torque values for comparison with the simulated values from the HILS model according to chapter 5 in Kokujikan No.281 there are two different methods used depending on the test procedure for the vehicle measurement testing. In both methods torque values are calculated at least partially out of the respective stationary torque map for the component obtained according to the component test procedures in chapter 2 of Kokujikan No.281 by the use of torque command values recorded from the CAN bus.
  a) System bench test according to Kokujikan No.281, chapter 5, paragraph 4-1, sub item (1)
"System bench test" is defined in the Kokujikan No.281 as testing the hybrid system consisting of combustion engine, electric motor and energy storage and their control unit but without the transmission. If the electric motor is integrated to the transmission the system to be tested has to be run in a fixed gear and gear shifting is not allowed during the test.
In case of a powertrain test bench it is easy to measure the total powertrain torque since the powertrain is mechanically connected to the dynamometer. The electric motor torque is calculated by using torque command values from the CAN bus out of the electric motor torque characteristic map according to the component test procedure (ref. HDH-03-03). With the measured rotational speed of the electric

motor and the torque command values from the CAN bus as input the delivered torque is calculated out of the electric motor torque characteristic map via an interpolation procedure. The interpolation is done using the Japanese Hermite interpolation program listed as part of the HILS system in Kokujikan No.281.

The combustion engine torque is then calculated as the difference between the total powertrain torque and the electric motor torque taking into account transmission efficiencies and transmission ratios between combustion engine, electric motor and test bench dynamometer. The use of CAN data for the "System bench test" may not be suitable for inclusion in the GTR.

b) Chassis dynamometer test according to Kokujikan No.281, chapter 5, paragraph 4-1, sub item (2)

Since accurate measurement of the torque delivered at the wheel hubs, respectively at the hybrid system output shaft is difficult on the chassis dynamometer the delivered torque of both the combustion engine and the electric motor are calculated out of the torque characteristic map for the component as described in sub item a). In case of the combustion engine measured rotational speed and torque command value recorded over CAN (e.g. throttle valve opening angle, fuel injection amount, target torque in %) are used as inputs for the interpolation program. In case of the electric motor measured rotational speed and torque command value are used.

With this method the recorded time sequential data of torque command values is converted into time sequential data of delivered torque values. The delivered torque values are the reference data used for comparison to the simulation output.

For this interpolation of delivered torque values via the stationary component map the torque command signal used as input has to be chosen in a way that the dynamic characteristic of the component is represented best (e.g. fuel injection amount for combustion engines).

Electric storage current, voltage and power values in HILS verification

The time sequential data for current and voltage of the electric storage can be obtained by actual measurement or recording CAN bus values according to Kokujikan No.281.

The time sequential data of the electric storage current and voltage are then used to calculate the electric storage charging and discharging power over time by multiplication of current and voltage.

These reference values for electric storage power are then directly compared to the respective values from the simulation output.

General issues in HILS verification

The calculation of the reference data for comparison by interpolation via the stationary component map with a CAN bus command signal used as input, as explained in chapter 4.1.2.5 under sub item 3, is done for the entire recorded data of the complete test cycle. This reference data from the actual measurements is then used for both methods of comparison listed in Kokujikan No.281, chapter 5, paragraph 5-2 – the one heap method which compares data only for a first short part of the test cycle as well as the entire cycle method which compares data for the overall test cycle.

The term "output" in Kokujikan No.281, chapter 5 could be misinterpreted especially in the context of chapter 5. In other chapters it is defined by formulas or in the text that output means the delivered power by the respective component and negative values are not considered in the calculations. Whereas for the values calculated for the comparison with the validation criteria in chapter 5 – except where explicitly defined differently – not only delivered power but also absorbed power is considered. For the amendment of the GTR the term "output" should therefore be exchanged for "power" or something similar which is a neutral wording and considers energy flow from and to the component.

In Kokujikan No.281, chapter 5, paragraph 6-1, Table 1 and paragraph 6-2-1, Table 2 one of the validation criteria is defined as "vehicle speed or engine revolution speed". In the application of the Japanese regulation the criterion is selected according to the vehicle measurement procedure used for model verification.

a) If the system bench test is used the selected criterion should be the rotational speed of the hybrid component that is connected to the dynamometer. This has not necessarily to be the combustion engine. Depending on the hybrid powertrain layout the rotational speed of the combustion engine and the rotational speed of the vehicle propulsion component are not necessarily linked together. In this case the term "engine revolution speed" has to be exchanged for the amendment of the GTR and should define the rotational speed of the driving part of the hybrid system.

b) If the chassis dynamometer test is used the selected criterion should be the vehicle speed. The Kokujikan regulation considers vehicle speed representative of the combustion engine rotational speed and allows to choose either of them. But depending on the hybrid powertrain layout the rotational speed of the combustion engine and the vehicle speed are not necessarily linked together. For the amendment of the GTR the definition should be that vehicle speed should be used as validation criterion in combination with chassis dynamometer tests.

3. HILS verification run

Additional information to the description of the driver model in chapter 1, 5. Driver model, Kokujikan No.281 should be provided with regard to the verification process.

The only purpose of the driver model is to track the reference vehicle speed from the chassis dyno test in the simulation. It is regardless how this is ensured. Either a PID or similar controller is used to do so or time history CAN data (e.g. accelerator and brake pedal positions) is used. This description is valid for the "entire cycle" test run where allowed time history data for automatic controlled transmission vehicles means gas pedal and brake pedal and for manual transmission vehicles gas and brake pedal and shift position are meant. For the "one heap" validation you have to use the same signals in the simulation as they occurred during the chassis dyno test on the CAN bus, e.g. gas and brake pedal position signals for automatic controlled transmissions or gas and brake pedal position and shift signals for manual controlled transmissions.

Additionally, the description of the driver model in Kokujikan No.281 is for manual controlled transmission vehicles only. In case of an automatic controlled transmission the accurate description would be: "The driver model makes the HEV model for approval to operate in such a way as to achieve the reference vehicle speed by generating accelerator and brake signals, and is actuated by the PID

control, etc. In addition, the driver model may be replaced by dot-sequential data of accelerator and brake." Shift signals are rejected from the original text.

4. SOC balance

This section should additionally describe how the achievement of a balanced SOC for a HILS exhaust gas emission run is described in the Japanese regulation and how it is handled practically during certification and verification.

Regarding chapter 4, paragraph 8-1-4, Kokujikan No.281 dealing with the test procedure for exhaust emission of HDH vehicles it is mandatory to achieve a balanced SOC during the HILS simulation run which generates the ICE operation pattern to be tested on the engine test bed in terms of emissions. This is of course reasonable but to be able to run the HILS model for a certification a model verification is needed first. The model verifications only purpose is to achieve the same vehicle behaviour as it was measured on the chassis dyno. For the test runs on the chassis dyno there is basically no restriction for a balanced SOC during the test runs but there is a verification criteria defined in paragraph 6-2-3, chapter 5, Kokujikan No.281 which specifies the range of electricity balance. Practically speaking this defines the allowable SOC tolerances between the chassis dyno test run and the HILS simulated run for the model verification. In order to fulfill that criteria easily it is useful to have a mostly balanced SOC during the chassis dyno test as well. To achieve this constraint a typical workflow was presented by Japanese experts:

- Chassis dyno/powertrain test run with arbitrary start SOC (to avoid multiple test runs for finding the correct start SOC to achieve a balanced SOC)
- HILS model verification with same conditions
- HILS model test runs to find a start SOC where the SOC is balanced over the entire test run
- Chassis dyno/powertrain run with identified start SOC from HILS model tests
- Model re-verification in order to fulfill the criteria for the range of electricity balance

For the calculation of the energy conversion value in paragraph 6-2-3, chapter 5 the formulas specified in paragraph 8-1-4, chapter 4 are used.

5. Re-certification / Re-Verification

Derived from the current Japanese regulation several questions raised internally and during the OEM meetings regarding the need of a re-certification/re-verification of a vehicle/vehicle model. This section should summarize the insights so far.

A model (re)-verification is necessary if

- The HILS system is used the first time
- The hybrid system layout of a verified HILS model is changed even though the same components are used
- Changes are made on the component models (e.g. structural change, increase of input parameters,…)
- The application of components changes (e.g. transmission is set from automatic to manual)
- Delay time or time constants of engine or electric motor models are changed
- Cases of other reasons appear

Cases of other reasons protects against the occurrence of unexpected failure of a HILS accuracy verification due to a free change of any specification where an example cannot be shown at this time. So it is placed as a final guard. Basically each change which affects the HILS verification result forces a new model verification but changing:

- Engine torque characteristics
- Electric motor torque, electricity characteristic
- Battery internal resistance, voltage characteristic
- Vehicle specifications except changing GVW cross over 12ton (this has to be discussed for a GTR adoption; see OIL/V2)

is allowed and does not request a new model verification.

However a new certification run may be required. For reasons of a re-certification please refer to section 4.3.2.4.

ECU software updates do not in principle force a new HILS certification as long as the update has no effect on the HILS verification results. If there is an impact strictly speaking there would be also a need of a new model verification which would result in a high test effort and should be avoided. This is also valid for changes necessary in the interface model and has to be discussed in the HDH investigating group.

6. Test cycle definition

The WVTC is like the WHTC exactly defined form second 1 to second 1800. This does not specify the conditions at second 0 when the measurement starts. It is a slightly trivial matter but has effect on the calculation of the cycle work in terms of comparability of data sets. An equation for the appropriate calculation of the cycle work should therefore be specified at the definitions of the resulting HDH test cycle. This is at earliest possible when a final test cycle is agreed in the HDH group.

7. Vehicle test weight

In the current Japanese regulation the test vehicle weight of a truck is equal to its kerb weight +1/2*max. payload + 55kg (driver) and that of bus is equal to its kerb weight + riding capacity * 55kg/2 + 55kg (driver). However, the test weight can also be derived from vehicle class specifications where the powertrain should be used (example see tables at 4.3.2.4). The selection depends on the type of certification, either vehicle specific or vehicle independent and is defining the reasons for a re-certification of the powertrain. Basically the kerb weight + half payload/riding capacity approach seems reasonable and is also basis of the Japanese vehicle class table but could be discussed in the work group if demands from stakeholders arise. Current on-going investigations regarding a connection of powertrain rated power and vehicle test weight are also on that basis. Defining the test weight is highly related to the way how the certification is done (vehicle dependent/independent) and on the final definition of the drive cycle. Investigations in that perspective are still on-going.

8. Calculation of engine cycle work

For the entire test cycle there are verification criteria for positive engine work and fuel economy. The omission of the data during gear change (see above point 2) is not allowed when calculating the cycle work nor when calculating the fuel efficiency.

4.1.3   *Summary*

In general Kokujikan No.281 is considered as a good baseline for the GTR drafting. Due to some links to the Japanese conventional vehicle testing procedure and the applicability on either a vehicle specific test or a vehicle independent test method and the resulting needs of a recertification of similar propulsion systems (please see 4.3.2.4 for a comprehensive perspective), some sections are suggested to be adapted for a GTR. Identified issues and suggestions are reported in the text above for each issue. Paragraphs not mentioned are considered as valid. Since the final test method for the GTR (need of re-certification of powertrains, family concept, valid test methods,..) is not decided yet, proposing only one desired solution for each issue is not possible. It is suggested that for a final GTR all provisions based on Kokujikan No.281 should be in line with the post-transm. powertrain test procedure proposed by the US EPA. This addresses especially the vehicle dependent/independent test method and the resulting needs of a recertification which are also not yet completely defined in the EPA proposal.

4.2   Task 3.2 - Provide the interface system for real ECU's

This task was intended to cover the preparation work on the interface system between the simulation model and the hardware ECU to provide signal ports including information on specific units.

The interface system/model itself is an OEM specific MATLAB Simulink software part. In this interface model level tuning of signals, fail release correspondence, generation of signals that are not provided by the simulation model but needed for the actual hardware ECU, conversion of signals etc. can be handled.
The new basic structure of the HILS model and the interface between hard- and software is shown in Figure 4.2. (in- and output interface model - yellow, vehicle model - light blue, driver model - purple, hardware ECU - grey)
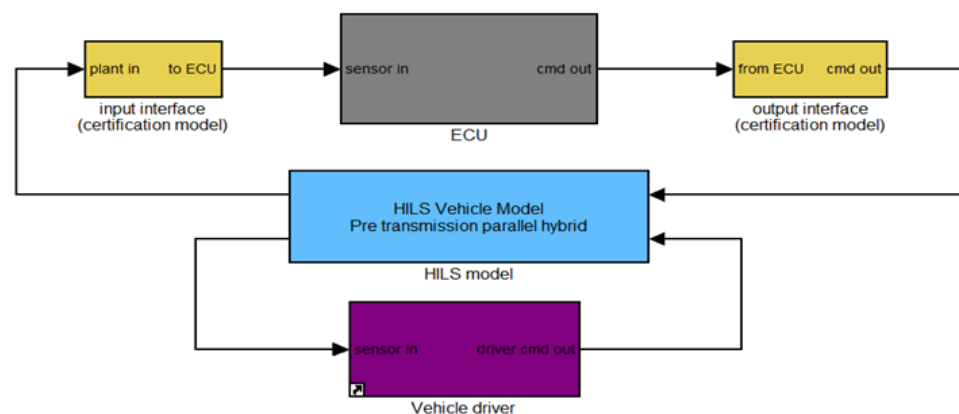
Figure 4.2: schematic of HILS setup

For this software interface a list of signals should be defined and the properties of each signal should be described properly (e.g. model component affiliation, signal name, clear characterization, unit etc.) based on the signal list given in Chapter 9 of the Japanese regulation Kokujikan No.281 [1]. Thinking about multiple ECUs on a test rig and about the variety of manufacturers a standardized interface signal list meeting the demands of all manufacturers seems rather unlikely to be

implemented. Due to the restructuring of the models where a flexible signal bus was established a standardized interface signal list is even no longer required. The flexible system bus allows the user to route each signal which is provided by the HILS model to the interface model. Missing signals can be easily added to this bus. A list of existing signals for each single component of the component library is available in Appendix B - Interface signals.

4.3   Task 3.3 - Adaptations and improvements

For eventual adaptation and improvement of methods suggested by the HDH group in the course of the project, two weeks of work was reserved and by far consumed.

4.3.1   *Development of a HDH test cycle*
In the previous project phase it was indicated that there would be two different types of certifying HD vehicles when a HILS method for HDH is introduced based on the Japanese legislation. Thus, there is a need to make both methods comparable.
The Japanese HILS approach – as vehicle based approach – is based on a speed cycle over time. The resulting engine load cycle will depend on the vehicle parameters when a vehicle speed cycle is used as input. This may lead to engine operating points with no full load operation which is not representative for real world driving of a vehicle. However, the emission test cycles for conventional engines are defined as engine speed and torque over time and lead to engine operating points which cover the relevant areas of the engine map from part load to full load. As a result, emissions measured for conventional HD and for HDH might not be comparable. Figure 4.3 shows a comparison of the resulting operating points in the engine map for a vehicle based speed cycle (left) and an engine cycle (right).
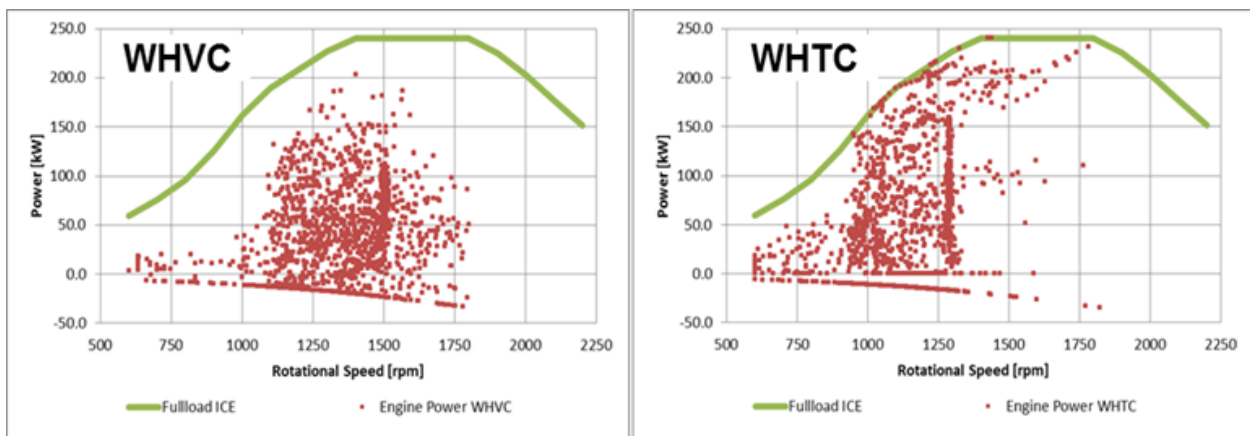
Figure 4.3: Comparison of engine load points for a conventional HD vehicle (14 ton / 240 kW) in a vehicle based speed cycle (left) and engine cycle (right)

In order to make both methods comparable a test cycle (called WHDHC) for HDH was developed based on the WHTC in the previous project phase which leads to similar load points for hybrid powertrains as the WHTC for conventional engines [2]. To run a HILS model with such a test cycle the driver model described in section 2.2 had to be developed. Although the SILS model test runs for a serial hybrid model based on the Japanese structure were positive several problems related to the test cycle occurred and should be reported in detail here.

4.3.1.1   *WHDHC derived from basic WHTC*

This chapter describes how the reference cycle is generated that consists of representative propulsion power demand for a similar conventional vehicle.

The WHTC for a conventional vehicle is a normalized engine test cycle consisting of torque and rotational speed over time. This test cycle was very basically derived from a speed cycle called WTVC [8]. To denormalize the WHTC for a specific engine the full load torque curve of the engine and characteristic engine speeds are used [9]. As the WHTC is an engine cycle it only consists of negative torques down to the motoring curve of the engine. But if a WHTC based test cycle should be used for HDHs sections of engine motoring have to be enriched with respective negative power (i.e. mechanical braking of the vehicle) in order to allow the HDH to recuperate energy. Therefore the equations of vehicle longitudinal dynamics were used to calculate the power at the wheel hub. Thinking about different hybrid drivetrain topologies the wheel hub was chosen as the most common reference point for considerations of the propulsion power. Therefore the positive WHTC power had to be reduced by a simplified differential gear (0.95) and gearbox (0.95) efficiency chosen with respect to Kokujikan No.281 [1]. During sections of deceleration at the WTVC the respective negative power is calculated and used to replace the corresponding WHTC-power at the same time (for additional information see [2]).

As a result you get a power time curve which is identical to the WHTC on the positive side and representative for the amount of available recuperation energy on the negative side.

This test cycle should be used to test hybrid systems either on an engine test bed (powertrain test) or using the HILS method by following the test cycle and derive the specific ICE emission test cycle with a HILS model.

Powertrain test: For the positive side of the WHDHC power time curve it would be possible to depict the rotational speed and torque time curves derived from the WHTC (even though this is not really suitable for HDH full load curves, see 4.3.1.3). Since gearshift events are already included in the power time curve this would be suitable for pre-transmission powertrain tests (without a gearbox, identical to the engine test). But for the negative side, only the power time curve derived from the vehicle dynamics is available. Specific torque and rotational speed are not derivable nor are gearshift maneuvers included. A generic tire radius, a generic final drive ratio and a generic gearbox including the gearshift strategy would have been needed to generate a fully valid test cycle consisting of rotational speed and torque which could be run on a pre-transmission powertrain test bed. Beside that also a redevelopment of the WHTC denormalization method for hybrid powertrains would have been needed. So the WHDHC test cycle using a pre-transmission powertrain test was rejected and the focus was laid on running the HILS method and using the WHDHC test cycle with stipulated power demand.

Since the Japanese HILS approach is a vehicle based test methodology using a speed cycle as input the rotational speed at the wheels is already defined implicitly. Speed (from WTVC) and power (from WHDHC) defines the load point at the wheel hub entirely.

The so generated test cycle consists of a vehicle speed time curve and simultaneously defines the power to be delivered by the powertrain, which would have been usually derived from the vehicle parameters (vehicle mass, rolling and drag resistance). Depending on the specific test vehicle it will occur that there is a deviation between the power demand of the vehicle to run a certain speed on flat

WTVC and the power, which is demanded by the new generated test cycle. This power gap can be closed by adapting additional loads to the system. Road gradients have been representatively chosen to increase/decrease the road load to the vehicle. Also headwind or a varying vehicle mass could have been chosen but a constantly varying mass during vehicle operation was supposed to force problems in the vehicles software. In the end it does not matter since all actions are only intended to regulate the road load on the vehicle and therefore define how much work has to be delivered by the vehicles propulsion system during the entire test run. However, head winds have been rejected in the HDH group during discussions and because of their applicability and conceivability, road gradients have been chosen as most likely implementation for road load correction. When the balanced altitude is discussed in the following section one has to keep in mind that the road gradient is only a tool to add road loads and consider that there is no physical need of balancing the altitude even if a positive road loads/gradient is applied.

4.3.1.2 *Representative road gradients to adapt road loads*

The basic idea was to adapt the road gradients in a way that the resulting powertrain power output exactly meets the power demand of the WHDHC test cycle. Closer feasibility investigations have been made which resulted in a rejection of that approach. The background will be declared in this section and a promising feasible solution will be explicated.

In order to be able to adjust the power output of the hybrid powertrain to the power demand of the test cycle second by second there is the need of every second changing road loads/road gradients. This is in general possible and even no problem for constant driving conditions but as soon as there occur abrupt changes in the power time curves very high load changes/road gradients can occur. Gear shift events which are included in the WHTC test cycle to represent the gear shift behaviour for a conventional engine have been identified as such sections. Figure 4.4 illustrates the power pattern during three gearshift events at the WHTC. In case of parallel hybrids the power demand drop down during a gear shift event in the test cycle would probably force the hybrid logic to also change gears. However it will for sure not be representative for serial hybrids or vehicle concepts without a gearbox.
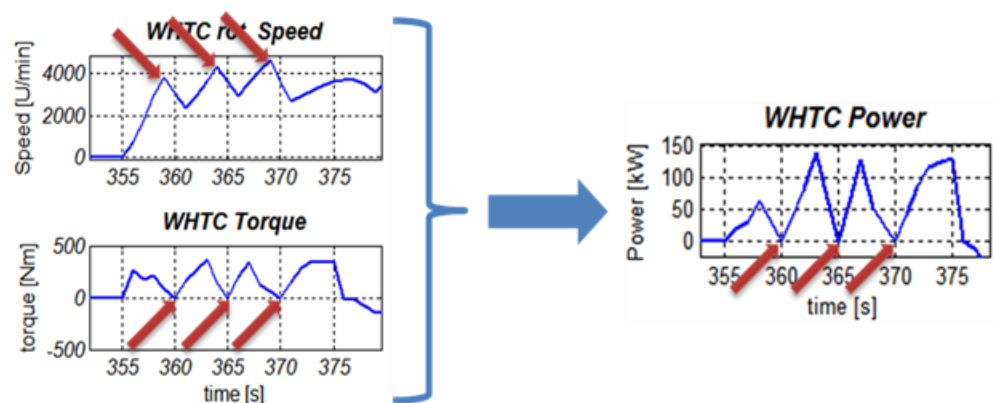
Figure 4.4: Gear shift events and corresponding power pattern in the WHTC

The right chart in Figure 4.5 illustrates a representative propulsion power demand for a vehicle without a gearbox (green line) during acceleration from zero to 40 km/h at the WTVC. To force the vehicles propulsion system to deliver the same power

pattern as the WHTC (blue line) while keeping the desired WTVC speed you will need to adapt highly fluctuating road loads/gradients which are calculated from the power difference between the blue and green curve. In terms of points with zero power for one curve the respective road gradient will per definition get infinitely or at least very steep if the power value is not exactly zero. Values up to 40% and above occurred during the investigations.
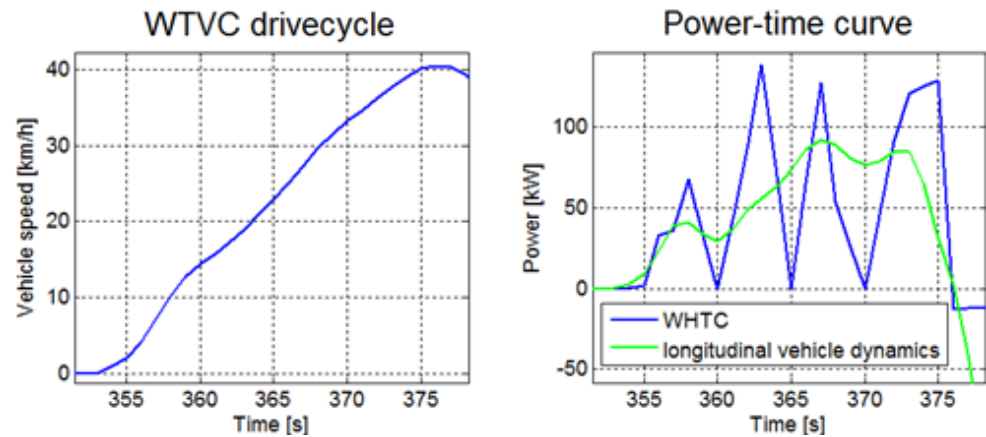
Figure 4.5: propulsion power demand with and without gear shift interruptions

Beside that there are also sections of clutch actuation where vehicle speed is still zero but the engine of a conventional vehicle already delivers power to the system (see Figure 4.6). These sections also cause high road gradients because delivering a certain power at zero speed would per definition give an infinite traction torque which would lead to an infinite road load/gradient.
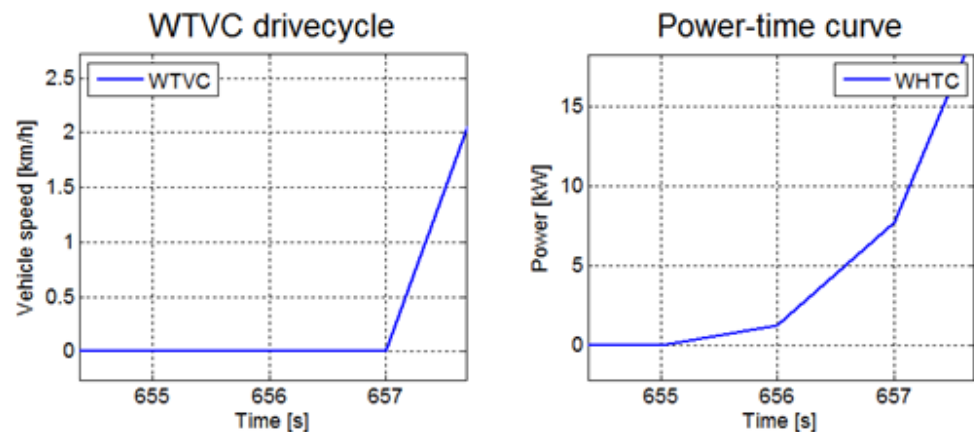
Figure 4.6: Sections of clutch actuation from second 655 to 657

Both effects lead to problems when a WHDHC test cycle is applied with a real HILS model and with an actual hardware ECU. It was shown by Japanese colleagues at JASIC that a high fluctuating road gradient pattern with high absolute values caused an ECU error during a HILS simulation run [3]. One approach which lowers the absolute values of road gradients was to smooth the WHTC power pattern. Although there was only a minor impact on the overall cycle work the modification slightly affected the load point distribution during the cycle. The lower chart of

Figure 4.7 indicates that certain full load points are removed by this smoothing method and the WHTC power is therefore not tracked accurately any more. A slightly different method to remove the gear shift events from the test cycle was presented by JASIC [3] where the gaps with zero power were filled by replacing the data before and after the gear shift event. This of course increases the overall cycle work but also lowers the resulting road gradients.
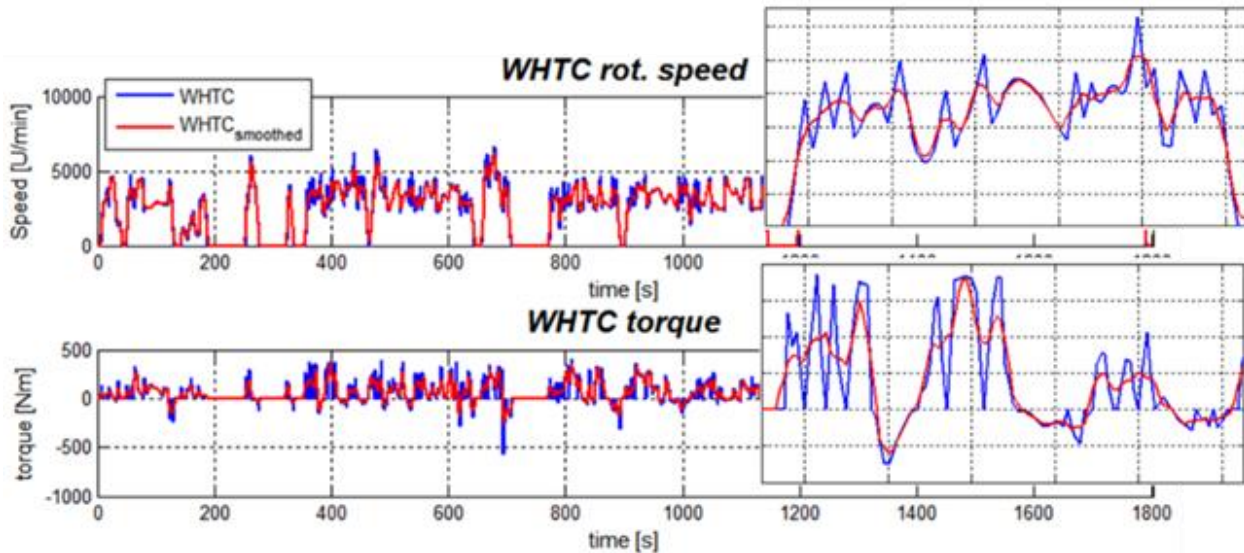
Figure 4.7: Example for smoothing of torque and speed

Regardless of the method used for removing the gear shift events, the sections of clutch actuation still remained problematic. The JASIC developed method to smooth the resulting road gradient pattern with a thirty second moving average mean value [13] is a very practical solution to avoid the appearance of unrealistically high road gradients but of course changes the propulsion power demand in a way that the WHTC power pattern is also not tracked accurately any longer and additionally is by design not able to match the overall WHTC cycle work at the end of the test run.

Considering the effort, complexity and number of modifications needed for adapting road loads/gradients to a second by second comparison of power time curves it was decided to shift the focus on an integral approach where the vehicle should be operated in way that it tracks the corresponding WHTC cycle work[1]. A first approach dealt with the application of one constant slope for the whole test cycle to get identical positive cycle work at test end. Even though this works it could be shown that the behaviour in the time history plot of the work between WHTC and WHVC with constant road gradient is too different. To adapt the behaviour of the work time curve the WTVC test cycle was divided into 12 sub sections called mini cycles (highlighted in Figure 4.8). Sub section 4 and 6 can be ignored since they only contain minor speed heaps lower than 1 km/h for just a few seconds. Dividing the test cycle in 12 sub cycles is a reasonable approach since the WTVC was developed using different representative vehicles for different sections of the test cycle (for additional information see [9]).

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[1] Recommendation: It should be proofed if a conventional HD vehicle gives similar results in terms of emissions when the resulting ICE test cycle derived from a WTVC with road gradients is compared with the corresponding WHTC (see simulation study in 4.3.1.4).
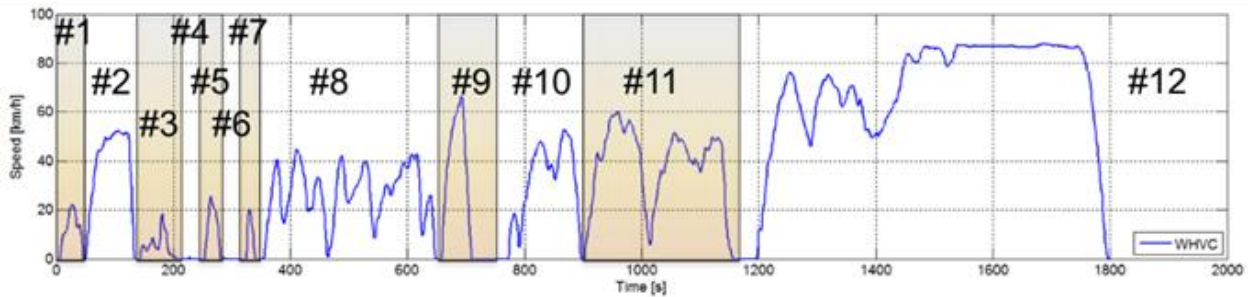
Figure 4.8: WTVC (WHVC) speed profile divided in 12 sub cycles

For each sub section the corresponding positive WHTC cycle work is calculated and compared to the positive cycle work of the test motor vehicle running at the respective WTVC vehicle speeds. The negative cycle work results from the speed profile and the respective vehicle data. To adapt the positive cycle work for each mini cycle also road gradients are used. An average road gradient for each mini cycle is calculated out of the difference between positive WHTC work and positive traction work of the vehicle at the respective mini cycle. Figure 4.9 gives an example of positive WHTC and WTVC cycle work and resulting road gradients.
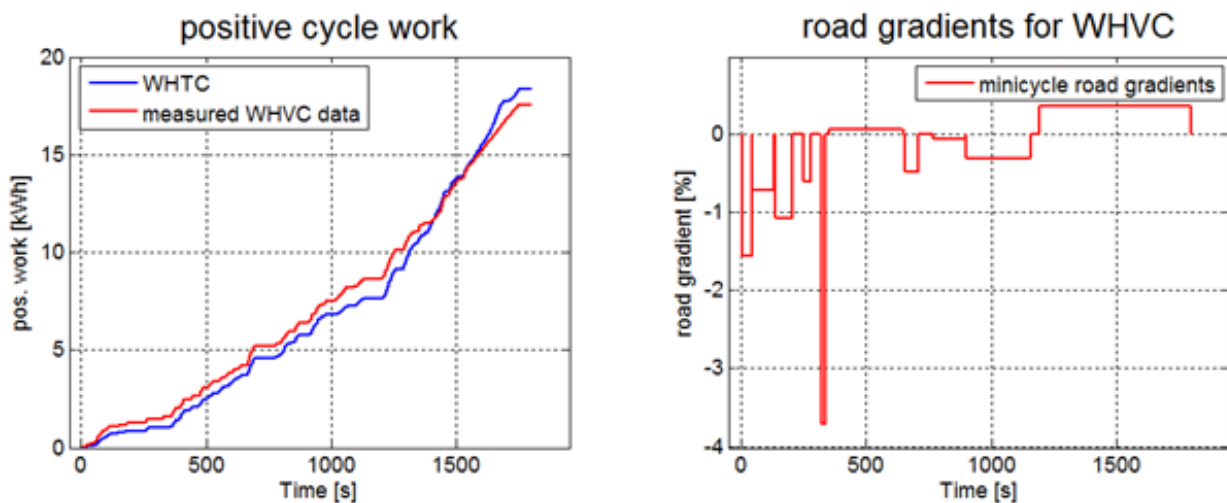
Figure 4.9: example of positive cycle work for a Volvo 7700 Hybrid Bus at WTVC

Due to the fact that the test vehicle mass to powertrain power ratio is similar to those used for generating the WHTC the cycle work is quite similar at test end but in the time history plot it can be seen that the tracking is not very accurate. This is caused by the way the WHTC was generated and at the same time means that the powertrain is operated in different ways/operating regions during the test cycle and is therefore not supposed to be comparable with the WHTC in terms of emissions. Figure 4.10 shows the impact of applied road gradients to the WTVC. The work can be tracked precisely during the first 11 mini cycles only the 12th mini cycle can't follow accurately. This is caused by the long duration of mini cycle #12 (longer than

600 seconds) and the fact that the WHTC has rather high fluctuating power demand during certain sections in that mini cycle even though the vehicle runs at constant speeds at the same time. For a better behaviour the last mini cycle was divided in further sub cycles. This modification was tested during the chassis dyno test runs of MAN at JRC and delivered promising results. Even though this method will not make the power time curve of the WHTC and the power time curve of the actual test vehicle identical second by second they look very similar on a greater time scale (e.g. 5-10 sec.) which is also indicated by the fact that their work time curve is nearly congruent. Therefore the emission behaviour is also supposed to be comparable (see OIL/D2).
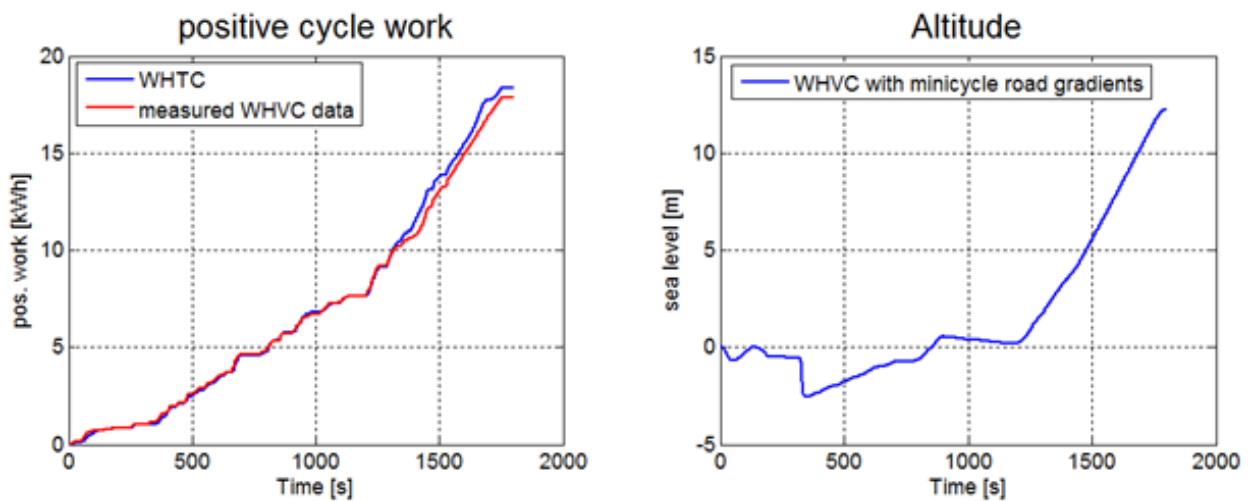
Figure 4.10:   example of positive cycle work for a Volvo 7700 Hybrid Bus with applied road gradients at WTVC

Figure 4.10 indicates another issue when adapting "road gradients" to a velocity dependent test cycle. Depending on the respective test vehicle mass and propulsion power road gradients can occur which force the vehicle to run uphill during the test cycle. In the illustrated case there is a minor impact but if the power to mass ratio of a vehicle is higher the altitude can be seriously increased. Considering the positive cycle work this would be no problem but since a HDH vehicle would have to recuperate energy during braking at positive road slopes this would be a clear handicap for hybrid powertrains because less energy is available for recuperation. Different vehicle data could also lead to negative road gradients during the whole cycle which would be a benefit for a hybrid powertrain since then more energy is available for recuperation. Using the altitude here is just a tool for a better imagination. Basically the altitude profile is representative of how much energy is additionally needed to match the positive WHTC cycle work with the respective vehicle during a WTVC test run. But this also means that it defines how much energy is provided by the test cycle for a HDH's energy recuperation system.

Different ways of adjusting the available energy for recuperation are possible. If it turns out that a vehicle would have to run uphill during the test cycle negative road gradients could be applied during sections of deceleration in a way that the gained altitude is reduced again respectively a certain amount of energy is available for recuperation. Since they are applied during sections of deceleration this would

have no impact on the positive cycle work. Figure 4.11 illustrates sections where negative road gradients could be applied. Sections of at least 12 seconds of deceleration were chosen for the first test runs. Of course you will get a more chopped slope pattern but this was not a problem during the chassis dyno test runs in VTP2.

WHVC drivecycle
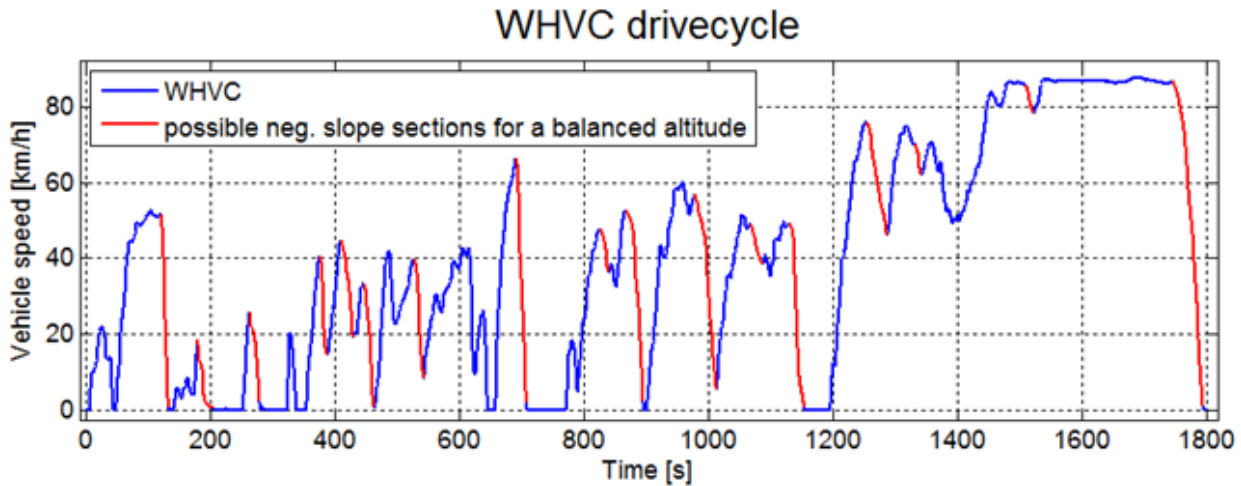
Figure 4.11:   possible sections of negative road gradients for a balanced altitude

For a vehicle with a high power to mass ratio (which would deliver too few cycle work during a WTVC and would therefore have to run uphill when applying road gradients) the approach of balanced altitude seems reasonable because it is supposed that the amount of available energy for recuperation is underrepresented for that vehicle (also braking during positive road gradients) but for a vehicle with a very low power to mass ratio the WTVC with road gradients would anyway go downhill during the entire test cycle to track the positive WHTC cycle work. There would be no possibility to gain altitude again during sections of deceleration, which also means that the vehicle would have an advantage by recuperating energy downhill.

To solve that problem road gradients at deceleration sections (see Figure 4.11) could at least be removed (set to zero). This would not balance the altitude (which is in fact only an imaginary one) but would make high and low powered vehicles comparable. Only setting negative road gradients for high power to mass vehicles and zero road gradients for low power to mass vehicles at sections of deceleration in the WTVC will not result in a fair comparison, even though the positive cycle work is identical with the respective WHTC for both methods. Probably setting the road gradients to zero for all vehicles during sections of deceleration would make the comparison fair but if this is reasonable and provides a representative amount of energy to recuperate for the vehicle is still part of the on-going investigations (see OIL/D3).

The procedure of calculating specific road gradients and therefore provide the test cycle for each specific test vehicle currently looks like this:

- Run the test cycle (WTVC on plane road) with a vehicle and record the work delivered (however, this can be done with a real vehicle or a verified vehicle model)
- Compare the positive work delivered with a positive reference work derived from the WHTC (which is in fact not as easy to determine – see section 4.3.1.3)
- Calculate road gradients which result in same cycle work as the WHTC to get the test cycle used for your HILS ICE certification run

This ultimately means that each powertrain system has its own test cycle (same velocity but different road gradients) and the cycle itself cannot be stipulated in a regulation, only the determination. To ease the procedure mentioned above the cycle work for the vehicle running a WTVC on a plane road could be also calculated by using the equations for vehicle longitudinal dynamics. This would make the first bullet point needless but since the entire vehicle behaviour (e.g. during gear shifts) cannot be represented by a simple longitudinal calculation slight deviations are assumed and a study of input parameters is needed first (e.g. inertia of rotating sections, see OIL/P1, P2 and P3). Nevertheless this is currently also done in the Japanese legislation to determine the work delivered during a chassis dyno test for a conventional vehicle.

Ideas have been presented in order to again simplify this procedure and define one common slope profile which could be established if the vehicle test mass is linked to the propulsion system's power.

- The vehicle test mass is representative of how much work is needed to run the given speed cycle.
- The reference work to be delivered during the test run, which is derived from the WHTC, is depending on the power of the powertrain.

If powertrain power and vehicle test mass are linked (like it was proposed by JASIC [3]) a common slope profile which could be 30.sec.mov.avg, mini-cycle or different approach based could be established. This approach is based on averaging different slope profiles for different vehicles. It could help to simplify the whole procedure, a test cycle consisting of speed and road gradient could be stipulated in the regulation, but still needs further investigations regarding the deviation between the different vehicles and the deviations between demanded and delivered power patterns. It can therefore not be reported yet..

Nevertheless the definition of an average power to mass ratio representative for both conventional as well as hybrid vehicles appeared difficult. Hybrids and especially serial ones turned out to have a power to mass ratio which is not comparable with conventional vehicles (see OIL/P1).

4.3.1.3   *Average WHTC*

Independent of the method used to define road gradients there is always the need of calculating the corresponding WHTC cycle work to be able to derive the road gradients for the certification (difference of positive WTVC and corresponding WHTC cycle work, see Figure 4.9). The corresponding WHTC for a HDH vehicle with a certain rated power would be a WHTC from a conventional ICE with the same power and the same shape of the full load curve. In order to denormalize the WHTC stipulated in the GTR No. 4 you need a full load torque curve and some

characteristic speeds of the ICE to derive the rotational speed and torque test pattern. This can be used to calculate the cycle work then.

Since the HDH powertrain can only be considered as a virtual ICE and not really available and since there are no denormalization methods for HDH full load curves ("idle" at zero rpm) developed yet one has to use the known denormalization methods for conventional engines to derive the cycle work to be delivered during the test cycle. Easily assumed, they do not really fit for hybrid powertrains. Figure 4.12 gives an example.
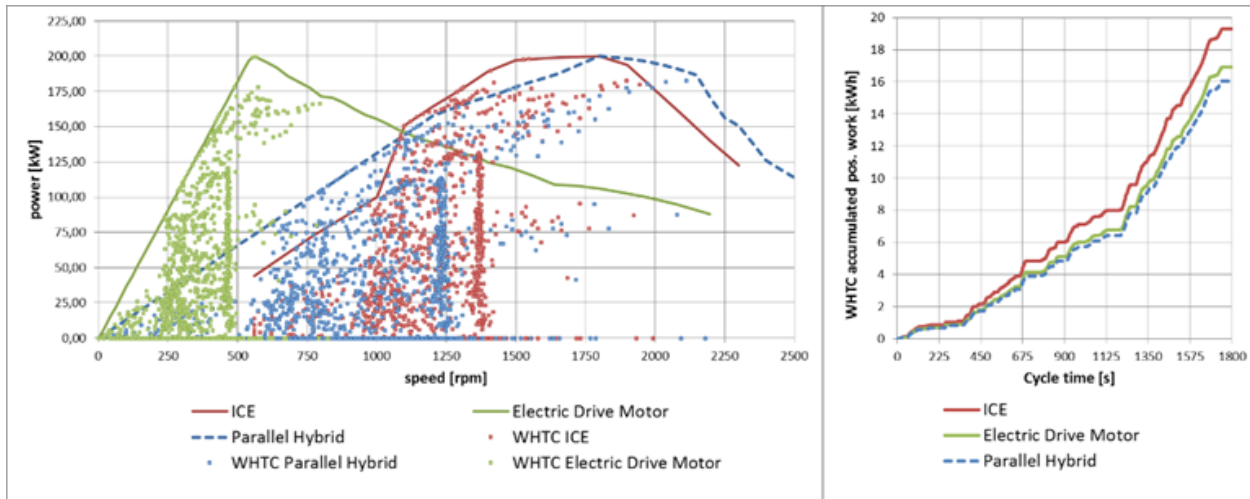
Figure 4.12: Comparison of cycle work and operating points for a 200kW ICE, a 200kW parallel hybrid powertrain and a 200kW electric machine using the WHTC denormalization method

Denormalizing a WHTC with a parallel hybrid powertrain full load curve will lead to a shift of rotating speeds to lower speeds. This is caused by the "idle" speed of the powertrain, which is zero for hybrids in general. Caused by this shift of rotating speeds and the shape of the full load curve the positive cycle work using the hybrid powertrain full load to denormalize the WHTC is here 17% lower than for the conventional vehicles propulsion engine with same rated power. A fair comparison would not be possible between conventional HD and HDH vehicles since there would already be a difference in the demanded cycle work. For serial hybrids it is even worse because the engine, which is compared to the conventional vehicle, can only be the one which is responsible for propelling the vehicle directly. This would be one or more electric machines and their full load characteristics are completely different to common ICEs. Because the results for parallel hybrids as well as for serial ones, even not thinking about alternative concepts were not satisfactory when using the known WHTC denormalization methods an alternative more practical method was developed.

Since currently insufficient data from HDH driving tests is available to generate an adapted method for HDH powertrains, a reference cycle which is in general a WHTC should be denormalized by just using the rated power of a powertrain. This would make the need of a full load curve and characteristic speeds unnecessary at all. Since the WHTC was derived from the WTVC and a normalized power time curve is also part of the WTVC definition the most obvious assumption would be to

use that normalized power time cure to define the cycle work to be achieved by the WTVC with applied road gradients. Figure 4.13 illustrates the normalized positive cycle work for WHTCs of 15 different combustion engines, for the normalized power of the WTVC and for the average of these 15 specific engines. For each engine the specific WHTC was calculated due to the shape of its full load and its characteristic speeds and then normalized by its rated power. Depending on the shape of the full load curve the cycle work is different even if the rated power of two engines is equal.
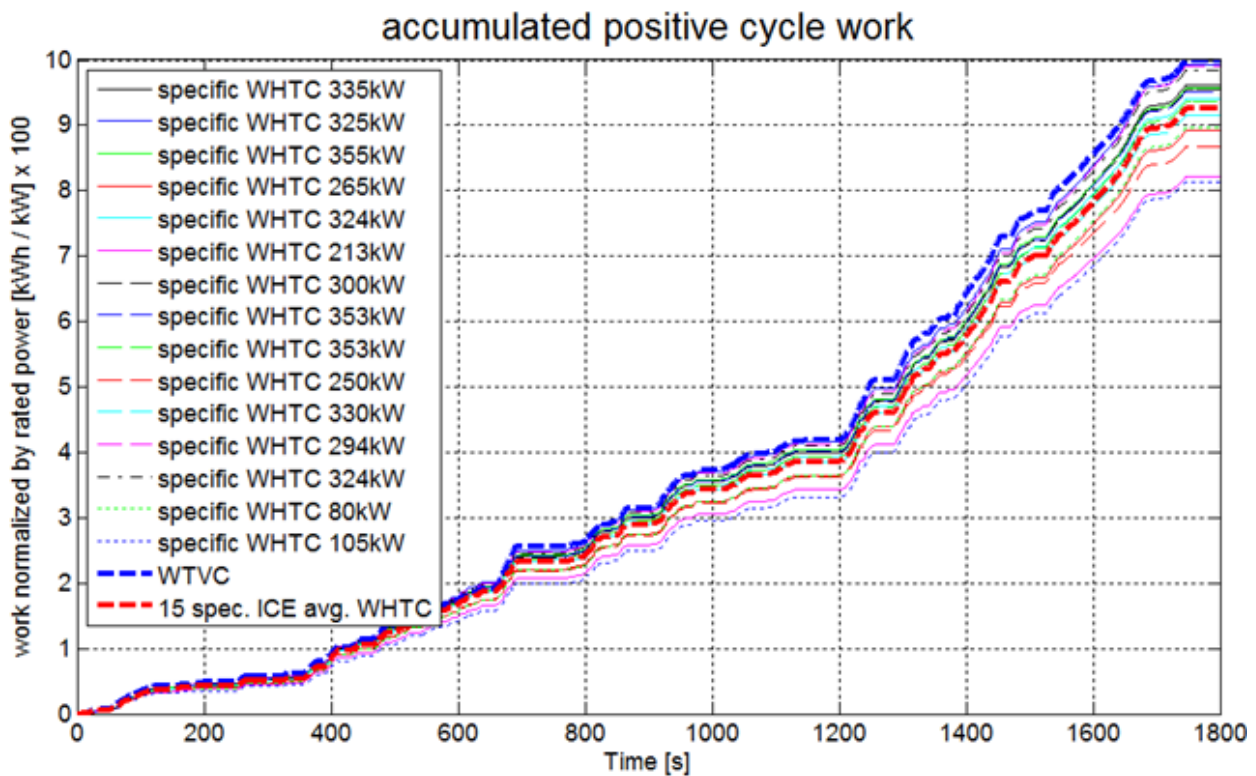
Figure 4.13:   positive WHTC cycle work of different combustion engines

For the considered heavy duty engines the highest positive cycle work is 7% higher and the lowest is 11 % lower than the positive average WTHC cycle work. On the very lower end of the chart you will find a passenger car engine with 12 % deviation from the average WHTC. The low positive cycle work is caused by its low torque at low rotational speeds which is also the case for the 213kW HD ICE (2nd lowest). Due to the modifications made during the development process of the WHTC the unspoilt normalized power time curve of the WTVC, which is on the very upper end of all engines investigated, produces 7% more positive cycle work than the average WHTC. This is caused by the normalized WTVC power time curve which is not comparable to the WHTC power time curve, and therefore not representative for conventional vehicles anymore (subsequently fitted gear shift events, partly different amplitudes, indicated in Figure 4.14). Because of that the average WHTC was used to calculate the positive cycle work for the tests in VTP2 although it is a very practical solution. Final decisions how to define the reference work have to be discussed in the HDH investigating group and have not been made until now (see OIL/D5).
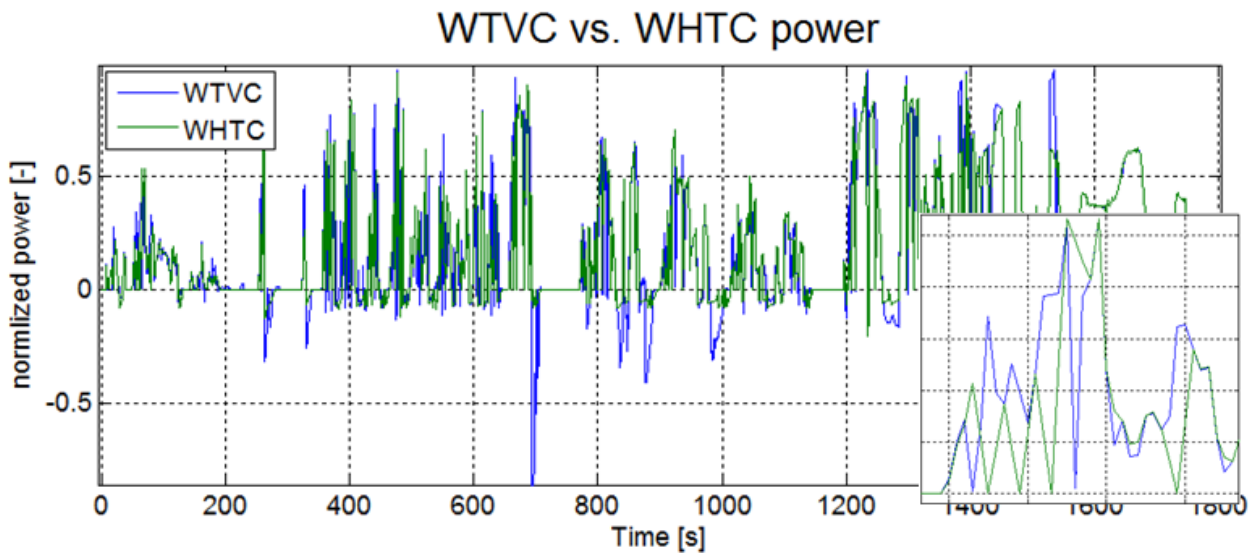
Figure 4.14:	normalized power time curve of WHTC and WTVC (the shape of the WHTC power pattern will always be identical, only amplitudes are changed due to different ICE full load)

Although the definition of the reference power pattern and cycle work seems quite manageable, defining how the rated power of a hybrid powertrain has to be specified is an open issue since the electric machines can partially deliver peak powers much higher than their rated continuous power (see OIL/D6).

4.3.1.4 *Emission simulation for a conventional HDV at WTVC with road gradients*

The basic aim of the new developed test cycle is to keep hybrids and conventional vehicles comparable even though the emission test methods are different. In order to proof if the new developed method containing a speed dependent test cycle with adapted loads by applied road gradients produces similar emissions than a respective WHTC a simulation test study was made. The only purpose in that early stage was to identify if the chosen approach is worth to be further investigated. This could be clearly affirmed. Actual measurements with a conventional HD vehicle could not be performed until now.

For the initial study a conventional 13 ton delivery truck equipped with a 248hp EURO5 ICE and a 12 speed gearbox was chosen. As emissions very much depend on the operation pattern of the ICE as well as on the transient behaviour the gearshift strategy for the vehicle was as good as possible set in a way that it is similar to the gearshifts included in the WHTC time curve.
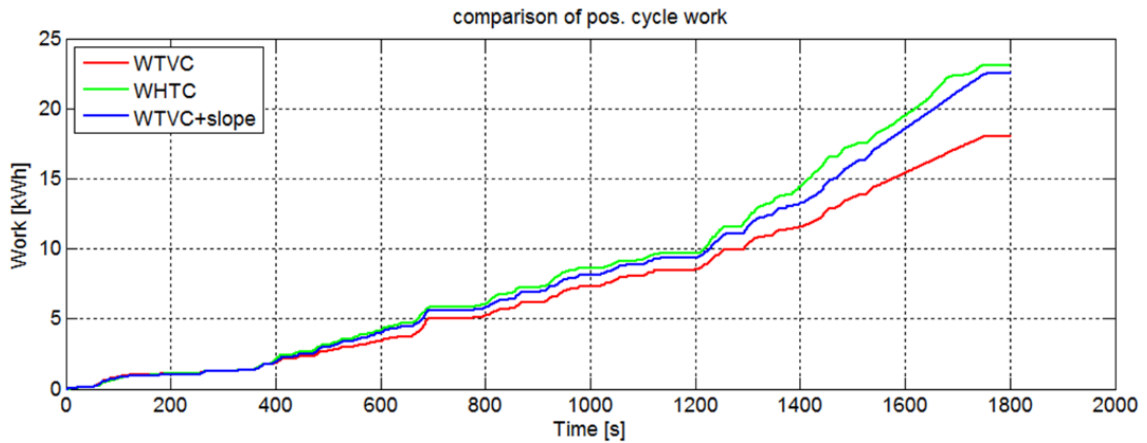
Figure 4.15:   positive ICE cycle work at different test cycles
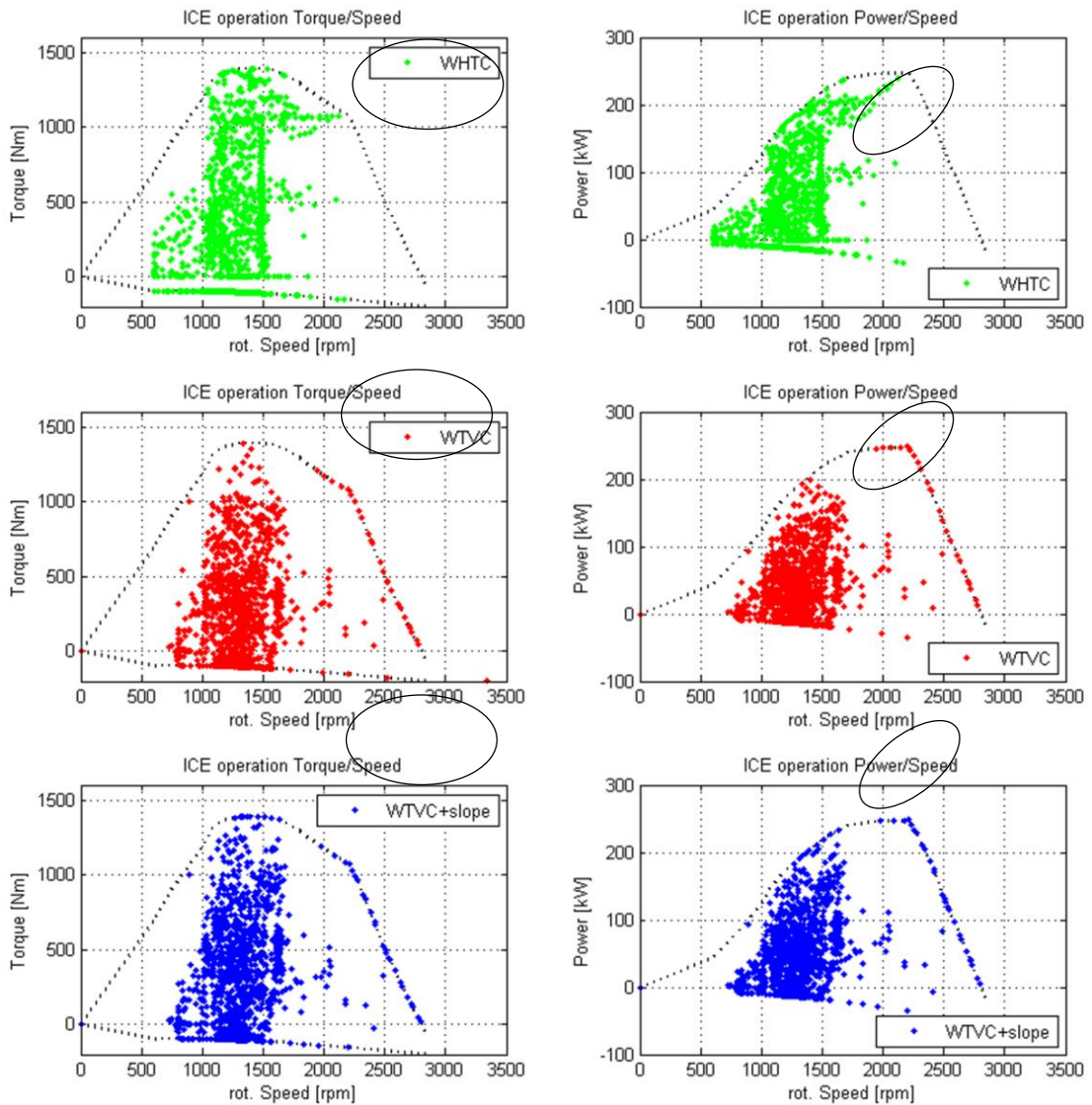
Figure 4.16:   ICE operation points for a 248 HP engine running a WHTC, propelling a 13 ton delivery truck at a
               WTVC on plane road and propelling a 13 ton delivery truck at a WTVC with applied road gradients

Figure 4.16 illustrates the resulting ICE operation pattern for the test vehicle running a WTVC. Considering that the WHTC was generated using an 8 speed generic gearbox and the specific vehicle used for the study was equipped with a 12-speed gearbox, the ICE operation pattern looks quite similar. The operation points which occur at full load and high revs are caused by the implemented gear shift strategy where the gear change was locked for 4 seconds after a gear shift event in order to prevent multiple shift events at certain thresholds. When the vehicle accelerates from standstill in gear 1 of 12 the time limit of 4 seconds is too high and the engine revs up too fast. Since the operation point on full load and high revs only appears during acceleration from standstill the gear shift strategy was not further development at this time. However, it can be seen that the percentage of full load is higher for the WHTC and the engine is burdened with lower load for that specific vehicle than at the WHTC. This is also reflected by the amount of positive work delivered during the test run (see Figure 4.15).

Out of the deviation between positive WHTC and WTVC work road gradients were calculated as it is described in the previous section and another test run was made. The WHTC work time curve could be tracked rather well for the first iteration (Figure 4.15) and the ICE operation pattern was shifted to higher loads (Figure 4.16).

Even though the gear shift strategy did not perfectly match the WHTC strategy it could be shown, that the engine load is adapted by the application of road gradients to adapt the road load to (better) match the WHTC load.

Finally the impact on emissions was investigated by using an emission simulation tool developed at TUG. The results are illustrated in Figure 4.17. Even though simulating emissions includes reasonable uncertainties and the deviations between the investigated test cycles seem rather small when the pollutants are referenced to the delivered respective cycle work the approach looked promising. Based on that it was decided to further investigate the described approach in VTP2.
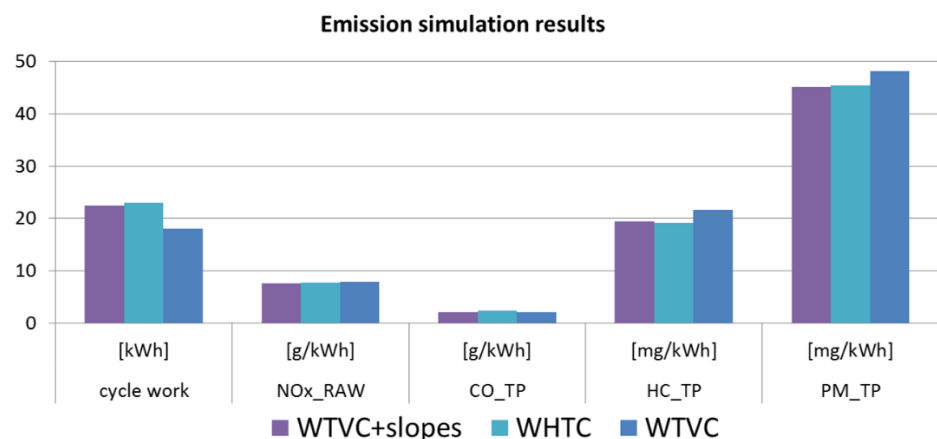
Figure 4.17:   Simulation results of specific emissions from different test cycles

4.3.1.5   *Drive cycle conclusions*
The mini cycle approach, the 30 second moving average approach as well as combinations of them are currently under further investigation and the descriptions here are just an outline of insights so far. Current investigations focus on a simplification of the procedure. E.g. calculating the road gradient pattern by using the equations for vehicle longitudinal dynamics instead of using measured data from a chassis dyno test run or a HILS model run would allow a simplification and a better handling for a legislation. The test cycle including the road gradient pattern could be calculated before any measurements are done and the HILS model could be verified by using the same cycle as for the ICE emission certification. However, a common valid vehicle test mass (e.g. bij calculation this based on the vehicle's rated power), which reference power pattern is used, how to deal with the consideration of a balanced altitude / which negative power provided is representative and how the rated power of a hybrid powertrain is defined have to be discussed in the HDH investigation group (see OIL/P1, D5, D3 and D6).

A fixed slope which can be defined in the GTR together with the test cycle would be desirable though needs to be further investigated.

4.3.2   *Additional issues to be discussed for a GTR adoption of the Japanese HILS method*

4.3.2.1   *Provisions for a chassis dyno test run*
This addresses the descriptions of Kokujikan No.280 "Measurement procedure for JE05-mode exhaust emissions by means of chassis dynamometer" which have to be proofed for a GTR adoption. Especially the method of setting the chassis dyno – paragraph 6, measuring of mapping torque curve – paragraph 9 and the driving procedure for test motor vehicle – paragraph 10 have to be reviewed. (see OIL/V1)

4.3.2.2   *Alternative HILS model verification test run*
On-road measurements for validating the HILS model have been proposed to be an attractive alternative to chassis dyno test runs. Especially for currently unconventional hybrid layouts (several driven axles with wheel hub motor,…) a verification run on a chassis dyno with one driven axle could be problematic. An on-road test without the need of manipulating the vehicles software would be more convenient if it is possible to reflect the road loads and the driving behaviour in the simulation. Although there was a positive feedback on the feasibility from OEM side in general no test runs could be performed until now. Investigation are planned for that reason in VTP2. Depending on the outcome a GTR adoption will be discussed (see OIL/V3).

4.3.2.3   *HILS model sample time*
A conversion method for ICE speed and torque from the HILS model sample time to 10Hz (or whatever is specified for the exhaust emission test later on) has to be defined. Especially sections of load change during gearshifts should not be filtered by inappropriate sampling or a too low resolution (see OIL/H3; see also section 4.1.2.4, paragraph 8-1-3).

4.3.2.4   *Vehicle independent certification method*
During meetings with OEMs the implementation of a vehicle independent powertrain certification similar to the WHTC method was discussed not exactly knowing that there is something similar already available in the Japanese

legislation. The general approach is of course to minimize test effort and avoid testing and certifying each specific vehicle.

Kokujikan No.155 therefore describes standardized vehicle specifications and how to handle them. Unfortunately no English translation is known to be available but an outline was kindly provided by our Japanese colleagues.

For conventional HD vehicles i.e. GVW > 3.5tons, it is required to prepare the program for the conversion from JE05 vehicle speed cycle into a reference engine running cycle by using individual vehicle and engine specifications in order to realize the test method for the engine installed in the individual vehicle. Because there are many kinds of HD vehicles with the same engine and in premise of keeping the sameness of exhaust gaseous emission and performance it is also allowed to use standard vehicle specifications described in Kokujikan No.155 (see tables below). Basically an OEM can decide freely if he uses real or standardized vehicle specifications for a certification.

- In case of using standardized vehicle specifications:
  An OEM is able to introduce a new vehicle with an already certified ICE in a certified category without a new certification (like it is handled for the WHTC). But this in fact means that the gearbox is not part of the certified powertrain – standardized gear ratios and a standardized gear changes provided by a conversion program have to be used. So practically only a pre-transmission powertrain test (like the Japanese system bench test) can be run to certify the engine. The combination of internal combustion engine and electric motor can be sold with any number of different gearboxes and gear shift strategies.

  If the gearbox would be part of the certification – which can be done as well - the current Japanese legislation would require to have the same gearbox as well as the same gear shift logics in each vehicle the powertrain is mounted and sold.

  One additional thing to be noted is in case of using standardized vehicle specifications a bus and a truck are handled in the same specification which means the air resistance is also the same for both, using worst case data i.e. the truck resistance should be used for a bus also.

- In case of using specific vehicle specifications:
  An OEM basically needs to run a new certification if he wants to introduce a new vehicle e.g. if a VAN is introduced in addition to a certified truck with rear flat body in the same category the exhaust gas shall be newly certified because the front area of a VAN is wider than for a flat body. But if a certified real vehicle specification can cover the new vehicle specifications, e.g. a VAN is already certified and a new truck with rear flat body will be introduced, a new certification is not necessary. The "worst case" vehicle in terms of emission shall be chosen for the certification test. The necessity of a new certification is judged case by case on different criteria which are not regulated but JAMA negotiated the basic concept with NTSEL:

  First priority is to choose the lowest V1000[2], second priority is to choose the widest front area and the third priority is to choose the heaviest GVW.

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[2] V1000 means the vehicle speed at 1000 rpm ICE speed using the highest gear position e.g. the 5th gear position of a 5 speed transmission gearbox

	V1000 [km/h]	Front area [m²]	GVW [10³ kg]	certify
Vehicle A	40	6	7,5	
Vehicle B	38	6,5	7,1	worst case
Vehicle C	38	5,6	7,4	

A re-certification becomes necessary if an even worse vehicle is introduced in the market.

	V1000 [km/h]	Front area [m²]	GVW [10³ kg]	re-certification
Vehicle D	39	7	7,5	not necessary
Vehicle D	38	6,5	7,5	necessary
Vehicle F	36	5	5,5	necessary

However in any case the negotiation with NTSEL might be necessary for OEMs since no regulation describes this proceedings. Interestingly the gear change pattern is currently not considered for defining the worst case vehicle. This in fact means that an OEM could certify a vehicle (e.g. city bus) with specific vehicle data and one specific gearbox ECU logic and he would be allowed to change the gearbox ECU's software without the need of recertification even though this could affect emissions.

Since defining the "worst case vehicle" for a hybrid HDV is quite difficult because there are more degrees of freedom influencing the ICE operation pattern and the resulting emissions practical solutions have to be found to minimize the full test effort (see OIL/C1).

An example for a solution to reduce test effort could look like:

- Validation of a HILS model with one vehicle on the chassis dyno (automatic gear changes by the gearbox ECU because no manual switching is possible)
- Use the validated HILS model and set standardized vehicle parameters + gear ratios to run the HILS model. Also Use standardized gear changes (e.g. VECTO) there and have the ICE certified for each bus in this vehicle class.

(having the gearbox ECU on the HILS test rig for model verification and then change vehicle parameters and switch to prescribed gear shift maneuvers (VECTO) could be problematic because it affects CAN bus simulation and may requires changes in interface model, the ECU software,....)

Another example could be:

- Validation of a HILS model with one vehicle on the chassis dyno (automatic gear changes by the gearbox ECU because no manual switching is possible)
- ICE operation pattern for certification will be derived using HILS (test weight derived from rated power, actual final gear and gearbox ratios used, actual gear shift ECU/logics used)
- Powertrain allowed to be used in other vehicles at same class as long as emissions are not higher there (OEM is in charge to ensure this/ to certify the worst case vehicle in order to reduce certification effort)

Whether these examples are feasible or not, different solutions have to be discussed in the HDH investigation group. Because this issue is very much related with the question of a family concept, the need of re-certification/re-verification and the implementation of the gearbox in the certification process it cannot be solved without a comprehensive consideration.

4.3.2.5　*Prevention against ECU failure mode*

When running a HILS system consisting of a vehicle simulation model linked with control units of the real vehicle as hardware parts it is essential to provide all sensor signals that affect operation of the hybrid system to the control units. This ensures that the simulated system operates equally to the real vehicle and keeps the control units from changing into a failure operation mode which is not representative for real life operation. There are basically two ways to provide the respective sensor values to the control units:

a) Using signals generated inside the OEM-specific, unique interface model which can be recorded values from previous in-vehicle measurements, artificially generated values or fixed, constant values for switches, flags or status signals

b) Implementation of an ECU test mode in the control unit where failure operation modes are not implemented and a reduced number of sensor signals is needed for a hybrid system operation. Even though this is a common approach for conventional engine testing it has to be ensured that this has no impact on the hybrid system operation itself. Signals used for the anti-brake-lock system, certain OBD functions, driver support systems, etc. could be removed for example to minimize the certification effort.

This issue is vital for the application of the HILS procedure since it can become very complex with several control units included in the certification process. However, if this issue has to be addressed in the addendum to the GTR or is handled by regional or local authorities shall be discussed inside the drafting group of the regulation (see OIL/V5). GTR No. 4 is currently also not addressing how to deal with preventing failure operation mode of the engine control unit during engine test run on the test bench.

standard vehicle specification(trucktor) by MLIT for fuel consumption

trucktor category		empty vehicle mass	maximum payload	number of persons	test vehicle mass	tire dynamic radius	overall hight	overall width	transmission gear ratio							diff gear ratio	rate of inter-city mode
category NO	tructor head mass range GVW(kg)	(kg)	(kg)		(kg)	(m)	(m)	(m)	1st	2nd	3rd	4th	5th	6th	7th		
TT1	&≤20t	10525	24000	2	22580,0	real vehicle data of the most close to average v1000	2,927	2,490				real vehicle data				real vehicle data of the most close to average v1000	20%
TT2	20t<	19028	40000	2	39083,0		2,890	2,490									10%

trucktor head GVW=tructor head empty mass+maximum pay load (5th wheel load)+(number of persons) x 55kg

test vehicle mass= empty vehicle mass +maximum pay load/2+55kg

standard vehicle specification by MLIT for exhaust gas

category NO	truck/tractor category vehicle mass range GVW/GCW(kg)	payload range	bus category NO	bus vehicle mass range GVW(kg)	fuel	curb vehicle mass (kg)	maximum payload (kg)	number of persons	test vehicle mass (kg)	tire dynamic radius (m)	overall hight (m)	overall width (m)	1st	2nd	3rd	4th	5th	6th	7th	diff gear ratio
T1	3.5t<&≦7.5t	≦1.5t	-	—	D·LPG·CNG	1957	1490	3	2757,0	0,313	1,982	1,695	5,076	2,713	1,529	1,000	0,795			4,615
					G·LPG·CNG	1659	1458	3	2443,0	0,303	1,975	1,695	4,942	2,908	1,568	1,000	0,834			4,477
T2	3.5t<&≦7.5t	1.5t<	B1	3.5t<&≦6t	D·LPG·CNG	2482	2396	3	3735,0	0,343	2,106	1,780	5,080	2,816	1,587	1,000	0,741			5,275
					G·LPG·CNG	2259	2016	3	3322,0	0,327	2,052	1,722	5,089	2,773	1,577	1,000	0,777			6,051
T3	7.5t<&≦8t	—	B2	6t<&≦8t	G·D·LP G·CNG	3543	4275	2	5735,5	0,388	2,454	2,235	6,350	3,876	2,301	1,423	1,000	0,762		4,771
T4	8t<&≦16t	—	B3	8t<&≦16t	G·D·LP G·CNG	4527	7737	2	8450,5	0,469	2,617	2,374	6,416	4,096	2,385	1,475	1,000	0,760		5,208
T5	16t<&≦20t	—	B4	16t<&≦20t	G·D·LP G·CNG	8688	11089	2	14287,5	0,502	3,049	2,490	6,331	4,224	2,410	1,486	1,000	0,763	0,612	6,309
T6	20t<&≦25t	—	B5	20t<	G·D·LP G·CNG	8765	15530	2	16685,0	0,473	2,934	2,490	6,304	4,170	2,393	1,456	1,000	0,752	0,604	5,102
T7	25t<	—	-	—	G·D·LP G·CNG	12120	24974	2	24662,0	0,507	2,961	2,490	6,147	4,000	2,281	1,434	1,000	0,760	0,597	6,061

truck GVW=empty vehicle mass+maximum pay load+(number of persons) x55kg
bus GVW=empty vehicle mass+(number of persons)x55kg
test vehicle mass=empty vehicle mass +maximum pay load/2+55kg

tractor
"the fifth wheel coupling load" is 8ton or more: you shall select "T7 category".
"the fifth wheel coupling load" is less than 8ton : GVW=Tractor vehicle weight + the fifth loading + 55kg x number of persons and select in T1 to T6

5   Open issue list for a GTR adoption

Open issues mentioned in the report are bundled and grouped by sections as far as possible in the following tables. Prioritization by using numerals 1-3, where 1 is the highest priority.

Table 5.1   OIL: certification procedure

#	Issue	Status	Priority
C1	Define which approach can be used for HDH certification (see 4.3.2.4) • Standardized generic vehicle • Worst case vehicle • Actual vehicle • Approaches as alternative option in parallel	HDH group	1
C2	Define which reasons define a re-certification, discussion depends on chosen approach in C1 (see 4.3.2.4) Is a suitable GTR definition of a worst case vehicle (like in Japan) possible? (see 4.3.2.4)	HDH group	2
C3	How should gearboxes and shift algorithms be handled? (4.3.2.4) • Should a gearbox in general be part of the certification? • If yes, should it be a standardized gearbox and shift algorithm or the individual gearbox and shift algorithm?	HDH group	1
C4	Should post-transmission powertrain test, HILS with verification on chassis dyno and HILS with verification on system bench (pre-transmission powertrain test) become alternative options for emission certification in the GTR? • For HILS system test bench (pre-transmission powertrain) a conversion program from speed cycle to rotational speed and torque powertrain cycle and therefore generic gearboxes and shift provisions would be needed • Alignment between post-transmission powertrain testing and HILS (chassis dyno) testing is necessary. • HILS (chassis dyno) and post-transmission powertrain test would be compatible	HDH group	1
C5	Definition of reference base for calculation of specific emissions (i.e. emissions per unit work) (see 4.1.2.4 - 11)	HDH group	3
C6	Electricity balance in HILS simulation run for generating ICE test cycle (see 4.1.2.4 item 8) a. Limit for delta SOC during simulation run specified in Kokujikan No.281 has to be checked b. Is the calculation with integrated current multiplied by nominal voltage according to Kokujikan No.281 valid?	VTP2	–
C7	On what basis does the procedure define whether conventional or hybrid part should be used for certification ? • Should WHTC+WHSC still stay valid as alternative or should HILS be mandatory for HDH (see section 3.1.2)? • Definition of various hybridization grades (Pbat/Pem/Pice) to differentiate between HDH and conv.HDV purposeful (e.g. treatment of battery el. vehicle, stop&go functionality,…)	HDH group	3
C8	Do other procedures/applications (like LD) provide a potential solution for HDH issues and is carry-over/implementation possible?	HDH group	–

#	Issue	Status	Priority
C9	Will other topics that are not directly included in the current GTR No.4 (like OBD, ISC and so on) require specific changes for HDH ?	HDH group	—

Table 5.2   OIL: drive cycle development

#	Issue	Status	Priority
D1	Matching of positive cycle work via additional road gradient (see 4.3.1) Define the best solution for the application of road slopes out of: a. Mini cycle or Moving average calculation b. Reference work analytically calculated or actual work from dyno test / HILS model	VTP2	1
D2	Comparability of developed method for HDH (WTVC + road gradients) with conventional vehicles (WHTC engine testing) in terms of emissions? (see sections 4.3.1.2 and 3.1.2)	VTP2	1
D3	The amount of negative cycle work (i.e. potential regenerative energy) available for a HDH has to be defined (corresponds to the balanced altitude approach, see 4.3.1.2).	VTP2	1
D4	Should the road gradient that is applied as additional driving resistance is fed as signal into the vehicle ECUs or not? Clarify if a. The road gradient should influence the gear shift decisions in the vehicle ECU. b. The road gradient should represent real road slopes or only additional road loads.	VTP2	—
D5	Should the "average normalized WHTC" or normalized WTVC power be used to define the reference cycle work? (see 4.3.1.3) • Are other options available and possible?	HDH group	3
D6	In order to denormalize the test cycle and/or to calculate vehicle parameters a definition for the rated power of a hybrid system needs to be established. (see 4.3.1.3) a. How are peak powers of a hybrid powertrain measured or determined? b. For a parallel hybrid: only ICE power or total powertrain power? c. For a serial hybrid: continuous or maximum power?	HDH group	2
D7	How to proceed with vehicles which are by design not able to follow a given speed cycle (e.g. city bus max. speed)? • Limit test cycle max. speed to max. vehicle speed? • Nevertheless demand the power of the corresponding WHTC during that sections (high road gradients which could lead to overheating since the vehicle is not built for such power demands) or scale down the power demand to the lower speed limit? Handling for a fixed slope approach if this will be followed	HDH group	2

Table 5.3   OIL: HILS model general issues

#	Issue	Status	Priority
H1	Who will be the owner of / responsible for the HILS model after HDH workgroup is terminated? a. Model maintenance b. developing and introducing new components c. error handling in model	HDH group	3

H2	a. Standardized gear shift model has to be developed (adapted from European CO_2 calculation tool VECTO) (see 4.3.2.4)
	• If transmission is included in certification process:
	   o At least for manual transmission a standardized gear shift logic is needed for certification
	   o Depending on approach for automatized transmissions a standardized gear shift logic might be needed as well (see open issues under OIL-certification procedure)
	• If transmission is not included in certification process (not proposed):
	   o standardized gear shift logic is needed for certification with standardized gear box
	b. Definition of generic shift parameters depending on powertrain characteristics (torque curve) for hybrid powertrains is needed
	c. Clarify if gearshift logic works for parallel hybrids, since it is developed for conventional ICE
	d. Implement gear shift logic in Simulink model and perform test runs
	e. Reference to transmission input torque or ICE output torque has to be defined
		VTP2 + HDH group	3

H3	Cycle transformation from HILS model output to engine test cycle (see 4.3.2.3 and 4.1.2.4 paragraph 8-1-3) • A conversion method has to be defined from the HILS model output in high frequency to the lower frequency of the reference points of the engine test cycle (e.g. 100 Hz to 10Hz) • A high frequency (at least 10Hz) model output is necessary to depict torque interruption during gear shifts	VTP2	–
H4	Consideration of traction force interruption and impact on emissions • The impact of traction force interruption during gear shifts on emissions (like included in the WHTC) has to be investigated. • Decision if the gear shift dynamics should be included in the resulting engine test cycle has to be made. (see 4.1.2.4 paragraph 8-1-3)	VTP2	–
H5	Should operation check of HEV model and HILS hardware using a software ECU be performed in advance to the HILS test? • Would require HILS dummy data (see 4.1.2.1-item 7 and 4.1.2.4-item 8)	HDH group	3

Table 5.4   OIL: HILS model input parameters

#	Issue	Status	Priority
P1	How to define the vehicle test mass? (see 4.3.1.2 and 4.3.2.4) 1. Vehicle classes 2. Specific vehicle + half payload 3. As a function of powertrain power (representative for HD and HDH or differentiation necessary?) (effect on drive cycle and dependent on certification approach C1)	HDH group	2
P2	Drag and rolling resistance values derived from Kokujikan No281 should be checked if they are representative for current vehicles (see OIL/P1) • They are defined as function of vehicle mass -> used mass? Agree on e.g. m=f(p_rated) or specific kerb mass + half payload	VTP2 (request to OEMs)	—
P3	7% of kerb mass are foreseen as inertia of rotating sections for each HILS certification run (because a validated HILS model is allowed to be used for different vehicles where rotating masses can not be checked) • check if representative for conv. HDV and HDH (see 4.3.1.2) • maybe set to at least 7% or, since the HILS model topology is not allowed to be changed, the value used for model verification	VTP2 (request to OEMs)	—

Table 5.5   OIL: HILS model verification

#	Issue	Status	Priority
V1	HDH chassis dyno test procedure available? (see Kokujikan No.280 and 4.3.2.1) • Definition of test start: "key on" or "board system already alive" or "propulsion system running" or…	Drafting group	—
V2	Specify when a model re-verification is necessary a. are changes in the interface model allowed (see 3.1.1) b. multiple ECUs and ECU functionalities in the interface model (see 4.1.2.5-1) c. vehicle mass exceeding the 12ton limit purposeful? (see 4.1.2.5-5)	HDH group+ OEMs	2
V3	On-road tests to be proofed as alternative for a model verification (see 4.3.2.2)	VTP2	—
V4	Which ECUs can be modeled as SILS solution in the interface model? • define an "actual ECU" which has to be at least present in hardware in the HILS test rig if possible (see 3.1.4)	HDH group	2
V5	How to avoid ECU failure modes due to missing signals on the HILS test rig (see 4.3.2.5) • Dummy signals generated in OEM specific interface • Software switch in ECU / ECU test mode (simplified software, failure handling disabled) Where is the borderline between HILS and SILS?	HDH group	2
V6	Tolerances between measurement test run (chassis dyno or system bench) and simulation run for HILS model verification specified in Kokujikan No.281 have to be checked (depending on VTP2 outcome) (see 4.1.2.5)	VTP2	—

#	Issue	Status	Priority
V7	Validation criteria for model verification according to Kokujikan No.281, chapter 5, paragraph 6 do not include the rotational speed of the combustion engine as separate criterion (vehicle speed for chassis dyno test or engine rev. for system bench test validation). Should rotational speed of the combustion engine be included as separate permanent criterion? (see 4.1.2.5)	VTP2 + HDH group	3

Table 5.6 OIL: cold start

#	Issue	Status	Priority
S1	Cold temperatures (20°C) are no issue for component performance BUT Is there a need to represent overheating in the HILS model or is it possible to assume normal operation? (see 3.1.4 and OIL/D7)	VTP2	–
S2	Engine mapping at warm and cold temperatures necessary? → Hermite interpolation for friction and fuel consumption could be used (see 4.1.2.2-3)	VTP2	–
S3	Cold start should be part of the model verification → Certify real or generic vehicle behaviour during system heat up? • Real behaviour 　▪ Unproblematic for model verification since measured temperature signals can be used as model input 　▪ Unproblematic for certifying vehicle where model was validated with 　▪ Problematic for different vehicles using an already verified HILS model (how to reflect real temperature behaviour here?) • Generic behaviour 　▪ Unproblematic for model verification since measured temperature signals can be used as model input 　▪ Use generic temperature models for every certification run of HILS model (could cause ECU errors/error-modes if temperatures differ from ECU models/estimations, could lead to different than in-vehicle ICE heat up operation where emission regulations can not be passed)	VTP2 + HDH group	2
S4	Would cold start test of ICE certification cycle (derived from HILS model at warm conditions) on the engine test bed be an option?	HDH group	3

Table 5.7 OIL: component test procedures

#	Issue	Status	Priority
T1	Are they just a guideline or are the procedures mandatory? • just to use supplier data would be convenient (NDAs could avoid publication), but since the changes of component maps is allowed in a verified HILS model, this data has to be proofed somehow (see 4.1.2.2)	HDH group	2
T2	Do they have to be proofed by authorities?	HDH group	3

Table 5.8 OIL: CO$_2$ interface

#	Issue	Status	Priority
F1	Different approaches to be investigated for HDH CO$_2$ determination • direct speed and torque interface (if vehicle dependent: CO$_2$ declaration would require a HILS test-run for each vehicle) • vehicle speed and power represented by road gradient (mini cycle or moving average) as input to match VECTO power • VECTO vehicle parameter and drive cycle used for HILS model • Vehicle family concept with one FC bonus factor • others…	Partly investigated in VTP2	—
F2	Handling over power demand for auxiliaries from conv. HDV CO$_2$ calculation program (e.g. VECTO) will not be desirable • conv. HDV calc. program would need to be able to represent all HDH accessories and their actuation • even no decision made yet in European CO$_2$ group for conv. vehicles how to handle auxiliaries for FC	open	—

6   Conclusions and recommendations

This report is the final report of the work of TUG, IFA, Chalmers and TNO performed within the research program on an emissions and CO_2 test procedure for Heavy Duty Hybrids (HDH). This report specifically refers to Validation Test Program 1 (VTP1). The work is performed according to specific contract SI2.631381, titled "Developing the Methodology for Certifying Heavy-Duty Hybrids based on HILS", and sponsored by the European Commission.

The work in VTP1 targets the identification of issues and possible improvements for applying HILS methodology (specifically based on the Japanese Kokujikan No.281 regulation) towards implementation in a Global Technical Regulation, more specifically towards GTR No.4.

6.1   Conclusions

The main objectives of Task 1 are:
- The preparation of a serial hybrid model using SIL simulation
- Providing additional powertrain components/models in order to meet stakeholder demands and ensure the establishment of a comprehensive model library
- Providing different driver models in order to be able to perform model test runs, investigate the model behaviour and the impacts of different test cycles

With regard to the previous bullet points, the achievements can be summarized as follows:
- A basic serial hybrid model provided by our Japanese colleagues could be extended and model test runs could successfully be performed with new components, different driver models and different vehicle parameters
- New powertrain components have been developed and already transferred into the later introduced new model structure (except planetary gear set)
- The implementation of a driver model capable of running a test cycle referenced to a certain power time curve could be successfully tested, but faced some serious weak points related to the test cycle itself. In contrast to the conventional driver model (tracking the vehicle speed), it was therefore not transferred into the new model structure until now.

The main objectives for Task 2 are:
- OEM and Stakeholders meetings to deliberate on HILS method
- Enhancement of the HILS model library and specifically the parallel hybrid topology

Responses from OEM on HILS methodology relate to:
- Changes allowed in the interface model
- Consideration of traction force interruption at the HILS model run
- Test cycle command frequency for an engine emission test
- Possibility of certifying HDHs using a WHTC engine test
- Multiple ECU handling
- Dummy signal handling / ECU test modes

Starting from the Japanese component models presented in Kokujikan No.281, a new model structure has been proposed and implemented for the suggested GTR HILS methodology. This includes the development of a new HILS library in which the component models with data bus structure are stored. The new library consists of many component models as building blocks that offer flexibility for building different conventional and hybrid system models. This also allows for easily adding new or future hybrid systems. In addition to the Kokujikan No.281 component models, several other components as well as basic thermal (warm up) behaviour are now available. All models have been tested to run numerically correct and provide physically representative results. Two example models are provided as part of the library, more specifically a series and a parallel hybrid topology. Complete vehicle model validation has not yet been performed (work in VTP2).

Task 3 focuses on the HILS procedure using Kokujikan No.281 as base document for adoption towards a Global Technical Regulation. The aim of this task was to review the procedures for component testing, application of the HIL simulator methodology and validation of the HILS set up. For all sections of the procedure, the technical issues are addressed and possible solutions may be indicated where suitable.

It is stated that Kokujikan No.281 provides a suitable base for drafting a GTR. Due to various reasons, a large number of Open Issues are defined and need further discussion and/or research to reduce the lack of clarity, ambiguities and (surplus-) options prior to finalizing the GTR procedures. As a result, suggestions for a draft text may not be available in all sections.

An (accepted) change in comparison to Kokujikan No.281 involves the building of the HILS model as part of the procedure, rather than using a predefined model. This allows for higher flexibility and more dedicated representation of OEM's hybrid powertrain topologies.

With regard to the component test procedures, it is identified that they can basically be copied from Kokujikan No.281, yet that it should be allowed to apply (already available) data from the OEMs and their Tiers for correctly calibration the models. Nevertheless the question how the reliability of this data could than be granted remains unsolved.

An important item within the test procedure is the definition of the test cycle. In the current GTR and regulations, all Heavy-Duty engines are subject to test on the WHTC engine test cycle (normalized engine speed and load) to determine emission performance. For HD Hybrid vehicles, the WVTC cycle (vehicle speed and normalized power) seems appropriate, yet the transformation through HILS towards engine or powertrain test cycle does require some additional measures. The investigation is still on-going. At the same time, the cycle work for calculation of the specific test results is subject in discussions. Kokujikan No.281 refers to vehicle drive shaft work for both CO_2 and pollutant emissions. Especially, the latter may more reliably be based on actual engine work as currently applicable and defined in GTR No.4.

6.2   Recommendations

Although many parts of the Kokujikan No.281 are suitable for adoption in a Global Technical Regulation, it is clearly identified that many details need further discussion before a Heavy-Duty Hybrid test procedure can be confirmed final. The Open Issue List in Section 5 targets to identify current items that have been addressed. For several items, Validation Test Program 2 has already been started to apply and further investigate the HILS procedure. For a larger part of the items, it is referred to the HDH Working Group to discuss need for additional investigations and where possible make justified choices based on technically valid rationales.

As the current GTR No.4 specifically targets the certification procedure for pollutant emissions, it is also advised to carefully evaluate the HILS procedure with regard to pollutant regulation versus CO_2 regulation. In order to ensure correct environmental and societal impacts, it may be necessary to define a procedure that incorporates clearly distinct conditions for one or the other. Other application field with similar technologies (like light-duty) may turn out, and are likely, to have similar issues and potential solutions in place already and should not be put aside without examination.

7   References

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8   Signature

Delft, 7 October 2013

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A   Component models

Appendices A and B reflect the status of the model library at the time the report was compiled. Updated information can be found in the new version of the library.

The component models are categorized into different categories. The models are categorized into the following categories:

- Auxiliary system
- Chassis
- Driver
- Electrical components
- Energy converters
- Mechanical components
- Rechargeable energy storage systems

Each category contains component models related to that specific category.

A.1   Auxiliary systems

A.1.1   Electric Auxiliary System

The electrical auxiliary system is modelled using a constant electrical power loss, $P_{el,aux}$. The current that is discharging the electrical energy storage, i_{aux}, is determined as

$$i_{aux} = x P_{el,aux}/u \tag{A.1}$$

where x is an on-off control signal for turning the auxiliary load on or off and u is the energy storage voltage.

A.1.2   Mechanical Auxiliary System

The mechanical auxiliary system is modeled in the same way as the electrical auxiliary system, using a constant power loss, $P_{mech,aux}$. The power loss is regarded as a torque loss current that is discharging the electrical energy storage, i_{aux}, is determined as

$$T_{out} = T_{in} - x P_{mech,aux}/\omega \tag{A.2}$$

where T_{in} is the in-going torque, x is an on-off control signal for turning the auxiliary load on or off, ω is the rotational speed and T_{out} is the out-going torque. If the mechanical component has an inertia, J_{aux} it can be included in the model as well.

A.2   Chassis

A basic model of the chassis (the vehicle), where the chassis is represented as an inertia. The model computes the vehicle speed given propeller shaft torque and

brake torque. The model considers rolling and aerodynamic drag resistance and takes the road slope into account, see Figure A.1.
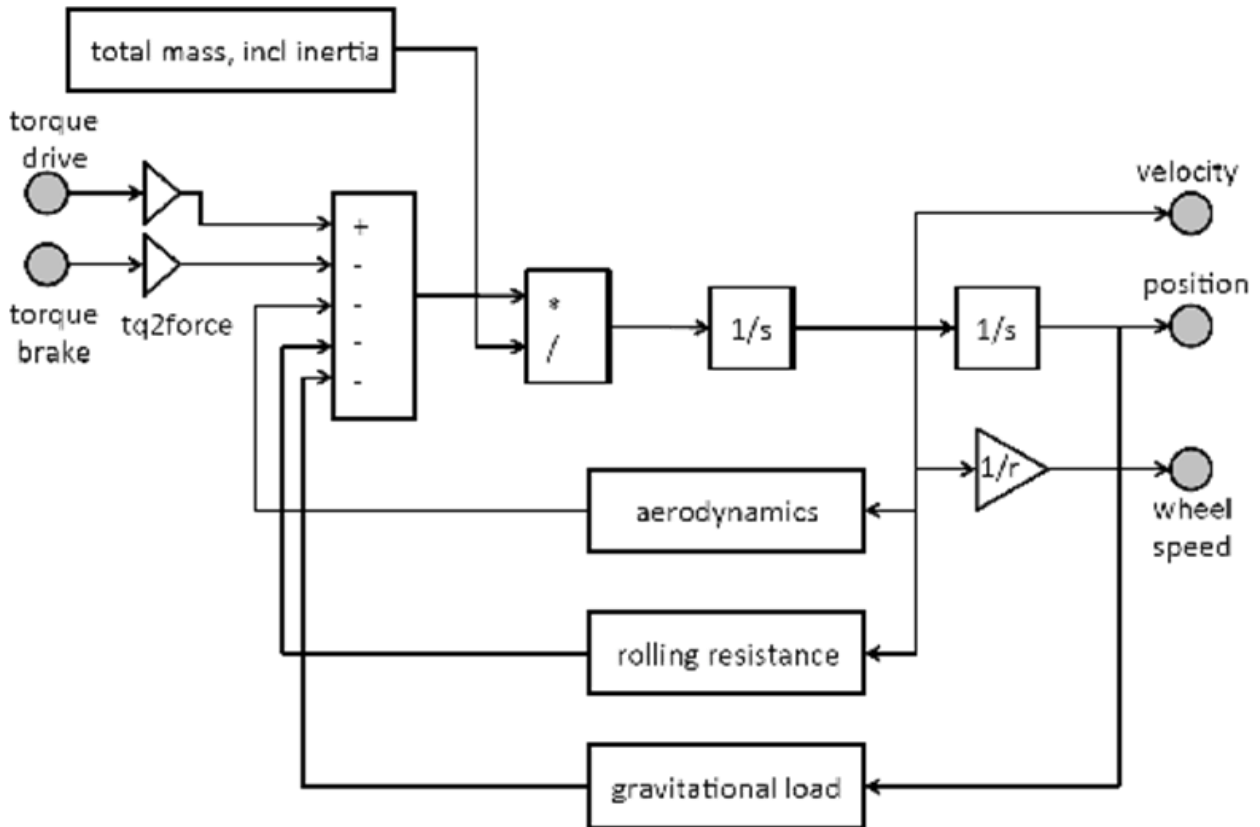
Figure A.1: Block scheme for chassis component model. The gravitational load can be switched between being position based and time based

The basic principle is that the input torque T_{in} goes through a gear reduction (final gear) with ratio r_{fg},

$$T_{drive} = \eta_{fg} T_{in} r_{fg} \tag{A.3}$$

where η_{fg} is the final gear efficiency. The drive torque T_{drive} is counteracted by the brake torque T_{brake} and the resulting torque turns into a drive force through the wheels with radius r_{wheel},

$$F_{drive} = \frac{T_{drive} - T_{brake}}{r_{wheel}} \tag{A.4}$$

and acts on the road to drive the vehicle forward. The force acts towards forces which models the aerodynamic drag, rolling resistance and gravitational force

$$m_{tot}\dot{v}_{vehicle} = F_{drive} - F_{aero} - F_{roll} - F_{grav} \tag{A.5}$$

where m_{tot} is the total mass of the vehicle and $v_{vehicle}$ is the vehicle speed. The total mass of the vehicle, m_{tot}, includes the inertial loads from the powertrain components.

$$m_{tot} = m_{vehicle} + (J_{fg} + J_{powertrain})r_{fg}^2 + 2J_{wheel}r_{wheel}^2 \tag{A.6}$$

where $m_{vehicle}$ is the mass of the vehicle, J_{fg} is the inertia of the final gear, $J_{powertrain}$ is the sum of all powertrain inertias (this is given via the physical interface) and J_{wheel} is the wheel inertia.

The wheel speed can be determined from the vehicle speed as

$$\omega_{wheel} = v_{vehicle}/r_{wheel} \tag{A.7}$$

The aerodynamic drag force can be calculated as

$$F_{aero} = \frac{1}{2}\rho C_d A_{front} v_{vehicle}^2 \tag{A.8}$$

where ρ is the air density, C_d is the drag coefficient and A_{front} is the frontal area of the vehicle. The rolling resistance is computed from the normal load as

$$F_{roll} = fm_{vehicle}g\text{sign}(v_{vehicle}) \tag{A.9}$$

where f is the fraction of the normal load that corresponds to rolling resistance. The *sign*-function is included in order to handle the case of zero speed. If gravitational forces are considered then the rolling resistance becomes

$$F_{roll} = fm_{vehicle}g\text{sign}(v_{vehicle})\cos(\alpha) \tag{A.10}$$

where α is the road slope. The gravitational force is

$$F_{grav} = m_{vehicle}g\sin(\alpha)) \tag{A.11}$$

The gravitational load can be position or time based.

A.3   Driver

The driver model was prepared by following a modular approach and therefore contains different sub-modules. The model illuminated in Figure A.3 is capable of running a vehicle equipped with either a manual gearbox with accelerator-, brake- and clutch pedal or a vehicle equipped with an automated gearbox where only accelerator- and brake pedal are used. For the manual transmission vehicle the decisions for gear shift maneuvers are taken by the gear selector sub-module. For automated gearboxes this is bypassed but can be engaged also if needed.

The present driver model contains a
a.   Sub-module controlling the vehicle speed (PID controller)

b. Sub-module taking decisions of gear change (based on the VECTO gearshift algorithm, see b.)
c. Sub-module actuating the clutch pedal
d. Sub-module switching signals either a manual or an automated gearbox is used

For specific demands single sub-modules can be easily removed or be planted in OEM specific driver models (e.g. VECTO gear shift module for OEM specific driver model)

a.) The sub-module controlling the vehicle speed is modeled using a simple PID-controller. It takes the reference speed from the driving cycle and compares it to the vehicles actual speed. If the vehicle's speed is to low it uses the accelerator pedal to demand acceleration, and vice versa if the vehicle's speed is too high, the driver uses the brake pedal to demand a deceleration of the vehicle. For vehicles not capable of running the desired speed (e.g. their design speed is lower than the demanded speed during the test run) the controller includes an anti-wind up function of the integral part, which can be also parameterized in the parameter file. If vehicles equipped with a manual transmission gearbox are driven it is considered that the accelerator pedal is not actuated during a gearshift manoeuvre.

b.) As it was agreed by the HDH group the VECTO gearshift algorithms have been included in the driver model in order to provide a gearshift policy primarily for HDH vehicles equipped with a manual transmission gearbox. VECTO stands for European Vehicle Energy Consumption Calculation Tool, which is currently as well in a development phase planned until March 2014. It is intended to be used to calculate the CO_2 emissions of conventional HD vehicles in Europe. The implemented gearshift strategy is based on the definition of shift polygons for up- and downshift maneuvers. Together with a full load torque curve and a negative torque curve they describe the permitted operating range of the system. Crossing the upper shift polygon forces a higher gear, crossing the lower one a lower gear (see figure A.2 below). Since the VECTO tool itself is still under development and not defined fully until now just a first draft version is implemented in the HILS model library. A model including full functionality as well as a comprehensive description will be available when all open issues in the VECTO workgroup are solved and the tool can be transferred fully in the model library.

The input signals needed for the gear selector sub-module to derive an actual gear request currently are
- The actual gear engaged
- The input torque and rotational input speed (if this is transmission input torque or ICE output torque is still an open issue and has to be discussed in the HDH working group)
- Status of the drivetrain (next gear engaged and all clutches closed and synchronized again)
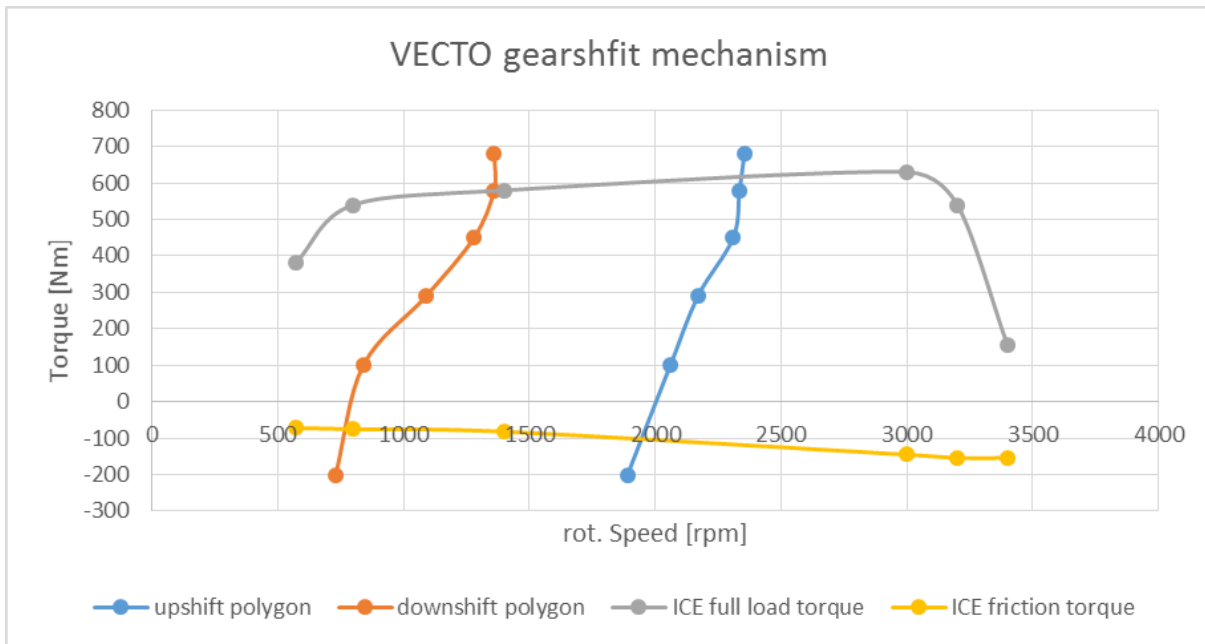
Figure A.2: example of up- and down shift polygons to define the system operating range

Internally also the test cycle and the time of clutch actuation during a shift manoeuvre are loaded in order to detect vehicle starts form standstill and engage the 1st gear on time before the desired speed is greater zero. This allows the vehicle to follow the desired speed within the given limits. The standard output value of the gearshift module when the vehicle stands still is the neutral gear. After a gear is changed a subsequent gear change is suppressed for a parameterized time and as long as the drivetrain is not connected to all propulsion engines and not fully synchronized again. The time limit is rejected and a next gear change is forced if rotational speed limits (lower than ICE idle speed or greater than ICE rated speed multiplied by 1.2) are exceeded.

c.) The sub-module actuating the clutch pedal was designed to actuate the pedal if a vehicle equipped with a manual transmission gearbox is used. Excluding the function from the speed controller sub-module enables the driver model to be used in a wider field of applications. The clutch sub-module is triggered by the gear selector module and actuates the pedal as soon as a gearshift manoeuvre is requested. The clutch module simultaneously forces the speed controller to put the accelerator pedal to zero as long as the clutch is not closed and fully synchronized again after the gearshift manoeuvre. The time of clutch actuation has to be specified in the driver parameter file.

d.) The AT/MT switch enables the driver model to be used either for a vehicle with a manual or an automated gearbox. The output signals for the MT mode are the requested gear and the accelerator-, brake-, and clutch pedal ratios. Using the AT mode the output signals are only accelerator- and brake pedal ratio. No gearshift maneuvers are considered and therefore the accelerator pedal is also not set to zero even though a gear change is detected. The standard values for the clutch pedal ration and for a desired gear are zero in AT mode. Nevertheless, if the gear selection of the actual

test vehicle should be overruled this can be done by enabling the desired gear output in the parameter file.
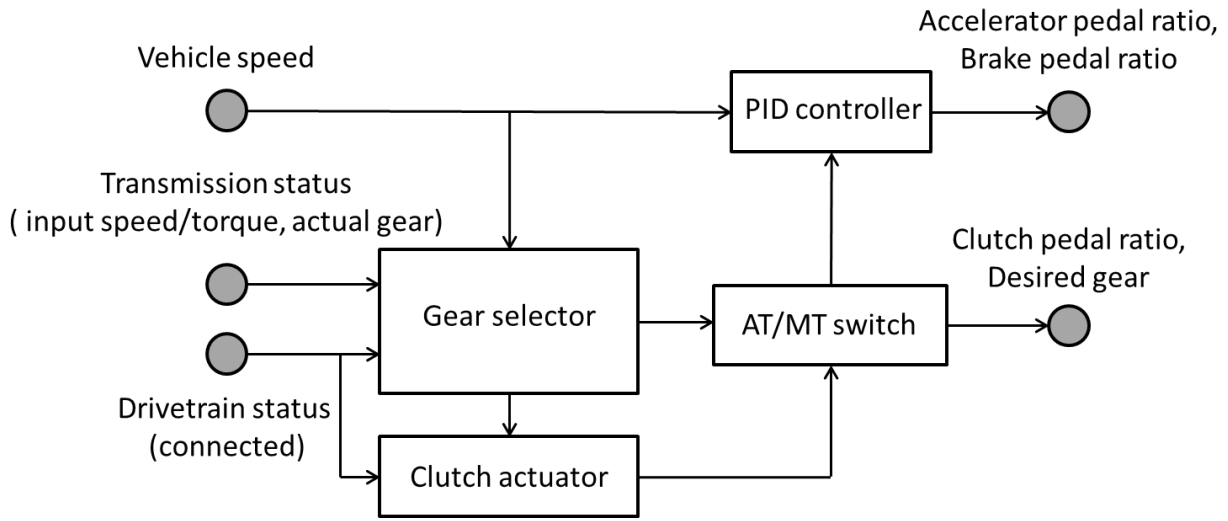
Figure A.3: Block scheme for driver model.

A.4   Electrical components

A.4.1   DCDC converter

The DC/DC converter is a device that changes the voltage level to desired voltage level. The converter model is general and captures the behaviour of several different converters such as Buck, Boost and Buck-Boost converters. As DC/DC converters are dynamically fast compared to other dynamics in a powertrain a simple static model is used:

$$u_{out} \quad = \quad x u_{in} \tag{A.12}$$

where u_{in} and u_{out} are the input voltage and output voltage levels respectively. x is the conversion ratio, i.e. the control signal. The DC/DC converter is controlled via an open-loop controller to the desired voltage, u_{req}, as:

$$x \quad = \quad u_{req}/u_{in} \tag{A.13}$$

Losses are considered to be current losses

$$i_{out} = \begin{cases} i_{in}\eta_{dcdc}/x & i_{in} \leq 0 \\ i_{in}/x\eta_{dcdc} & i_{in} > 0 \end{cases} \tag{A.14}$$

where η_{dcdc} is the DC/DC converter's efficiency.

A.5 Energy converters

A.5.1 Electric machine

An electric machine can generally be divided into two parts, the stator and the rotor. The rotor is the rotating part of the machine. The electric machine is modeled using maps, see Figure A.2. The main reason is that these maps are rather easy to obtain, the model representation becomes accurate, and several different types of machines can be characterized, such as DC-motors, PMSMs and induction machines.
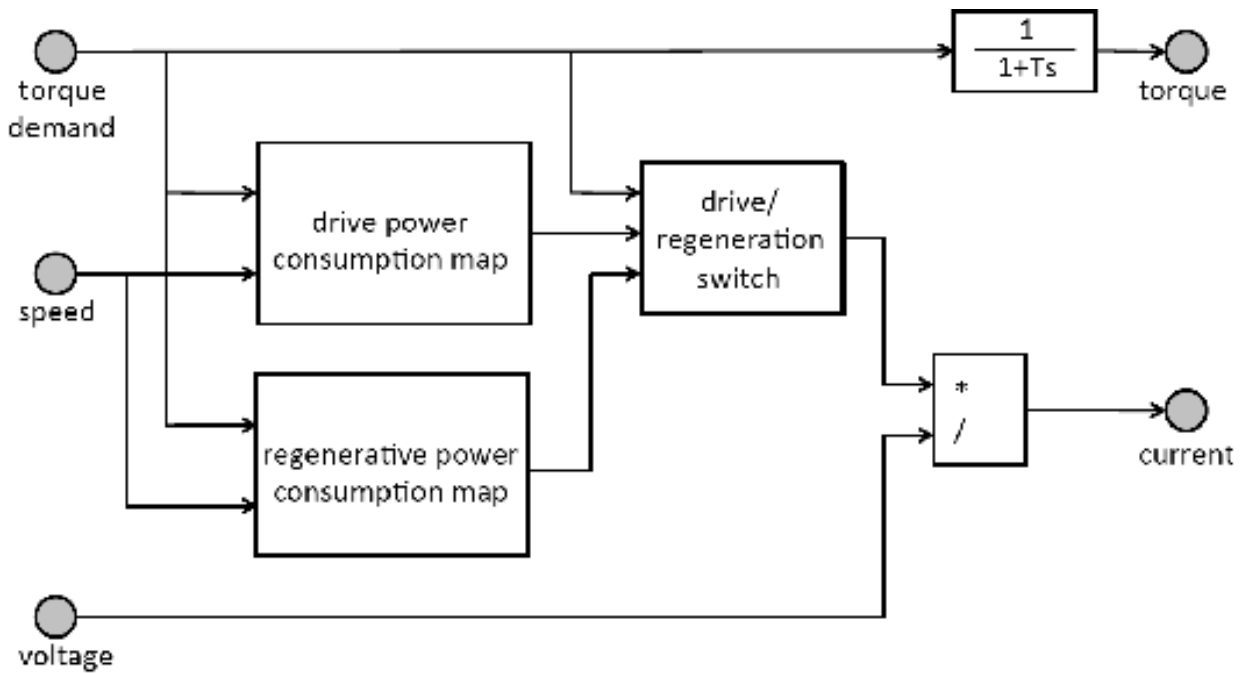
Figure A.2: Block scheme for electric machine component model.

The electric machine dynamics is modeled as a first order system

$$\dot{T}_{em} = -\frac{1}{\tau_1}(T_{em} - T_{em,des})$$

(A.15)

where T_{em} is the machine's torque, $T_{em,des}$ is the desired torque and τ_1 is the electric machine's time constant. The electric power needed to produce the torque at a certain speed is mapped as function of torque and speed

$$P_{el} = f_{em}(T_{em}, \omega_{em})$$

(A.16)

One map is used for positive torque and another map is used for negative torque.

The efficiency of the electric machine can be calculated as

$$\eta_{em} = \frac{T_{em}\omega_{em}}{P_{el}}$$ (A.17)

and the current needed, can be calculated as

$$i = \frac{P_{el}}{u}$$ (A.18)

where i is the current and u is the battery voltage.

The model is complemented with a simple thermodynamical model. The losses in the electric machine can be determined as

$$P_{loss} = P_{el} - T_{em}\omega_{em}$$ (A.19)

The losses are transformed into heat, heating the machine. The temperature for the machine, ϑ_{em}, can be modelled as

$$\dot{\vartheta}_{em} = \frac{1}{\tau_{em,heat}}\left(P_{loss} - (\vartheta_{em} - \vartheta_{cool})/R_{th}\right)$$ (A.20)

where $\tau_{em,heat}$ is the time constant for the thermal mass of the machine and ϑ_{cool} is the machine's cooling media temperature. R_{th} is the machine's thermic resistance.

The electric machine can be torque or speed controlled. The physical model is complemented with a local controller. The speed controller is a PI-controller, while the torque controller is an open-loop (feed-forward) controller.

A.5.2   *Hydraulic Pump/Motor*

A hydraulic pump/motor is a device that converts the energy stored in the accumulator to mechanical energy.

The pump/motor torque is, in general, given as:

$$T_{pm} = xD(p_{acc} - p_{res})\eta_m^*$$ (A.21)

where T_{pm} is the torque, x is the control signal, between 0-1, D is the pump's displacement, p_{acc} and p_{res} are the hydraulic pressure in the accumulator and the reservoir respectively and η_m is the mechanical efficiency. The mechanical efficiency is

$$\eta_m^* = \begin{cases} \eta_m & T_{pm} > 0 \\ \frac{1}{\eta_m} & T_{pm} \leq 0 \end{cases}$$ (A.22)

and consists of friction losses, hydrodynamic losses and viscous losses

$$\eta_m = f(x, p_{acc}, p_{res}, \omega_{pm})$$ (A.23)

where ω is the pump/motor's speed. The efficiency can be determined by experiments.

The volumetric flow through the pump/motor is given as

$$Q_{pm} = xD\omega_{pm}\eta_v^*$$
(A.24)

where Q_{pm} is the volumetric flow and η_v^* is the volumetric efficiency. The volumetric efficiency is

$$\eta_v^* = \begin{cases} \frac{1}{\eta_v} & T_{pm} > 0 \\ \eta_v & T_{pm} \le 0 \end{cases}$$
(A.25)

and consists of laminar losses, turbulent losses and compressibility losses. The efficiency can be determined from measurements and mapped as function of the control signal, the pressure difference of the pump/motor and the speed as

$$\eta_v = f(x, p_{acc}, p_{res}, \omega_{pm})$$
(A.26)

The control signal, x, is as mentioned before a signal between 0 and 1. In order to make it more general the model is complemented with a controller. The pump/motor can be torque or speed controlled. The speed controller is a PI-controller, while the torque controller is an open-loop feed-forward controller.

A.5.3 Internal combustion engine (ver1)

The internal combustion engine is also an energy converter as the electric machine. For the combustion engine chemical energy is converted to mechanical energy. Compared to the electric machine can combustion engines only convert energy in one direction. The internal combustion engine is modeled in a similar way as the electric machine, see Figure A.3.
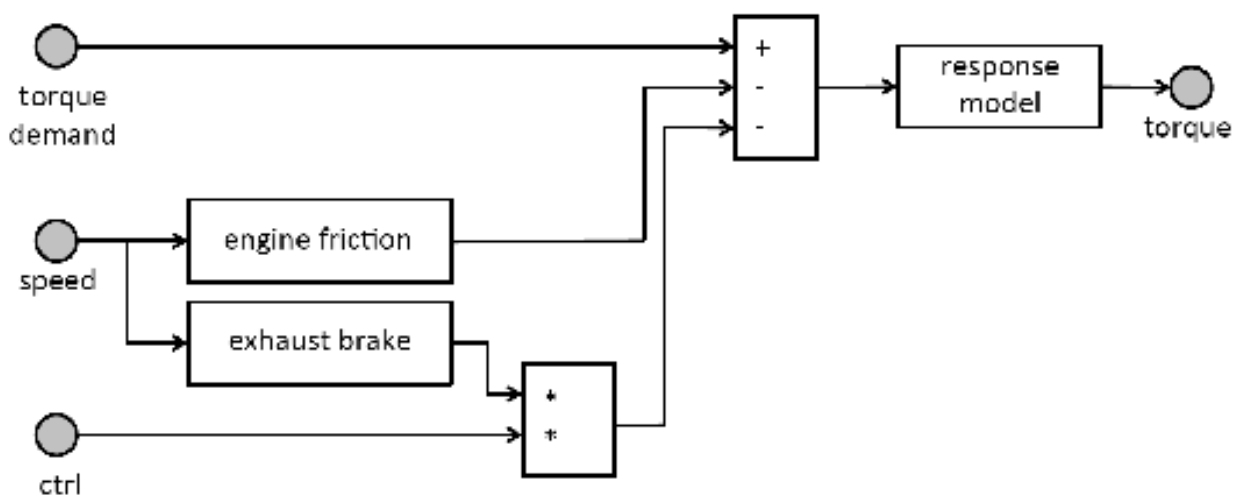
Figure A.3: Block scheme for internal combustion engine component model.

The torque build-up of the internal combustion engine is modeled in a similar way as the electric machine, as a first order system:

$$\dot{T}_{ice} = -\frac{1}{\tau_{ice}}(T_{ice} - T_{ice,des})$$

(A.27)

where T_{ice} is the engine's torque, τ_{ice} is the internal combustion engine's time constant and $T_{ice,des}$ is the demanded engine torque. The demanded torque is the input signal to the system. The model uses the same dynamics independent of engine speed.

The model also includes engine friction, T_{fric}, and exhaust braking, T_{exh}. These are modeled as function of engine speed, and are implemented as maps. The exhaust brake can be controlled, i.e. on or off.

The model is also complemented with a simple thermodynamic model. A thermodynamic model for the combustion engine is important to include if cold start is to be included in the test procedure, and especially if different control strategies are used during cold operation and normal operation. As the engine is equipped with its own cooling system, the thermodynamical model for the engine is only covering the heating of the engine. When the engine reaches it's normal operating temperature, the cooling system starts controlling the temperature, keeping it more or less constant. The heating of the engine is modeled as a limited integrator:

$$\vartheta_{ice,oil} = \max(\int_0^t \eta P_{ice,loss}d\tau + \vartheta_{ice,oil,cold}, \vartheta_{ice,oil,hot})$$

(A.28)

where $\vartheta_{ice,oil}$ is the engine's oil temperature, $P_{ice,loss}$ is the engine's power loss, η is the amount of the power loss that goes to heating the engine, $\vartheta_{ice,cold}$ is the engine's temperature at start of use and $\vartheta_{ice,oil,hot}$ is the engine's normal operating temperature. The model can be calibrated using the tunable parameter η. The integral part of the model corresponds to engine heating due to usage, the limit, set by $\vartheta_{ice,oil,hot}$, corresponds to the case when the cooling system is controlling the temperature. A similar model is also used for modeling the cooling fluid temperature.

The internal combustion engine can be torque or speed controlled. The physical model is complemented with a local controller. The speed controller is a PI-controller, while the torque controller is an open-loop (feed-forward) controller.

A.5.4 *Internal combustion engine (ver2)*

The internal combustion engine modeled is also available in a second version. The only difference between version 1 and version 2 is the engine torque response model. Because of the turbo dynamics a fixed first order linear system model, might not be accurate enough. Instead, a simple speed-dependent torque response model is proposed:

$$\begin{aligned}
\dot{T} &= -\frac{1}{\tau_{ice}(\omega_{ice})}(T - T_{ice,des1}(\omega_{ice})) \\
T_{ice} &= T + T_{ice,des2}(\omega_{ice})
\end{aligned} \tag{A.29}$$

where $\tau_{ice}(\omega_{ice})$ is the engine's time constant. The time constant is dependent on engine speed, ω_{ice}. The demanded torque $T_{ice,des}$ is divided into two parts, one dynamic term, $T_{ice,des1}(\omega_{ice})$, and one direct feed through term, $T_{ice,des2}(\omega_{ice})$. It should be noted that the demanded torque is dependent of speed as well. Furthermore, if the demanded torque is less than the direct feed through term, no dynamic term is needed to capture the engine torque response, i.e. the engine torque is available instantaneously. The time constant and the division of the two parts of the demanded torque are mapped as function of speed.

A.6   Mechanical components

A.6.1   Clutch

A simple model of a clutch. The working principle behind the clutch is that if the clutch is closed then the input torque T_{in} is transferred to the output torque T_{out}. If the clutch is open, the input shaft spins freely and no torque is transferred.
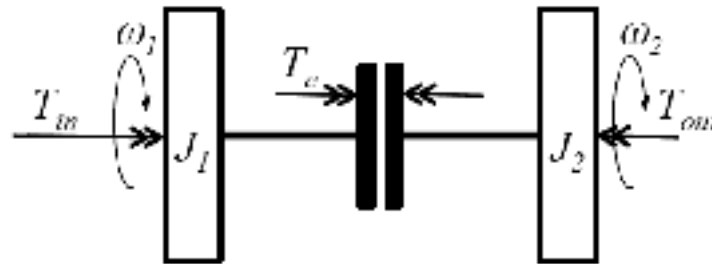
Figure A.4: Simple clutch model.

The equations of motion for the clutch, with notation according to Figure A.4:

$$\begin{aligned}
J_1\dot{\omega}_1 &= T_{in} - T_c \\
J_2\dot{\omega}_2 &= T_c - T_{out}
\end{aligned} \tag{A.30}$$

The clutch is working in three different phases; closed, open or in between closed and open, slipping. When the clutch is open, $T_c = 0$, and during slipping

$$\begin{aligned}
T_c &= uT_{maxtorque}\,sign(\omega_1 - \omega_2) \\
\omega_1 &= \omega_2\,|_{t=0} + \int_0^t (T_{in}(\tau) - T_c(\tau))d\tau
\end{aligned} \tag{A.31}$$

where $T_{maxtorque}$ is the maximum torque that is to be transferred through the clutch and u is the control signal, between 0 and 1, where 0 means disengaged or open and 1 means engaged or closed. When the clutch is closed the $T_{in} \equiv T_{out}$.

A.6.2   *Continuous Variable Transmission (CVT)*
A conventional mechanical transmission can usually take a finite number of different numbers of gear ratios. In contrast, a CVT is a mechanical transmission that can take an infinite number of gear ratios.

The block structure is the same as used in transmission model and the same as used in Kokujikan No.281.

Given the definition of a fixed gear ratio, the output torque of a gear can be calculated as

$$T_{out} = N_{CVT} T_{in} \eta_{CVT} \tag{A.32}$$

where T_{out} is the output torque from the transmission, N_{CVT} is the gear ratio, T_{in} is the input torque and η_{CVT} is the efficiency of the CVT. The efficiency is dependent on input torque, speed and gear ratio:

$$\eta_{CVT} = f(N_{CVT}, T_{in}, \omega_{out}) \tag{A.33}$$

where ω_{out} is the output speed (or the feedback speed). If only torque losses are assumed in the CVT, the transmission speed can be determined as

$$\omega_{in} = N_{CVT} \omega_{out} \tag{A.34}$$

These equations are actually valid for all types of gears, the main difference between a fixed gear and a CVT is the fact that the gear ratio can be changed continuously instead of in steps at discrete time instances. This means that the gear ratio, N_{CVT}, can be controlled, both in timing and in magnitude. The actuator for controlling the gear ratio can be assumed to be represented by a first order system:

$$\frac{d}{dt} N_{CVT} = \frac{1}{\tau_{CVT}} (-N_{CVT} + N_{des}) \tag{A.35}$$

where τ_{CVT} is the actuator time constant and N_{des} is the desired gear ratio.

A.6.3   *Flywheel*
A flywheel is also a basic mechanical component that is needed to be included in some model to create rotational speed. See Section A.7.3 for modeling details.

A.6.4   *Mechanical connection*
This component is used to connect two input shafts. Each shaft is connected through gears. The output torque is calculated as

$$T_{out} = \eta_{out} r_{out} (\eta_{in,1} T_{in,1} r_{in,1} + \eta_{in,2} T_{in,2} r_{in,2}) \tag{A.36}$$

where $T_{in,i}$, $i \in 1,2$ are the torques on the input shafts respectively, $r_{in,i}$ is the input shaft gear ratio, $\eta_{in,i}$ is the efficiency, T_{out} is the output torque, r_{out} is the output shaft gear ratio and η_{out} is the output gear efficiency. Each shaft/gear has its own inertia which is added to the total inertia.

A.6.5   *Retarder*
The retarder is a braking device used as a complement to the service brakes. A retarder is usually a fluid dynamic device. A simple torque loss model is proposed to capture the function of the retarder. Furthermore, it is speed dependent as the effectiveness of the retarder decreases with speed.

$$T_{out} = T_{in} - u T_{loss}(\omega)$$
(A.37)

where *u* is a control signal to turn the retarder on or off and T_{loss} is the retarder brake torque.

A.6.6   *Spur gear*
The spur gear is modeled as two cogwheels in contact, with a ratio of r_{spur}

$$\omega_{out} = \omega_{in}/r_{spur}$$
(A.38)

Losses η_{spur} for the spur gear is considered to be torque losses, meaning that T_{out} is actually calculated as

$$T_{out} = \begin{cases} T_{in}r_{spur}\eta_{spur} & T_{in} \leq 0 \\ T_{in}r_{spur}/\eta_{spur} & T_{in} > 0 \end{cases}$$
(A.39)

The total inertia depends on the gear ratio as:

$$J_{out} = J_{in}r_{spur}^2 + J_{spur}$$
(A.40)

A.6.7   *Torque converter*
A torque converter is a widely used powertrain component, mainly in combination with automatic shift transmissions. The basic function is torque multiplication. The working principle is that power is transmitted from the impeller or pump to the turbine via the working fluid movement, see Figure A.5. The torque multiplication is done by the stator, which changes the angular momentum of the fluid between the turbine exit side and the impeller entrance side. If no stator is used, a torque converter works as a fluid coupling with no torque multiplication.
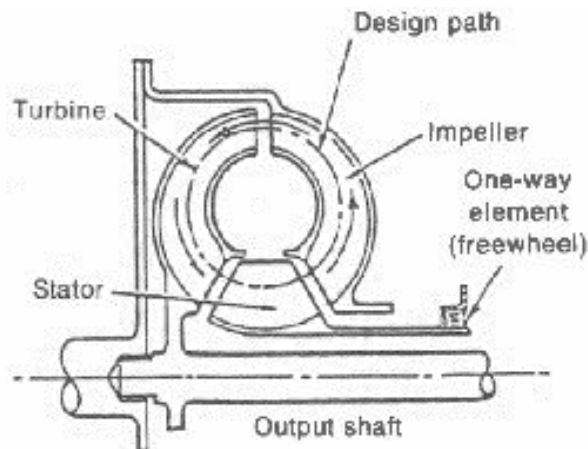
Figure A.5: Torque converter picture.

The torque converters ingoing speed can be determined by treating the ingoing shaft and the impeller as an inertia:

$$(J_{in} + J_{impeller})\frac{d}{dt}\omega_{in} = T_{in} - T_{out} \tag{A.41}$$

Torque converter characteristics are usually expressed in terms of speed and torque ratios between ingoing and outgoing speed and torque respectively, see Figure A.6.
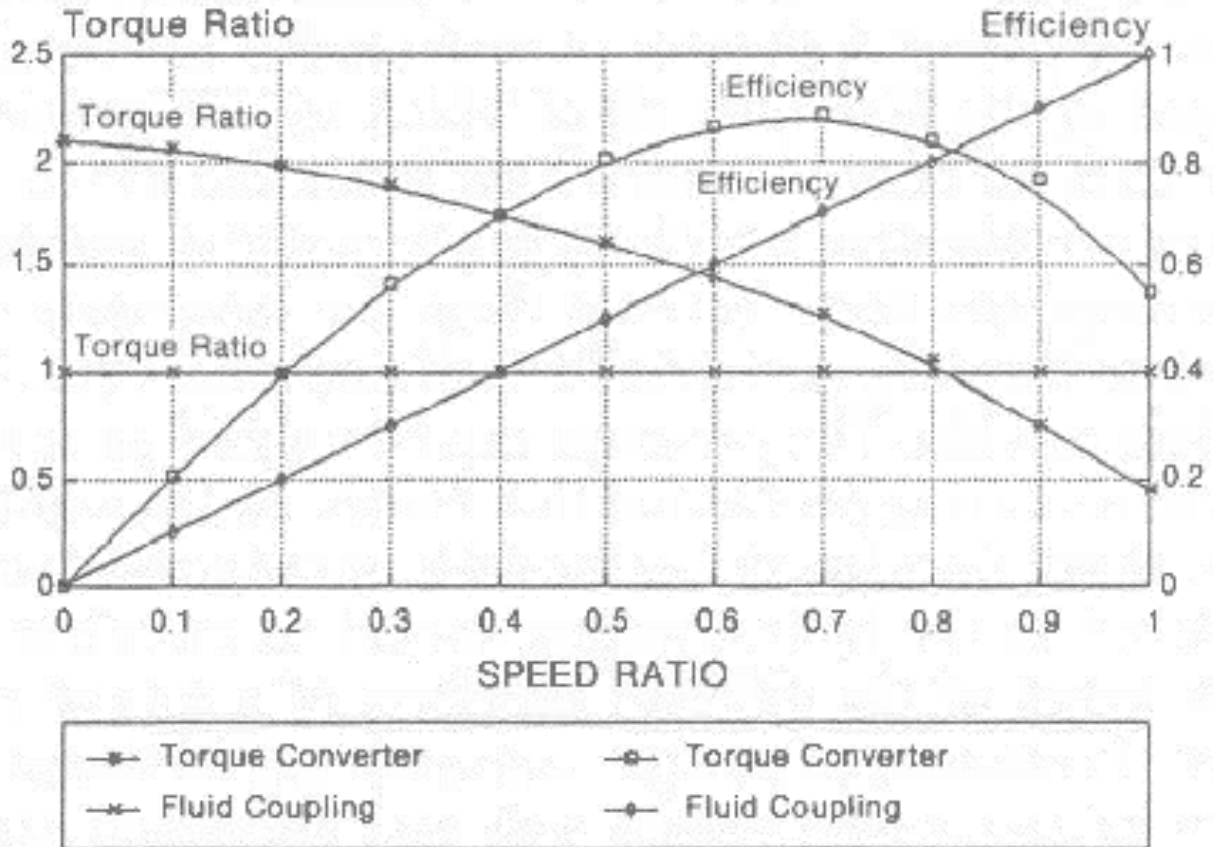
Figure A.6: Torque converter characteristics (example).

The speed ratio, ω_r, and the torque ratio, T_r, is defined as:

$$\begin{aligned} \omega_r &= \omega_{out}/\omega_{in} \\ T_r &= T_{out}/T_{in} \end{aligned} \tag{A.42}$$

As the ingoing torque, T_{in} and the outgoing speed (or feedback speed), ω_{out} are known, together with (A.41) and (A.42) the outgoing torque, T_{out} can be determined. In Figure A.7 a schematic picture of the torque converter model is presented.
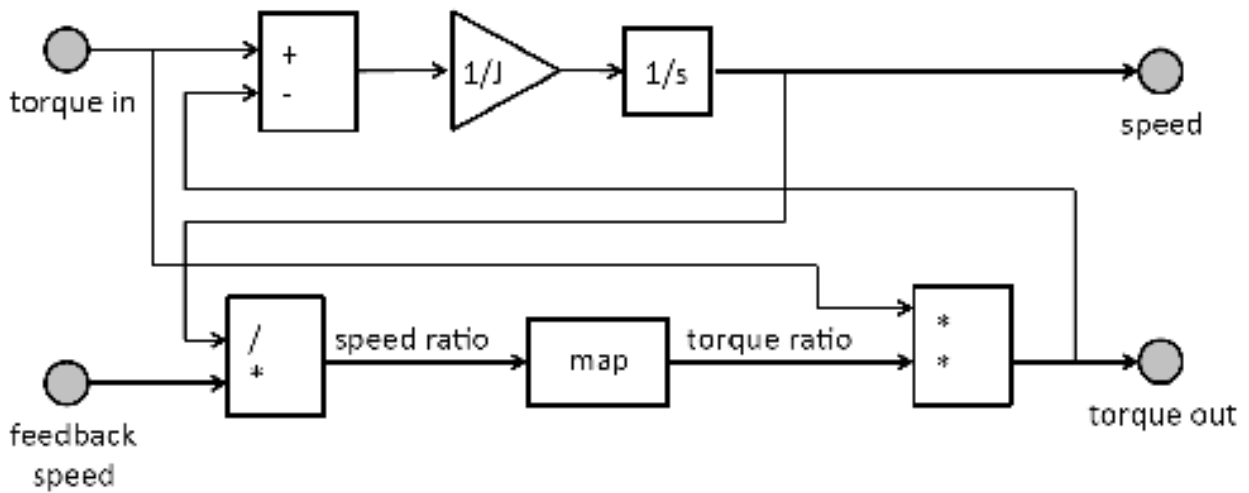
Figure A.7: Torque converter, block scheme model.

A.6.8   Transmission

The transmission is modeled as two gears in contact, with a ratio of r_{gear}

$$\omega_{out} \quad = \quad \omega_{in}/r_{gear} \tag{A.43}$$

Losses for the gearbox is considered to be torque losses, meaning that T_{out} is actually calculated as

$$T_{out} = \begin{cases} T_{in}r_{gear}\eta_{gear} & T_{in} \leq 0 \\ T_{in}r_{gear}/\eta_{gear} & T_{in} > 0 \end{cases} \tag{A.44}$$

Losses are given for each gear.

The total gearbox inertia depends on the active gear:

$$J_{out} = J_{in}r_{gear}^2 + J_{gearbox} \tag{A.45}$$

The model also includes a clutch in order to get a torque interrupt. The number of gears is set by a parameter.

A.7   Rechargeable energy storage systems

A.7.1   Battery (Resistor model)
The battery is modeled using a resistor model, see Figure A.8.
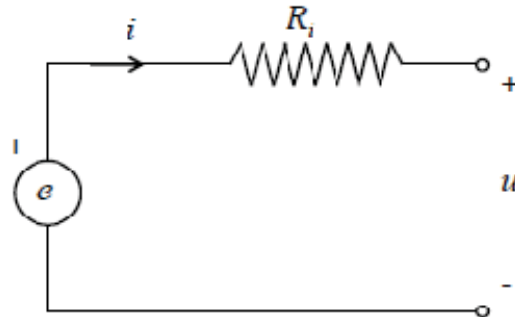
Figure A.8: Simple battery model.

The battery voltage can be determined from Kirchhoff's law as

$$u = e - R_i i \tag{A.46}$$

The open circuit voltage e and the internal resistance R_i are depending of energy level in the battery, state-of-charge SOC. The dependency is modeled using tabulated values in maps. State-of-charge is defined as

$$SOC = SOC(0) - \int_0^t \frac{i}{3600C} d\tau \tag{A.47}$$

where C is the batter capacity. The battery is scalable via the number of cells used, n_s number of battery cells in series and n_p number of cells in parallel.

In Figure A.9, a schematic picture of the battery model is presented.
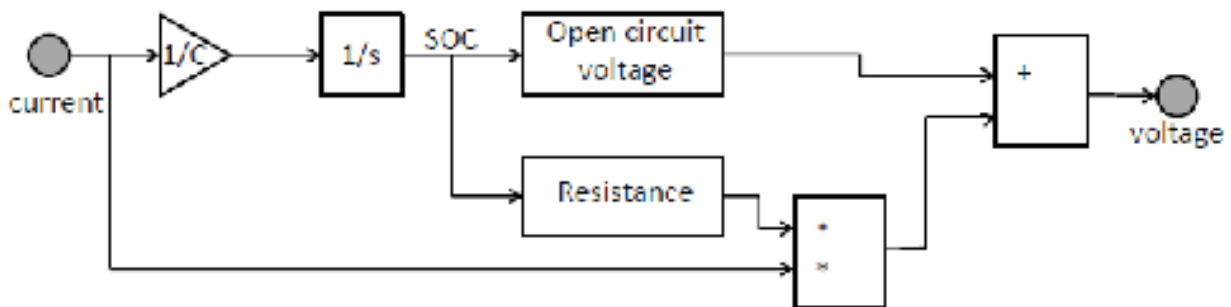
Figure A.9: Battery model, single cell.

The same model can be used to simulate a *super capacitor*. Just set the open circuit voltage to to linearly increase with *SOC*. The slope should correspond to the capacity of the super capacitor.

The battery model also includes a thermodynamical model. The thermodynamics is model in the same way as for the electric machine. The losses in a battery cell are mainly resistive losses:

$$P_{loss} = R_i i^2$$

(A.48)

The losses transforms into heat, heating the battery cell. The temperature for the battery system, ϑ_{bat}, can be modelled as

$$\dot{\vartheta}_{bat} = \frac{1}{\tau_{bat,heat}}(P_{loss} - (\vartheta_{bat} - \vartheta_{cool})/R_{th})$$

(A.49)

where $\tau_{bat,heat}$ is the time constant for the thermal mass of the battery and ϑ_{cool} is the battery's cooling media temperature. R_{th} is the battery's thermal resistance.

A.7.2 *Battery (RC model)*

An alternative model including some additional dynamics is also available. The battery is modeled using a resistor and an RC circuit, see Figure A.10.
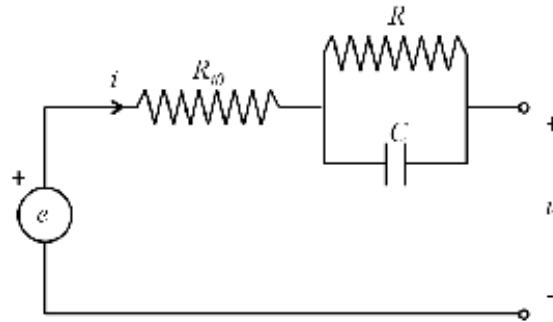
Figure A.10: RC circuit battery model.

The battery voltage can be determined from Kirchhoff's law as

$$u = e - R_{i0}i - u_{RC}$$

(A.50)

where u_{RC} is the voltage over the RC circuit. The voltage u_{RC} can be determined using Kirchhoff's law, Ohm's law and the relation for a capacitor as:

$$\frac{d}{dt}u_{RC} = -\frac{1}{RC}u_{RC} + \frac{1}{C}i$$

(A.51)

The open circuit voltage e, the resistances R_{i0} and R and the capacitance C are depending on state-of-charge SOC. The dependency is modeled using tabulated values in maps. The battery is scalable via the number of cells used.

A.7.3 Flywheel

A flywheel is basically a rotating mass, which can store kinetic energy as

$$E_{fly} = J_{fly}\omega_{fly}^2 \tag{A.52}$$

where E_{fly} is the energy stored in the flywheel, J_{fly} is the inertia of the flywheel and ω_{fly} is the rotating speed of the flywheel.

The block describing the model structure for the flywheel is presented in Figure A.11. The block takes a torque as input and the output is the rotational speed of the flywheel.
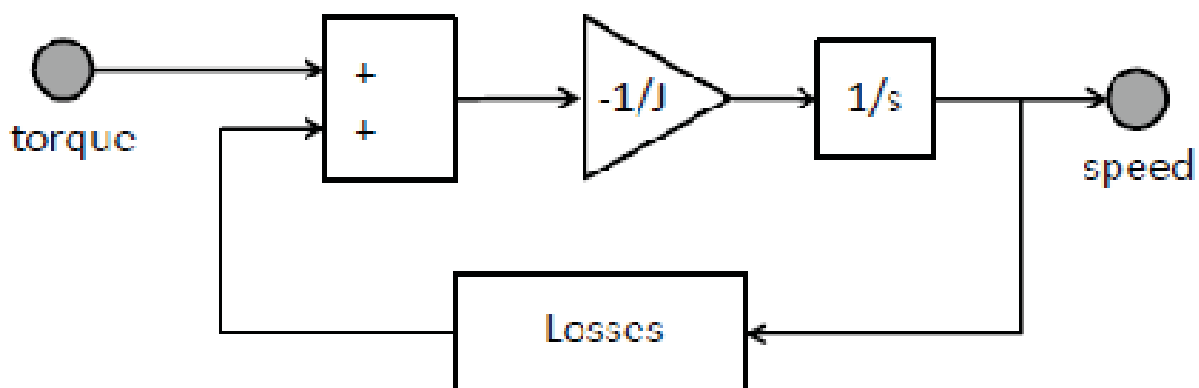
Figure A.11: Flywheel.

The model of flywheels can be derived using Newton's second law:

$$J_{fly}\frac{d}{dt}\omega_{fly} = -T_{in} - T_{loss}(\omega_{fly}) \tag{A.53}$$

where T_{in} is the input torque and $T_{loss}(\omega_{fly})$ is the loss torque. The loss torque is dependent on the speed of the flywheel. The loss torque can be determined from measurement data.

A.7.4 Accumulator

An accumulator is a pressure vessel that is used to store a medium (fluid or gas) in a high-pressure portion of the system. A hydraulic system consists of at least two accumulators, one high-pressure accumulator, used for storing energy, and one low pressure accumulator, used as a reservoir. When the accumulator is empty, all fluid is in the reservoir. As fluid flows in and out of the accumulator, the *charge gas* acts as a spring storing potential energy.

The volume occupied by the fluid or the medium is

$$\frac{d}{dt}V_f = Q \tag{A.54}$$

where V_f is the volume of the fluid or the medium, and Q is the volume flow to or from the accumulator. Q is positive if the flow is into the accumulator. The hydraulic accumulator is divided into two parts, the fluid part and the charge gas part, see Figure A.12.
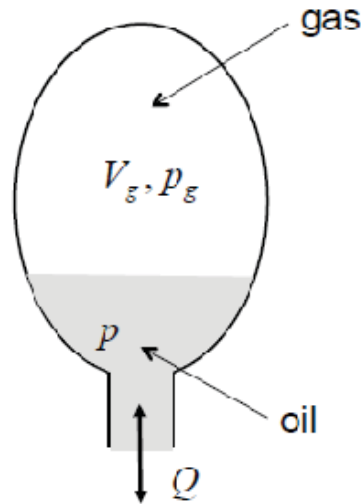
Figure A.12: Hydraulic accumulator.

The parts are separated by a piston, bladder or diaphragm. If V is the accumulator volume, the volume occupied by the charge gas is $V_g = V - V_f$. As the volume of the accumulator is fixed, this means that the charge gas volume is given as:

$$\frac{d}{dt}V_g = -Q \tag{A.55}$$

Using the ideal gas law, $pV = mR\vartheta$, the gas pressure can be determined as

$$p_g = \frac{m_g R \vartheta_g}{V_g} \tag{A.56}$$

where m_g is charge gas mass, R is the gas constant and ϑ_g is the temperature of the gas. A simple assumption is that the gas pressure is approximately equal to the fluid pressure, $p \approx p_g$, this means that there are no pressure losses. Furthermore, if no heat transfer to the surrounding is assumed the hydraulic pressure in the accumulator is given as:

$$p = \frac{m_g R \vartheta}{V_g} \tag{A.57}$$

As mentioned, this assumption means that there are no losses in the accumulator. A simple heat transfer model can rather easily be introduced to handle the case if the model is not accurate enough:

$$m_g c_v \frac{d}{dt}\vartheta_g = -\frac{m_g R\vartheta_g}{V_g}Q - hA_w(\vartheta_g - \vartheta_w) \qquad (A.58)$$

where c_v is the charge gas specific volume, h is the heat transfer coefficient of the accumulator, A_w is the accumulator's wall area and ϑ_w is the accumulator wall temperature. The accumulator model has then two dynamic states, the volume and the temperature. The pressure p is still determined through (A.57). This model describes the accumulator dynamics with a simple loss model.

A reservoir can be modeled in the same way.

B   Interface signals

This appendix shows how the Simulink implementation structure, the parameter data and the model equations are related between each other.

B.1   Electrical Auxiliary Systems

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
$P_{el,aux}$	W	Auxiliary system load	dat.auxiliaryload.value

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd	x	Aux_flgOnOff_B	Turn auxiliary system on-off (flag)	0/1

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	i_{aux}	Aux_iAct_A	Auxiliary system current	A

Physical interfaces

Electrical interface:

Node	Variable name	Name	Description	Unit
elec in [V]	u	phys_voltage_V	voltage	V
elec fb out [A]	i_{aux}	phys_current_A	current	A

B.2   *Mechanical Auxiliary Systems*

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
$P_{mech,aux}$	W	Auxiliary system load	dat.auxiliaryload.value
J_{aux}	kgm^2	Inertia	dat.inertia.value

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd	x	Aux_flgOnOff_B	Turn auxiliary system on-off (flag)	0/1

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	T_{out}	Aux_tqAct_A	Auxiliary system torque	Nm

Physical interfaces

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech in [Nm]	T_{out}	phys_torque_Nm	torque	Nm
	J_{aux}	phys_inertia_kgm2	inertia	kgm^2
mech fb out [rad/s]	ω	phys_speed_radps	speed	rad/s

B.3 Chassis

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
$m_{vehicle}$	kg	Vehicle mass	dat.vehicle.mass.value
r_{fg}	-	Final gear ratio	dat.fg.ratio.value
η_{fg}	-	Final gear efficiency	dat.fg.efficiency.value
J_{fg}	kgm^2	Final gear inertia	dat.fg.inertia.value
A_{front}	m^2	Vehicle front area	dat.aero.af.value
C_d	-	Drag coefficient	dat.aero.cd.value
r_{wheel}	m	Wheel radius	dat.wheel.radius.value
J_{wheel}	kgm^2	Wheel inertia	dat.wheel.inertia.value
f	-	Rolling resistance coefficient	dat.wheel.rollingres.value

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd	T_{brake}	Chassis_tqBrakeReq_Nm	Requested brake torque	Nm

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	$v_{vehicle}$	Chassis_vVehAct_mps	Actual vehicle velocity	m/s
	ω_{wheel}	Chassis_nWheelAct_radps	Actual wheel speed	rad/s
	m_{tot}	Chassis_massVehAct_kg	Vehicle mass	kg
	α	Chassis_slopRoad_rad	Road slope	rad

Physical interfaces

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech in [Nm]	T_{drive}	phys_torque_Nm	torque	Nm
	$J_{powertrain}$	phys_inertia_kgm2	inertia	kgm^2
mech fb out [rad/s]	ω_{wheel}	phys_speed_radps	rotational speed	rad/s

B.4 Driver*

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
-	-	select gearbox mode - MT(1) or AT(0)	dat.gearboxmode.value
-	-	VECTO gear selection model	dat.gearselectionmode.value
-	s	clutch time	dat.clutchtime.value
-	m/s	Clutch is automatically actuated if speed is below this value	dat.clutchthreshold.value
-	-	Driver PID controller	dat.controller

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
driver cmd out		Drv_AccPedl_rat	Accelerator pedal position	[0 – 1]
		Drv_BrkPedl_rat	Brake Pedal position	[0 - 1]
		Drv_CluPedl_Rat	Clutch pedal position	[0 – 1]
		Drv_nrGearReq	Gear request	-

	Drivecycle_RefSpeed_mps	Reference speed	m/s

The following signals are needed for making the model to work:

Node	Name	Description	Unit
sensor in	Chassis_vVehAct_mps	Chassis speed	m/s
	Transm_nInAct_radps	Transmission speed in	rad/s
	Transm_tqInAct_Nm	Transmission torque in	Nm
	Transm_grGearAct	Actual gear ratio	-
	Transm_flgConnected_B	Is the transmission engaged or disengaged	Boolean
	Clu_flgConnected_B	Is the clutch engaged or disengaged	Boolean

*Since the VECTO gear shift algorithm could not be implemented fully yet and though no model test runs for manual transmission vehicles actuated by the VECTO gearshift logics could be performed, the model will not be documented in detail here before it is finished and tested.

B.5 DCDC converter

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
η_{dcdc}	-	Efficiency	dat.efficiency.value

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd	u_{req}	Dcdc_uReq_V	Requested output voltage	V

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	u_{out}	Dcdc_uAct_V	Actual output voltage	V

Physical interfaces

Electrical interface:

Node	Variable name	Name	Description	Unit
elec in [V]	u_{in}	phys_voltage_V	voltage	V
elec out [V]	u_{out}	phys_voltage_V	voltage	V
elec fb in [A]	i_{out}	phys_current_A	current	A
elec fb out [A]	i_{in}	phys_current_A	current	A

B.6   *Electric machine*

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
J_{em}	kgm^2	Inertia	dat.inertia.value
τ_1	-	Time constant	dat.timeconstant.value
-	Nm	maximum torque =f (speed)	dat.maxtorque
-	Nm	minimum torque =f (speed)	dat.mintorque
-	-	Speed controller (PI)	dat.ctrl
f_{em}	W	power map = f(speed, torque)	dat.elecpowmap
-	kg/s	mass flow cooling fluid	dat.mflFluid
$T_{em,heat}$	J/K	Thermal capacity	dat.cm.value
R_{th}	K/W	Thermal resistance	dat.Rth.value
-	-	Properties of the cooling fluid	dat.coolingFluid

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd		ElecMac_nReq_radps	Requested speed	rad/s
		ElecMac_flgReqSwitch_B	Switch - speed/torque	Boolean
		ElecMac_tqReq_Nm	Requested torque	Nm

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	T_{em}	ElecMac_tqActAct_V	Actual machine torque	Nm
	ω_{em}	ElecMac_nAct_radps	Actual machine speed	rad/s
	i	ElecMac_iAct_A	Current	A
	ϑ_{em}	ElecMac_tAct_K	Machine temperature	K

Physical interfaces

Electrical interface:

Node	Variable name	Name	Description	Unit
elec in [V]	u	phys_voltage_V	voltage	V
elec fb out [A]	i	phys_current_A	current	A

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech out [Nm]	T_{em}	phys_torque_Nm	torque	Nm
	J_{em}	phys_inertia_kgm2	inertia	kgm^2
mech fb in [rad/s]	ω_{em}	phys_speed_radps	rotational speed	rad/s

B.7   Hydraulic pump/motor

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
J_{pm}	kgm²	Inertia	dat.inertia.value
τ_1	-	Time constant	dat.timeconstant.value
D	m³	Displacement volume	dat.displacement.value
η_v	-	volumetric efficiency	dat.volefficiency
η_m	-	mechanical efficiency	dat.mechefficiency
-	-	PI controller	dat.ctrl

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd		Hpm_nReq_radps	Requested speed	rad/s
		Hpm_flgReqSwitch_B	Switch - speed/torque	Boolean
		Hpm_tqReq_Nm	Requested torque	Nm

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	T_{pm}	Hpm_tqActAct_V	Actual machine torque	Nm
	ω_{pm}	Hpm_nAct_radps	Actual machine speed	rad/s
	Q_{pm}	Hpm_flowAct_m3ps	Actual volumetric flow	m³/s
	p_{acc}	Hpm_pInAct_Pa	Accumulator pressure	Pa
	p_{res}	Hpm_pOutAct_Pa	Reservoir pressure	Pa

Physical interfaces

Fluid interface:

Node	Variable name	Name	Description	Unit
fluid in 1 [Pa]	p_{acc}	phys_pressure_Pa	pressure	Pa
fluid in 2 [Pa]	p_{res}	phys_pressure_Pa	pressure	Pa
fluid fb out [m3/s]	Q_{pm}	phys_flow_m3ps	volume flow	m³/s

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech out [Nm]	T_{pm}	phys_torque_Nm	torque	Nm
	J_{pm}	phys_inertia_kgm2	inertia	kgm²
mech fb in [rad/s]	ω_{pm}	phys_speed_radps	rotational speed	rad/s

B.8   *Internal combustion engine*

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
J_{ice}	kgm²	Inertia	dat.inertia.value
τ_{ice}	-	Time constant	dat.timeconstant.value
T_{fric}	Nm	engine friction torque	dat.friction
T_{exh}	Nm	exhaust brake torque	dat.exhaustbrake
-	Nm	maximum torque	dat.maxtorque
-	-	PI controller (speed)	dat.ctrl
-	kg/s	fuel flow	dat.fuelmap
-	kJ/kg	net calorific value of fuel	dat.ncv.value
-	kg/m³	fuel density	dat.rho.value
-	-	power loss to cooling and oil	dat.eta.value
-	-	Properties of oil	dat.oil
-	-	Properties of the cooling fluid	dat.cf

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd		Eng_nReq_radps	Requested speed	rad/s
		Eng_flgReqSwitch_B	Switch - speed/torque	Boolean
		Eng_tqReq_Nm	Requested torque	Nm
		Eng_flgExhaustBrake_B	Exhaust brake on/off	Boolean

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	T_{ice}	Eng_tqCrkSftAct_Nm	Crankshaft torque	Nm
	$T_{ice} + T_{fric} + T_{exh}$	Eng_tqIndAct_Nm	Indicated torque	Nm
	ω_{ice}	Eng_nAct_radps	Actual motor speed	rad/s
	ϑ_{ice}	Eng_tOilAct_K	Oil temperature	K

Physical interfaces

Chemical interface:

Node	Variable name	Name	Description	Unit
chem fb out [kg/s]	-	phys_massflow_kgps	fuel flow	kg/s

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech out [Nm]	T_{ice}	phys_torque_Nm	torque	Nm
	J_{ice}	phys_inertia_kgm2	inertia	kgm^2
mech fb in [rad/s]	ω_{ice}	phys_speed_radps	rotational speed	rad/s

B.9 Clutch

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
J_1	kgm^2	Inertia	dat.in.inertia.value
J_2	kgm^2	Inertia	dat.out.inertia.value
$T_{maxtorque}$	-	max torque transmitted	dat.maxtorque.value

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd	*u*	Clu_ratReq_Rt	requested clutch pedal position	ratio (0-1)

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor		Clu_flgConnected_B	Disengaged or not	Boolean

Physical interfaces

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech in[Nm]	T_{in}	phys_torque_Nm	torque	Nm
		phys_inertia_kgm2	inertia	kgm^2
mech out [Nm]	T_{out}	phys_torque_Nm	torque	Nm
		phys_inertia_kgm2	inertia	kgm^2
mech fb in [rad/s]	ω_1	phys_speed_radps	rotational speed	rad/s
mech fb out [rad/s]	ω_2	phys_speed_radps	rotational speed	rad/s

B.10   Continuous variable transmission

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
τ_{CVT}	-	time constant	dat.timeconstant.value
η_{CVT}	-	efficiency	dat.efficiency

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd	N_{des}	CVT_ratGearReq	requested gear ratio	ratio (0-1)

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	N_{CVT}	CVT_ratGearAct_Rt	Actual gear ratio	ratio
	ω_{out}	CVT_nOutAct_radps	output speed	rad/s
	ω_{in}	CVT_nInAct_radps	input speed	rad/s

Physical interfaces

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech in[Nm]	T_{in}	phys_torque_Nm phys_inertia_kgm2	torque inertia	Nm kgm^2
mech out [Nm]	T_{out}	phys_torque_Nm phys_inertia_kgm2	torque inertia	Nm kgm^2
mech fb in [rad/s]	ω_{out}	phys_speed_radps	rotational speed	rad/s
mech fb out [rad/s]	ω_{in}	phys_speed_radps	rotational speed	rad/s

B.11   Flywheel

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
J_{fly}	-	Inertia	dat.inertia.value
T_{loss}	-	loss map	dat.loss

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd		no ctrl signal		

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	N_{CVT}	Flywheel_nAct_radps	speed	rad/s

Physical interfaces

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech in [Nm]	T_{in}	phys_torque_Nm phys_inertia_kgm2	torque inertia	Nm kgm^2
mech fb out [rad/s]	ω_{fly}	phys_speed_radps	rotational speed	rad/s

B.12   *Mechanical connection*

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
J_1	kgm^2	inertia	dat.in1.inertia.value
$r_{in,1}$	-	gear ratio	dat.in1.ratio.value
$\eta_{in,1}$	-	efficiency	dat.in1.efficiency.value
J_2	kgm^2	inertia	dat.in2.inertia.value
$r_{in,2}$	-	gear ratio	dat.in2.ratio.value
$\eta_{in,2}$	-	efficiency	dat.in2.efficiency.value
J_{out}	kgm^2	inertia	dat.out.inertia.value
r_{out}	-	gear ratio	dat.out.ratio.value
η_{out}	-	efficiency	dat.out.efficiency.value

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd		no ctrl signal		

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor		no signal		

Physical interfaces

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech in1 [Nm]	$T_{in,1}$	phys_torque_Nm phys_inertia_kgm2	torque inertia	Nm kgm^2
mech in2 [Nm]	$T_{in,2}$	phys_torque_Nm phys_inertia_kgm2	torque inertia	Nm kgm^2
mech out [Nm]	T_{out}	phys_torque_Nm phys_inertia_kgm2	torque inertia	Nm kgm^2
mech fb in [rad/s]		phys_speed_radps	rotational speed	rad/s
mech fb out1 [rad/s]		phys_speed_radps	rotational speed	rad/s
mech fb out 2[rad/s]		phys_speed_radps	rotational speed	rad/s

B.13 *Retarder*

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
T_{loss}	-	brake torque map	dat.braketorque

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd	*u*	Ret_flgOnOff_B	Retarder on/off	Boolean

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	T_{loss}	Ret_tqBrkAct_Nm	brake torque	Nm

Physical interfaces

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech in[Nm]	T_{in}	phys_torque_Nm phys_inertia_kgm2	torque inertia	Nm kgm²
mech out [Nm]	T_{out}	phys_torque_Nm phys_inertia_kgm2	torque inertia	Nm kgm²
mech fb in [rad/s]		phys_speed_radps	rotational speed	rad/s
mech fb out [rad/s]		phys_speed_radps	rotational speed	rad/s

B.14 *Spur gear*

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model

J_{spur}	kgm²	inertia	dat.in.inertia.value
r_{spur}	-	gear ratio	dat.in.ratio.value
η_{sput}	-	efficiency	dat.in.efficiency.value

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd		no ctrl signal		

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor		no signal		

Physical interfaces

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech in [Nm]	T_{in}	phys_torque_Nm	torque	Nm
		phys_inertia_kgm2	inertia	kgm²
mech out [Nm]	T_{out}	phys_torque_Nm	torque	Nm
	J_{out}	phys_inertia_kgm2	inertia	kgm²
mech fb in [rad/s]	ω_{out}	phys_speed_radps	rotational speed	rad/s
mech fb out [rad/s]	ω_{in}	phys_speed_radps	rotational speed	rad/s

B.15   Torque converter

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
$J_{impeller}$	kgm²	inertia	dat.inertia.value

	-	torque ratio map	dat.torqueratiomap

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd		TC_flgLockUp_B	Torque converter lock up	Boolean

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	ω_{out}	TC_nTurbineAct_radps	Turbine speed	rad/s

Physical interfaces

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech in [Nm]	T_{in}	phys_torque_Nm	torque	Nm
	J_{in}	phys_inertia_kgm2	inertia	kgm²
mech out [Nm]	T_{out}	phys_torque_Nm	torque	Nm
		phys_inertia_kgm2	inertia	kgm²
mech fb in [rad/s]	ω_{out}	phys_speed_radps	rotational speed	rad/s
mech fb out [rad/s]	ω_{in}	phys_speed_radps	rotational speed	rad/s

B.16 Transmission

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
-	s	time to shift	dat.shifttime.value

-	Nm	maximum torque	dat.maxtorque.value
-	-	number of gears	dat.nofgear.value
-	-	gear number [vector]	dat.gear.number.value
r_{gear}	-	gear ratio [vector]	dat.gear.ratio.value
$J_{gearbox}$	kgm^2	inertia [vector]	dat.gear.inertia.value
η_{gear}		gear efficiency [vector]	dat.gear.efficiency.value

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd		Transm_nrGearReq	requested gear number	

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor		Transm_nrGearAct	Actual gear number	
		Transm_flgConnected_B	connected?	Boolean
	ω_{out}	Transm_nOutAct_radps	output speed	rad/s
	ω_{in}	Transm_nInAct_radps	input speed	rad/s

Physical interfaces

Mechanical interface:

Node	Variable name	Name	Description	Unit
mech in[Nm]	T_{in}	phys_torque_Nm	torque	Nm
	J_{in}	phys_inertia_kgm2	inertia	kgm^2
mech out [Nm]	T_{out}	phys_torque_Nm	torque	Nm
	J_{out}	phys_inertia_kgm2	inertia	kgm^2
mech fb in [rad/s]	ω_{out}	phys_speed_radps	rotational speed	rad/s
mech fb out [rad/s]	ω_{in}	phys_speed_radps	rotational speed	rad/s

B.17   *Battery (Resistor model)*

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
n_s	-	number of cells connected in series	dat.ns.value
n_p	-	number of cells connected in parallel	dat.np.value
C	Ah	cell capacity	dat.capacity.value
$SOC(0)$	%	initial state of charge	dat.initialSOC.value
e	V	open circuit voltage = f(SOC)	dat.ocv.ocv
R_i	Ω	cell resistance	dat.resistance.R0

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd		no ctrl signal		

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	i	ReESS_iAct_A	Actual current	A
	u	ReESS_uAct_V	Actual output voltage	V
	SOC	ReESS_socAct_Rt	State of charge	ratio (0-1)
	ϑ_{bat}	ReESS_tAct_K	Battery temperature	K

Physical interfaces

Electrical interface:

Node	Variable name	Name	Description	Unit
elec out [V]	u	phys_voltage_V	voltage	V
elec fb in [A]	i	phys_current_A	current	A

B.18   Battery (RC model)

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
n_s	-	number of cells connected in series	dat.ns.value
n_p	-	number of cells connected in parallel	dat.np.value
C	Ah	cell capacity	dat.capacity.value
$SOC(0)$	%	initial state of charge	dat.initialSOC.value
e	V	open circuit voltage = f(SOC)	dat.ocv.ocv
$R0$	Ω	cell resistance	dat.resistance.R0
R	Ω	number of cells in series	dat.resistance.R
C	F	number of cells in series	dat.restatnce.C

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd		no ctrl signal		

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	i	ReESS_iAct_A	Actual current	A
	u	ReESS_uAct_V	Actual output voltage	V
	SOC	ReESS_socAct_Rt	State of charge	ratio (0-1)
	ϑ_{bat}	ReESS_tAct_K	Battery temperature	K

Physical interfaces

Electrical interface:

Node	Variable name	Name	Description	Unit
elec out [V]	*u*	phys_voltage_V	voltage	V
elec fb in [A]	*i*	phys_current_A	current	A

B.19 *Accumulator*

Parameters and constants

Parameter name	Unit	Description	Name in Simulink model
ϑ	K	gas temperature	dat.gas.temperature.value
m_g	kg	mass of gas	dat.gas.mass.value
R	J/kg	gas constant	dat.gas.constant.value
V_g	m³	tank volume	dat.capacity.volume.value
V_f	m³	fluid volume	dat.capacity.fluid.value
-	%	initial fluid volume	dat.capacity.fluid.init.value

Signal interfaces

When using this component model, the following control signals must be sent to the component model in a signal bus:

Node	Variable name	Name	Description	Unit
cmd		no ctrl signal		

The following measurement signals are available from the component model:

Node	Variable name	Name	Description	Unit
sensor	p	Acc_presAct_Pa	Pressure	Pa
	ϑ_g	Acc_tGasAct_K	Gas temperature	K
	V_g	Acc_volGas_Rt	Gas volume	ratio (0-1)

Physical interfaces

Fluid interface:

Node	Variable name	Name	Description	Unit
fluid out [Pa]	p	phys_pressure_Pa	pressure	Pa
fluid fb in [m3/s]	Q	phys_flow_m3ps	volume flow	m³/s