Road-to-Rig-to-Desktop – Virtual Development Using Real-Time Engine Modelling and Powertrain Co-Simulation

Jakob Andert¹, *Feihong Xia¹, Serge Klein¹ and Daniel Guse²

¹Mechatronic Systems for Combustion Engines

^{1,2}Institute for Combustion Engines, RWTH Aachen University

Forckenbeckstrasse 4, 52074 Aachen, Germany

Rene Savelsberg, Raul Tharmakulasingam, Matthias Thewes and Johannes Scharf

Gasoline Powertrains, FEV Europe GmbH

Neuenhofstrasse 181, 52078 Aachen, Germany

Abstract

By front-loading of the conventional vehicle testing to engine test bench or even further forward to off-line simulations, it is possible to assess a large variation of powertrain design parameters and testing manoeuvres in the early development stages. This entails a substantial cost reduction compared to physical vehicle testing, and hence an optimisation of the modern powertrain development process. This approach is often referred to as Road-to-Rig-to-Desktop (R2R2D). To demonstrate the potential of this R2R2D methodology as a seamless development process, a crank-angle resolved real-time engine model for a turbocharged gasoline engine was built with the simulation tool GT-POWER[®]. The model was validated with measurement data from an engine test bench and integrated into a vehicle co-simulation, which also includes a dual clutch transmission, the chassis, the environment and the automated driver. The most relevant functions of the engine and the transmission control systems were implemented in a SIMULINK[®]-based software control unit. To verify the engine model in the transient vehicle simulation, two 900 s time windows from a 2 hours Real Driving Emission (RDE) test, representing urban and motorway conditions, are simulated using the developed co-simulation platform. The simulation results are compared with the respective vehicle measurement data. The fuel consumption deviation caused by the combustion engine model is within 5 %. The transient system behaviour and the dominant engine operation points could be predicted with a satisfying accuracy.

Keywords

Real-time engine modelling, Real-Drive-Emission, Virtual development, Model-in-the-Loop

INTRODUCTION

Growing customer requirements regarding comfort and real-world fuel consumption reduction, combined with increasingly stringent emissions regulations, lead to higher complexity of modern combustion engines. This results in additional cost, not only for the engine hardware itself but also for the associated development effort for the engine control functions and the following calibration process. In order to shorten the development phase and thus the cost for an entire vehicle development program, conventional vehicle testing should be front-loaded to engine test benches or even further forward to hardware-in-the-loop and model-in-the-loop simulations. This approach is often referred to as Road-to-Rig-to-Desktop (R2R2D)¹.

The main motivation of R2R2D is a seamless system integration and testing process with an early start using virtual components. The benefits could be a multidimensional system optimisation in transient driving condition during concept study and function development^{2,3}, or an early determination of component weaknesses in a safe test bench environment close to reality without expensive prototype manufacturing^{4,5}. Even a pre-calibration can be possible to provide the initial parameter sets for powertrain control units^{6,7}. As the simulation models can easily be adapted according to existing design changes, the virtual testing platform can be maintained up-to-date during the whole development process.

One of the challenges using the R2R2D concept is the trade-off between accuracy and computational time of the simulated models. While simulation tools like 3D-CFD (three-dimensional computational fluid dynamics) provide the most accurate modelling for the physical system behaviour, they are too computationally expensive to simulate typical vehicle tests which take from minutes to hours. The simplest modelling approach consists in representing the physical components with data based characteristic look-up tables. The disadvantage of such an approach is the requirement of measurement data. The necessary data could be unavailable in case of an early concept study, or not enough for a sufficient modelling quality in case of a complex system behaviour.

Between these two approaches, there are simulation tools combining physical functions with phenomenological or empirical models. Such tools provide the opportunity to customise the degree of detailing depending on the application area and project scope so that they are most suitable for the R2R2D system testing.⁸

On the one hand, building the whole vehicle model including testing environment and keeping it up-to-date during development, usually means enormous effort. On the other hand, while model-based-development is becoming state-of-the-art, there are well-maintained component models existing already in each development phase in most cases. One of the main ideas behind the R2R2D approach is to reuse these models by combining them into a co-simulation scenario. The Functional Mockup Interface (FMI) as a tool

independent standard, for both model exchange and co-simulation, can be used for the model coupling ⁹.

Focusing on the combustion engine development, GT-POWER from Gamma Technology is a good example of the commercial off-the-shelf tools for a model-based design process. Its main simulation scope is on detailed one-dimensional calculation. Such models focus on several engine cycles and can capture high-frequency dynamic effects, which are important for the engine breathing process and when necessary also for the boosting system. Additionally, a more detailed component analysis is possible through a 1D/3D coupling. On the other side, low-frequency 0D models or even empirical map-based models can be derived from the detailed models and implemented in simulations for physical processes with longer duration, such as driving cycles and warm-up simulations. This comprehensive application range provides a good reusability of the engine models in the complete development process.

There have been several research activities regarding the engine modelling based on GT-POWER for transient driving cycle simulations in the last decade. Birckett et al. ¹⁰ have implemented a detailed 1D engine simulation coupled with a vehicle model and a driver model. The influence of different boosting systems on the transient engine behaviour and further on the driver pedal position and shifting points during the driving cycles has been analyzed. Nanjundaswamy et al. ^{11,12} have been focusing on Hardware-in-the-Loop testing and have built a mean value engine model to satisfy the hard real-time requirement. The signals from the vehicle to engine model and engine control unit were recorded from vehicle testing. Wu and Li ¹³ have investigated another simulation approach for Hardware-in-the-Loop, namely crank-angle resolved 0D modelling, using one dummy model from the example library. The real-time capability has been verified on a dSPACE DS1006 simulator. Using the same crank-angle resolved modelling approach, Cosadia et al. ¹⁴ have derived a simplified model from a detailed 1D simulation. The simulation results from the simplified model are consistent with the base detailed model in a driving cycle, with imposed boundary conditions from vehicle measurement, including engine speed and injected fuel mass.

For this paper, the simplified, but still crank-angle resolved modelling approach is chosen, as the simulation speed should be near wall-clock-time while achieving a high modelling accuracy. To demonstrate the potential of the R2R2D concept, the engine model is integrated into a vehicle co-simulation platform developed by Klein et al. ¹⁵. The platform includes a dual clutch transmission, the chassis, the environment, the automated driver and the most relevant functions of the engine and the transmission control systems. The models are parameterised for an A-segment mini car and have been validated with vehicle measurements including coast down on the test track and NEDC (New European Driving Cycle) on chassis dynamometer test bench. The closed-loop simulation works with predefined driving manoeuvres and can simulate the interaction between engine, transmission and driver depending on component characteristics and control functions. For a front-loaded vehicle testing, it is a pre-condition that this interaction is close to reality.

The paper is structured as follows: It begins with an overview of the system simulation structure, followed by reviewing the combustion engine modelling, including model introduction, model validation, control function implementation and model integration. Finally, an exemplary vehicle simulation under real drive conditions is implemented and verified with measurement to show the potential of virtual vehicle testing on a desktop PC.

SYSTEM STRUCTURE

To demonstrate the R2R2D approach, a base co-simulation platform is developed that can simulate transient driving cycles like NEDC, WLTP (Worldwide Harmonized Light-Duty Vehicles Test Procedure) and RDE (Real Driving Emissions). A co-simulation structure overview is shown in Fig. 1.

Fig. 1 System simulation structure: Engine, transmission, vehicle dynamics and control units

The co-simulation is between three simulation tools. Besides the combustion engine modelled in GT-POWER, the SIMULINK-based Automotive Simulation Models (ASM[®]) tool suite from dSPACE is used to simulate the vehicle behaviour and characteristics, such as steering, suspension, brake system and aerodynamics. It also provides the possibilities for road and traffic modelling with dedicated tools for customisation and visualisation. The transmission and the drivetrain are simulated in the commercial simulation tool SimulationX[®] from ESI ITI. It is a system simulation tool using the open modelling language Modelica. It provides predefined libraries for 1D rotational and transmission elements.

For setup of the co-simulation platform, the interfaces between these models have to be defined in a first instance. In this paper, the driving torque of the vehicle is

calculated within the combustion engine subsystem and is transmitted through the dual clutch transmission to the wheels. Based on this driving torque, the vehicle velocity is calculated by the vehicle dynamics submodule, which defines the rotating speed of the wheels and the crankshaft, accordingly. The FMI standard for co-simulation is used to couple the simulation models with their numerical solvers. The data exchange between the models is restricted to discrete communication time steps. In the time between two communication points, the models are solved independently from each other.⁹

Such a “straightforward” dynamic simulation including a physical engine model, is different from the map based “backwards” simulations, which calculate the engine torque based on vehicle reference speed profile and acquire fuel consumption from the engine map¹⁶. While the “backwards” simulations work with steady state conditions and usually do not need any control logic, the “forward” simulations must work with controllers and can capture transient phenomena in the system influenced by the controller implementation and the component layouts.

For the driving cycle simulation, an automated driver is implemented in the ASM tool suite. It works with a feed-forward part, which calculates the required engine brake torque from the reference acceleration and the driving resistance, and a feed-back loop, which is implemented as a PID (proportional–integral–derivative) controller reacting on the actual deviation of the vehicle velocity. Furthermore, a relatively simple transmission control unit (TCU) is implemented in SIMULINK for the dual clutch transmission. The TCU controls the torque capacity of both clutches and the four synchronizers and sends the torque requests to the engine control unit (ECU) during gear shifts.

Some ECU basic control functions, like idle control and boost pressure control, are also mandatory for transient engine operation. These functions implemented in the model will be introduced later in the section - Engine Control Unit.

ENGINE MODEL INTRODUCTION

There are several simulation approaches for combustion engines, which can be implemented for different development phases in V-model¹⁷, as shown in Fig. 2. Due to the high computational power demand of 3D-CFD simulation, it is rarely used to simulate the entire air path. Therefore, the detailed 1D-CFD is the state-of-the-art simulation approach for steady-state investigations during engine air and combustion system design process, e.g. intake manifold design, valve timing optimisation and turbocharger matching. The air path volume is discretized along the pipe length, with a discretisation length of approx. half of the cylinder bore diameter. Using an explicit flow solver, the solver time step is restricted to below 1°CA ¹⁸ by the relatively small discrete spatial grid according to the Courant’s condition¹⁹. Hence, the detailed 1D engine models have a high model complexity and a large real-time factor, which is defined as a ratio of the time needed for the simulation calculation and the physical process duration. Taking the simulation tool GT-POWER as an example, the detailed 1D model for engine cycle simulation usually runs 100 times slower than wall-clock time.

Fig. 2 Classification of engine modelling approach with reference to different development phases in V-model ^{14,17}

Moving from the detailed engine cycle simulation to a driving cycle simulation for virtual system integration and testing, a higher simulation speed becomes essential for the combustion engine model. To fulfil the run-time requirement, a detailed 1D engine model can be turned into a reduced 1D or 0D model. Larger solver time step can be enabled by increasing the discretisation length of the 1D components and combining multiple 1D components to 0D volumes. Larger time steps and fewer sub-volumes reduce the total amount of equations to be solved within one engine cycle. As the highest simulated dynamic wave frequency also decreases with the discretisation level and may lead to model inaccuracy, an adaptation of friction losses and heat transfer could be necessary to maintain the most important simulation results like boost pressure and volumetric efficiency.

The significant advantage of a 1D/0D reduced engine model compared to map-based mean value model is the maintenance of the crank-angle resolved cylinder modelling. The volumetric efficiency is still calculated by modelling the gas exchange process. The cylinder pressure is calculated in the same way as in detailed 1D modelling, which usually means combustion heat release calculation based on the Wiebe function ²⁰ and wall heat transfer modelling based on Woschni's correlation ²¹. This extension of simulation enables modelling of dynamic interactions between ambient condition, throttle control, valve timing, injection, ignition, engine coolant, turbocharger operation, exhaust gas after-treatment system and exhaust gas recirculation during driving cycles. The pulsation in engine torque resulting from the cylinder pressure variation can also be used for an acoustic analysis of the whole powertrain.

A comparison of the simulated pumping loop from a 1.5 L, 4-cylinder, turbocharged gasoline engine at low-end torque operation ($n = 1500$ 1/min, BMEP = 19 bar) between a 1D detailed model and a 0D reduced model is shown in Fig. 3. This operation point is chosen for its critical sensitivity of pressure modelling due to scavenging during valve overlap. The detailed model has a discretisation length around 50 mm and a simulation time step of 0.1° CA. The 0D model has a minimum element

volume of 1.3 L and a simulation time step of 3 ° CA. It runs approx. 30 times faster than the detailed model. Although some delicate high-frequency wave dynamics are missing in the 0D model, the main characteristic pressure change in the exhaust manifold remains the same. This pressure pulsation is essential for volumetric efficiency and turbocharger operation. The scavenging effect can be simulated by the 0D model with a sufficient accuracy.

Fig. 3 Comparison between the simulated pressure in cylinder, intake and exhaust manifolds and mass flow through valves from a detailed 1D model and a reduced 0D model at low-end torque

The 0D combustion engine model in the focus of this paper is built for a 0.9 L, 3-cylinder turbocharged gasoline engine with port injection and variable valve timing. The reference engine has been measured on a thermodynamic dynamometer test bench with cylinder pressure indication for model validation.

An overview of the model is shown in Fig. 4. The model is optimised for simulation speed. The intake and exhaust ducts are modelled by six sub-volumes between the air filter, the compressor, the cylinders, the turbine and the environment. The minimum volume is 0.62 L. The turbocharger is modelled by characteristic maps over mass flow, pressure ratio, rotating speed and efficiency. The inertia of the turbocharger is considered in the mechanical shaft. The compressor recirculation valve is modelled using an orifice connection. The wastegate is included in the turbine model. The flow properties of the cylinder head according to valve lift are implemented according to detailed measurements taken in the flow laboratory. Wiebe parameters imposed as multi-dimensional look-up tables based on cylinder pressure indication from engine test bench are used for modelling the combustion processes. Since no engine cooling system is modelled yet, the wall temperature in the system is partly imposed depending on engine power, partly calculated based on the heat capacity of wall materials and ambient temperature. Engine friction is modelled using the Chen-Flynn model ²² depending on the maximum cylinder pressure and the mean piston velocity.

Fig. 4 Overview of the 0D engine model

A high-level controller is used to determine the required fuel mass flow with respect to the current air mass flow through the throttle. There are the following control signals needed for a transient simulation:

- Throttle angle
- Wastegate diameter
- Ignition timing
- Valve timing
- Target air-fuel ratio

The engine model has a fixed solver time step of 0.25 ms, which corresponds to 9 ° CA at 6000 1/min. It runs faster than wall-clock time on a regular Windows laptop with an Intel Core i5 processor. The real-time capability has been demonstrated using a dSPACE SCALEXIO Hardware-in-the-Loop simulator⁴.

From the author's point of view, this model is sufficient to create a base system structure for the driving cycle co-simulation platform. Using this model, the first verification using RDE vehicle measurement can be carried out, and the first investigation on the influence of engine layout on RDE can be implemented. For other investigation aims in the future, the model can be extended individually by changing modelling options or by coupling with other component models, for example using a predictive combustion model instead of the Wiebe function.

ENGINE MODEL VALIDATION

Before being integrated into the powertrain co-simulation, the 0D engine model is calibrated to steady-state engine test bench data. As most of the model parameters are directly derived from engine specification and component measurements, the main calibration parameters are the wall temperature and the heat transfer coefficient in cylinders and intake as well as exhaust manifolds respectively. These parameters are hard

to measure on a standard engine test bench but have a strong influence on engine volumetric efficiency and turbocharger operation.

To show the modelling quality, the p-V diagram and the pumping loop are shown in Fig. 5 for the three engine operation points representing the whole engine map: Low-end torque ($n = 2000$ 1/min, BMEP = 18 bar), rated power ($n = 5000$ 1/min, BMEP = 16.5 bar), and a part load operating point with moderate throttling and internal exhaust gas recirculation ($n = 2000$ 1/min, BMEP = 6 bar). The measurement data is averaged over 50 engine cycles. The simulation results are calculated with the engine load controlled to the measurement data.

Fig. 5 Comparison of pressure in cylinder, intake and exhaust manifolds between measurement and simulation

First of all, no solver instability issue is notable, even at maximum engine speed. The cylinder pressure, as well as the pressures in the intake and exhaust manifolds, can be simulated with satisfying accuracy resolved over crank angle within one engine cycle, which ensures a precise calculation of engine indicated efficiency.

Based on the accurate cylinder modelling and a proper calibration of the wall heat transfer in the exhaust manifold, also the exhaust gas temperature can be modelled close to the measurement data. The comparison of temperature upstream turbine at engine full load is shown in Fig. 6, and it can be seen that the maximum temperature is well matched. Good accuracy in modelling the exhaust gas temperature is essential for simulating turbine operation and exhaust after-treatment system performance.

Fig. 6 Comparison of the time-averaged exhaust gas temperature upstream turbine between simulation and measurement at engine full load

Last but not least, the comparison of engine brake specific fuel consumption (BSFC) is shown in Fig. 7. The maximum deviation (around 15 %) occurs at very low engine speeds and high engine load. The reason is a higher scavenging effect in the simulation, which results in a lower air-fuel ratio in the combustion chamber, as the air-fuel ratio is imposed after the total air mass flow through the throttle. Another deviation over 12 % results at high engine speed and very low engine load, where the ignition delay referenced on crank angle becomes very long, which is challenging for the combustion simulation. Besides a certain potential of improvements, the model provides a good accuracy with an error below 6 % in the primary engine operation area under the real driving conditions and is thus sufficient for the driving cycle simulation.

Fig. 7 Comparison of the engine brake specific fuel consumption between simulation and measurement in steady state engine map

With sufficient data inputs from engine test bench, hot gas turbocharger test bench and flow laboratory, the combustion engine modelling and calibration process took approximately 100 person-hours. The calibration process of the reduced engine model is noticeably shorter than that of a detailed 1D engine model due to obviously fewer calibration parameters and strongly increased simulation speed. The process duration could be further reduced if a well-calibrated detailed engine model was already available.

Additional effort has been taken for the function development and the model integration, which is described in the following sections.

ENGINE CONTROL UNIT

Additional to the steady state validation, the engine transient behaviour is essential for modelling of driving cycles. In case of a target vehicle speed profile, the variation in engine transient behaviour affects the driver pedal position, which again has a direct influence on gear shifting and therefore results in a different engine operation point. However, the transient response of the engine depends not only on the thermodynamic engine modelling but also on the engine control application and the actuator characteristics. To represent the interaction of an engine control unit and the engine according to driver torque demand, the following basic control functions are implemented in SIMULINK:

Throttle angle feed-forward control

Variable valve timing feed-forward control

Idle control using throttle and ignition timing

Wastegate feed-forward and feed-back control of boost pressure

Compressor recirculation valve activation along compressor surge line

Fuel cut according to torque request and engine speed

Torque intervention from transmission control unit during shifting

The control functions are developed in parallel to the engine model calibration process. An early established co-simulation as Model-in-the-Loop testing is immensely beneficial for function calibration and application. An example of the system transient behaviour in comparison to engine test bench measurement is shown in Fig. 8 as a load step at a constant engine speed of 2000 1/min. The slight difference is mainly due to different control function implementation and parameter calibration in comparison to the hardware ECU. For a model validation point of view, a Hardware-in-the-Loop test with the original ECU could be a solution to avoid time-consuming reconstruction of the original control functions⁷. However, the hardware ECU won't be available in the early development phases such as concept design and engine layout, on which this paper is focusing.

Fig. 8 Comparison between a simulated load step at 2000 1/min and the measured data from an engine dyno test bench

ENGINE MODEL INTEGRATION

The engine model built in GT-POWER can be integrated into SIMULINK model using S-function. There is an adapted solver especially for the real-time application so that the models exported as s-function can be compiled for certain target real-time hardware, for example, dSPACE SCALEXIO. In case another simulation tool is used as a master in co-simulation, the FMI standard for co-simulation can be used for tool coupling. The engine model is then compiled and zipped with an XML description file into a functional mock-up unite (FMU). On the other side, GT-POWER also supports FMU import from other simulation tools and acts as simulation master.²³

As the vehicle simulation tool dSPACE ASM does not support FMU export yet, the engine model is integrated into the SIMULINK environment. Besides the engine speed and engine control signals as input, the model delivers engine torque, engine fuel mass flow, 50 % mass fuel burned point, exhaust lambda, engine friction, turbocharger speed and the temperature and pressure conditions in the intake and exhaust systems as outputs.

The integrated vehicle co-simulation runs around two times slower than wall-clock-time in SIMULINK environment on a regular Windows laptop with an Intel Core i5 processor. The simulation speed is mainly restricted by the vehicle dynamics model in ASM. However, real-time application can be realized using a dSPACE SCALEXIO simulator⁴.

RESULTS VERIFICATION

The co-simulation platform is a closed-loop simulation with a driver model reacting to predefined driving manoeuvres. Therefore, it can be used to predict the vehicle behaviour in any driving scenario. To verify the simulation accuracy regarding the combustion engine model, one vehicle measurement under real drive condition is chosen as reference. The test vehicle had a total mass of 1250 kg. The CAN bus was monitored by a CAN adapter. The emissions were measured by a Portable Emissions Measurement System (PEMS). The ambient pressure was around 1006 mbar. The ambient temperature was 20 °C. The max. wind speed was 12 km/h, and the weather was dry and cloudy.

The whole test drive took 6911 s and covered a distance of 115 km, including urban, rural and motorway operation. The altitude varied between 80 m and 590 m during the test. The driving dynamic is plotted in Fig. 9 as vehicle acceleration over driving velocity using data points with 1 s interval, together with the 95 percentile upper boundary defined by the latest draft of RDE legislation²⁴.

Fig. 9 Dynamics of the complete RDE measurement and the two chosen 900 s time windows for urban and motorway condition

To analyse the engine behaviour separately for low load and high load conditions, two 900 s time windows are chosen from the vehicle test to represent urban and motorway scenarios. The selected data ranges are highlighted in Fig. 9. The average velocity of the urban part is 37 km/h, including vehicle stops. The average velocity of the motorway part is 116 km/h. Both driving profiles are set up in the model with the corresponding altitude profile from GPS measurement signals.

For the overall comparison of the simulation results to the measured data, the total driving energy is calculated for both cases using torque and rotating speed of the crankshaft integrated over time. This value mainly represents the modelling accuracy of the transmission efficiency and vehicle driving resistance. The driving distance within the given 900 s is used to check the precision of the driver controller, which could influence the energy consumption in case of strong gradients in the road profile. The total fuel consumption as one of the primary results of the simulation is used to rate the modelling quality of the combustion engine.

As the deviation in the total driving energy can have a direct influence on the total fuel consumption and hence affects the valuation of the engine model itself, two simulations are implemented with imposed accelerator pedal positions and engine speeds recorded from vehicle tests additional to the closed-loop simulations. Since no other input is needed for the combustion engine, these open-loop simulations with playback signals are implemented without the vehicle and the transmission model.

The relative differences of the closed-loop simulations with driver controller and the open-loop simulations with playback signals referred to vehicle tests for urban and motorway are shown in Fig. 10. In the cases with playback of the acceleration pedal position and the engine speed, the idling phases are neglected for the comparison, since the idle controller in the standalone engine model has to be deactivated. As there is no vehicle model, the distance is not calculated.

Fig. 10 Comparison of measured and simulated driving energy on wheels, fuel consumption and driving distance during the selected 900 s time windows for urban and motorway condition

As it is shown in Fig. 10, the open-loop simulations show only slight deviation to the measurement in the total driving energy, which signifies a precise load control in the reconstructed engine control unit. The deviations of the simulated fuel consumption to the measurement are within 5 % for both urban and motorway. The inaccuracy is partly due to the engine model's steady state calibration itself, partly due to the different engine operating condition in the vehicle compared to engine test bench. Extending the model with a cooling system coupled with the cylinder wall temperature and taking the influence of oil temperature on engine friction into account should further improve the modelling accuracy.

In the closed-loop simulations, the deviations in the simulated driven distance to measurement data are below 1 %. The vehicle speed profile can be followed precisely by the automatic driver. However, the driving energy shows increased deviation to the measurement compared to the open-loop simulations. This mismatch of the energy leads to an increased simulation error in the fuel consumption. As the driving resistance under real drive condition can be influenced by temperature, humidity, wind, and different road surfaces, a fine calibration of the vehicle dynamic model according to the testing conditions can be beneficial. However, as the vehicle model should be generally valid for different vehicle tests, an adaptation for this particular RDE test has not been implemented.

Besides the overall fuel consumption, the individual engine operation points during the driving scenarios have high relevance for the powertrain design. The engine operation can be analysed using the fuel share diagram, which presents the dominant engine operating areas in the engine map with the most fuel consumption during the whole driving scenario. The fuel share is integrated over time within 125 1/min engine speed and 0.5 bar BMEP grids referred to the total fuel consumption. A comparison of the measured and the closed-loop simulated fuel share diagram during both urban and motorway driving is shown in Fig. 11.

Fig. 11 Comparison of measured and simulated distribution of the percentage fuel share during the selected 900 s time windows for urban and motorway condition

Compared to the measurement, the simulated total driving energy is lower in the urban part and higher in the motorway part. Therefore, the dominant engine operating areas in the simulation are slightly shifted to lower engine speed and lower load in the urban part and to higher engine speed and higher load in the motorway part. The fine distribution of the operating points could also be influenced by different reaction of the driver controller compared to a human driver, and particular transient modifications in the shifting strategy due to drivability, which are too complicated to reconstruct for the transmission control in the co-simulation. Despite the mentioned deviations, the engine operation characteristic during both driving scenarios can be predicted with a sufficient accuracy.

The main advantage of a 0D engine simulation in the vehicle co-simulation compared to a map based solution is the modelling of the internal engine process during the transient driving profile, including crank angle resolved temperature, pressure, mass flow, and the indicated torque on the crankshaft. These detailed simulation results can be used to verify and optimise the main components within the engine air path. For example, the simulated engine cycle averaged operating points of the turbocharger compressor are shown in Fig. 12. The compressor is operating at a higher rotating speed and pressure ratio on the motorway and hence closer to the area of optimum efficiency. With this simulation platform, the influence of a turbocharger adaptation on the engine operation and vehicle behaviour during driving cycle can easily be analysed by changing the turbocharger characteristic maps and turbocharger inertia. By extending the co-simulation platform with electric components like generator and battery, also electrified engine boosting systems and their electric energy consumption can be investigated in various driving conditions ²⁵.

Fig. 12 Simulated compressor operation points during the selected 900 s time windows for urban and motorway condition

SUMMARY

Within the scope of the Road-to-Rig-to-Desktop approach, conventional vehicle testing can be front-loaded to the engine test bench and further to offline simulations, to ensure an early system integration and validation. A seamless toolchain for component modelling in the development process is required to reduce modelling effort and improve simulation accuracy.

In this study, a crank angle resolved real-time capable 0D combustion engine model is set up in the simulation tool GT-POWER and integrated into a co-simulation platform including simulation models for the transmission, vehicle dynamics and driving environment. Basic control functions for engine and transmission, together with a driver controller, are implemented to ensure a closed-loop simulation reacting to the reference driving manoeuvres.

To verify the engine model in transient operation, two speed profiles from an RDE vehicle measurement, representing urban and motorway driving condition, are used as reference driving profiles in the simulation. Using a stand-alone engine model with playback signals of the accelerator pedal position and engine speed, the deviation of simulated total fuel consumption to measurement data is under 5 % for both urban and motorway operation. The closed-loop co-simulations show bigger deviations mainly due to the inaccuracy in the vehicle dynamics model. Nevertheless, the main engine operation areas in the engine map show good accordance with the measurement data. The usability of this engine model in the system testing phases could be approved. The effort of the model calibration and adaptation has been shown to be acceptable.

The 0D engine model can be derived from the detailed 1D-models with a customised level of simplification according to the runtime requirement. In the development process, such combustion engine models integrated into the powertrain co-simulation platform can be used to evaluate different concept layouts, to define

component requirements, to predict engine behaviour and fuel consumption, to test control functions, or even to support pre-calibration of the engine control units. The engine model can easily be extended by co-simulation and continually adapted in parallel to engine test bench measurements.

Acknowledgements

The authors would like to thank Gamma Technologies, dSPACE and ESI ITI for their helpful suggestions, and Mr Jens Achenbach as a student assistant at VKA, RWTH Aachen University, for his assistance in modelling and results analysing.

Declaration of conflicting interests

The authors declare that there is no conflict of interest.

Funding

This work was partially performed as part of the research project “Advanced Co-Simulation Open System Architecture” (ACOSAR) within the European EUREKA cluster program ITEA3.

It also was partially performed within the “Center for Mobile Propulsion” funded by the German Research Foundation (DFG).

References

1. Hülshorst T, Richenhagen J, Richert F and Nase A. Neue Dimensionen in der automobilen Softwareentwicklung. *ATZ Elektron* 2015; 10: 50–53.
2. Klein S, Griefnow P, Guse D, Xia F and Andert J. Virtual 48 V mild hybridization - efficient validation by Engine-in-the-Loop. *SAE Technical Paper* 2018; 2018-01-0410.
3. Deppenkemper K, Özyalcin C, Ehrly M, Schönen M, Bergmann D and Pischinger S. 1D engine simulation approach for optimizing engine and exhaust aftertreatment thermal management for passenger car diesel engines by means of variable valve train (VVT) applications. *SAE Technical Paper* 2018; 2018-01-0163.
4. Xia F, Griefnow P, Klein S, et al. Crank angle resolved real-time engine modeling for HiL based component testing. *The 19th Asia Pacific Automotive Engineering Conference & 2017 SAE-China Congress & Exhibition*; 2017.
5. Andert J, Klein S, Savelsberg R, Pischinger S and Hameyer K. Virtual shaft: Synchronized motion control for real time testing of automotive powertrains. *Control Engineering Practice* 2016; 56: 101–110.
6. Lee S-Y, Andert J, Quérel C, et al. Virtual calibration based on X-in-the-Loop: HiL Simulation of Virtual Diesel Powertrain. *VPC, Simulation und Test, 19. MTZ-Fachtagung* 2017.

7. Xia F, Lee S-Y, Andert J, et al. Crank-Angle Resolved Real-Time Engine Modelling - A Seamless Transfer from Concept Design to HiL Testing. *SAE Technical Paper* 2018; 2018-01-1245.
8. Albrecht A, Grondin O, Le Berr F and Le Sollic G. Towards a Stronger Simulation Support for Engine Control Design: A Methodological Point of View. *Oil & Gas Science and Technology - Rev. IFP* 2007; 62: 437–456.
9. Blochwitz T, Otter M, Akesson J, et al. Functional Mockup Interface 2.0: The Standard for Tool independent Exchange of Simulation Models. In: *9th International MODELICA Conference, Munich, Germany*, Sept. 3-5, 2012, pp. 173–184: Linköping University Electronic Press.
10. Birckett A, Tomazic D, Bowyer S, Wetzel P and Keidel S. Engine and Vehicle Simulations of a 1.6L Light Duty Diesel Engine with Supercharger-Turbocharger and Twin Turbocharger Boosting Systems. In: *Aufladetechnische Konferenz*, Dresden, 2012.
11. Nanjundaswamy H, Tatur M, Tomazic D, et al. Development and Calibration of On-Board-Diagnostic Strategies Using a Micro-HiL Approach. *SAE Technical Paper* 2011; 2011-01-0703.
12. Nanjundaswamy HK, Dahodwala M, Farsodia M, Tomazic D and Koehler E. Virtual Testing and Simulation Environment [Micro-HiL] for Engine and Aftertreatment Calibration and Development -Part 2. *SAE Technical Paper* 2012; 2012-01-0928.
13. Wu H and Li M-F. A Hardware-in-the-Loop (HiL) Bench Test of a GT-Power Fast Running Model for Rapid Control Prototyping (RCP) Verification. *SAE Technical Paper* 2016; 2016-01-0549.
14. Cosadia I, Silvestri JJ, Papadimitriou I, Maroteaux D and Obernesser P. Traversing the V-Cycle with a Single Simulation - Application to the Renault 1.5 dCi Passenger Car Diesel Engine. *SAE Technical Paper* 2013; 2013-01-1120.
15. Klein S, Savelsberg R, Xia F, et al. Engine in the Loop: Closed Loop Test Bench Control with Real-Time Simulation. *SAE International Journal of Commercial Vehicles* 2017; 2017-01-0219.
16. Liermann M. Backward simulation - A tool for designing more efficient mechatronic systems. In: *9th International MODELICA Conference, Munich, Germany*, Sept. 3-5, 2012, pp. 867–876: Linköping University Electronic Press.
17. van Loon H. *Process assessment and ISO/IEC 15504: A reference book*. 2. ed. New York, NY: Springer, 2007.
18. GT-SUITE: Flow Theory Manual. VERSION 2017.
19. Courant R, Friedrichs K and Lewy H. On the Partial Difference Equations of Mathematical Physics. *IBM J. Res. & Dev.* 1967; 11: 215–234.
20. Wiebe I. Halbempirische Formel für die Verbrennungsgeschwindigkeit. In: *Verlag der Akademie der Wissenschaften der UdSSR*. Moskau.

21. Heywood JB. *Internal combustion engine fundamentals*. International ed. London u.a.: McGraw-Hill, 1988.
22. Chen SK and Flynn PF. Development of a Single Cylinder Compression Ignition Research Engine. In: *National Powerplant and Transportation Meetings*, OCT. 18, 1965.
23. GT-SUITE: Controls Coupling and Real Time Manual.
24. THE EUROPEAN COMMISSION. *COMMISSION REGULATION (EU) 2016/646 of 20 April 2016 amending Regulation (EC) No 692/2008 as regards emissions from light passenger and commercial vehicles (Euro 6)*, 2016.
25. Lückmann D, Schloßhauer A, Müller A, et al. Assessment of an Electrified Boosting System to Increase the Efficiency of a Hybrid Powertrain by Exhaust Energy Recovery. In: *22. Aufladetechnische Konferenz*. Dresden, 2017.