

Engine in the Loop: Closed Loop Test Bench Control with Real-Time Simulation

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ABSTRACT

The complexity of automobile powertrains grows continuously. At the same time, development time and budget are limited. Shifting development tasks to earlier phases (frontloading) increases the efficiency by utilizing test benches instead of prototype vehicles (road-to-rig approach). Early system verification of powertrain components requires a closed-loop coupling to real-time simulation models, comparable to hardware-in-the-loop testing (HiL). The international research project Advanced Co-Simulation Open System Architecture (ACOSAR) has the goal to develop a non-proprietary communication architecture between real-time and non-real-time systems in order to speed up the commissioning process and to decrease the monetary effort for testing and validation. One major outcome will be a generic interface for coupling different simulation tools and real-time systems (e.g. HiL simulators or test benches). In this paper, we show the seamless transition from a purely simulated vehicle in a MiL (model in the loop) co-simulation to a heterogeneous testing scenario with an engine test bench linked to real-time models. First, the offline co-simulation consisting of a GT-POWER engine model, a SimulationX transmission model and a dSPACE ASM vehicle dynamics model is set up. All models are parametrized based on extensive vehicle measurements. Second, the engine model is substituted by an engine test bench which is coupled with the same real-time simulation of the transmission and the vehicle. A suitable closed-loop interface between the simulation, the dSPACE HiL simulator SCALEXIO® and a test bench automation system is created. Finally, the simulation results and test bench measurements are compared with a real test vehicle. The results show a good congruence and a high potential for the use of test benches and MiL co-simulation in early project phases. Standardization requirements for a real-time capable simulation interface are derived from this use case.

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INTRODUCTION

Development cycles in the automotive industry are becoming shorter and shorter. At the same time, the complexity of vehicle powertrains is growing. Researchers from the industry and science have developed a variety of approaches to overcome this issue [1, 2, 3, 4]. One common solution is to shift development tasks to earlier stages of the development process. In the automotive industry, hardware-in-the-loop (HiL) is widely used for rapid prototyping, validation and parametrization of control units in early development phases. In general, the HiL approach can be paraphrased as the coupling of any physical component with real-time simulation [5]. Especially engine test benches embedded in realistic simulations (engine-in-the-loop, road to rig) seem to be a promising approach to reducing development efforts. Challenges and advantages of engine-in-the-loop (EiL) testing are described in [6, 7, 8]. A common view

throughout all of the publications is that EiL is considered to have great potential. However, comparable results between EiL and real prototype vehicles are rare or still part of ongoing research [9].

The quality of EiL tests heavily depends on the used simulation models, parameter accuracy and the implementation in the test bench. Time and costs needed for a simulation increase rapidly with a growing level of detail. An economic balance between the simulation's quality and the necessary efforts needs to be found eventually. Co-simulation approaches like FMI [10] allow for modular model exchange and reduce the simulation development effort. In contrast to a conventional test bench, the effort to couple real-time simulation with a test bench is also higher. Although EiL has great potential, the increased integration work can outweigh its benefits. The international research project Advanced Co-Simulation Open System Architecture (ACOSAR) emphasizes the issue of great efforts to set up real-time

simulation and engine test bench. Therefore, one major goal is the development of a non-proprietary interface between real-time simulation and engine test bench. This study contributes to the ACOSAR development process by showing the challenges of the coupling process of test bench and real-time simulation.

The paper is divided into the four chapters “Vehicle measurements”, “Simulation”, “Engine-in-the-loop” and “Comparison of results”. Usually, vehicle measurements are conducted at the end of the development process. However, due to the lack of available vehicle data, the measurements are taken at the beginning. During a normal development process, the data is available or provided by the original equipment manufacturer (OEM). The vehicle tests are performed with a reference vehicle and deliver parameters for the simulation of vehicle dynamics, the combustion engine and the transmission. Co-simulation is then explained with respect to a foresighted interface between each model. Advantages of the modular co-simulation approach are emphasized and explained. The next step involves the substitution of the combustion engine by a combustion engine test bench. An overview of the test bench is provided along with a description of the challenges during the coupling process. The last chapter deals with the comparison and discussion of the reference measurements, simulation data and test bench measurements.

VEHICLE MEASUREMENTS

The required system parameters are determined through extensive drive tests with the reference vehicle. The system parameters can be divided into the three sub-systems “vehicle”, “combustion engine” and “transmission”. The specific tests for each sub-system are explained below.

Reference Vehicle

The reference vehicle contains a turbocharged three-cylinder gasoline engine and a dual-clutch transmission. The relevant control units for this study are the engine control unit and the transmission control unit. Each of these control units has an additional calibration interface to measure signals or to modify calibration parameters. Furthermore, a CAN adapter allows the CAN bus to be monitored. With the help of additional measuring devices, different operating points of engine temperature, vehicle acceleration and generator load can be determined.

Vehicle Model Calibration Tests

The vehicle is subjected to repeated coastdown testing on an appropriate test track to determine the parameters for the real-time vehicle model. The track has a length of about 2500 meters, is protected against cross-winds and is absolutely flat.

During the coastdown tests, the real vehicle mass including the external measuring equipment and two test drivers is taken into account. The tire pressure is in line with the manufacturer’s recommendations. An ambient temperature of 20-22°C, an ambient

air pressure of 1015 mbar and a maximum wind velocity of 2.8 m/s were measured, which meets specific legal requirements regarding the environmental conditions for coastdown tests [11]. The vehicle coasted from 135 km/h down to a standstill 16 times in total. The average deviation from the coastdown time within a 10 km/h speed interval (e.g. 105-95 km/h) is lower than two percent.

Engine Model Calibration Tests

In order to calibrate the engine model, additional tests are carried out. To identify the transient behavior of the mean effective pressure, tip-in maneuvers are executed with the reference vehicle and load steps are performed on an engine test bench. The friction’s mean effective pressure is also identified on the engine test bench. For this purpose, the engine is run in overrun condition.

Transmission Model Calibration Tests

Several further drive tests are required to identify transmission parameters. Therefore, the reference vehicle was accelerated from a standstill with a defined position of the acceleration pedal between 0 % and 100 % until a steady vehicle velocity (balance of forces) was reached. For a fixed accelerator pedal position (e.g. 25 %), the test was repeated five times in order to minimize environmental influences, e.g. wind or road conditions. The focus throughout all the tests lied on the identification of the logical procedures in the DCT with respect to engine load.

With the help of these tests, the moment when the passive gear is synchronized with the drive side can be identified. Furthermore, the duration of the torque crossfade between active and passive clutch is examined through this drive tests. Also the duration of the engine’s slowdown during gear change is measured, and the characteristic shape and amplitude of the engine’s torque reduction during gear change is identified.

All these characteristic values are illustrated in [Figure 1](#) for a gear change from 3rd to 4th gear at 40 % acceleration pedal position. The first graph shows engine torque and the torque capacity of both clutches. The second graph shows the engine speed and the speed of both gear shafts. At the beginning, the dashed line shows the passive gear shaft and the dash-dotted line the active gear shaft. In graph 2, the break point at 18.8 s indicates the synchronization of the passive shaft. After 1 s, the torque capacity is transferred from the active to the passive gear shaft in 0.4 s. The duration of the engine’s slowdown during the whole shifting process is about 0.5 s (second graphic, solid line). The reduction of the effective engine torque is clearly visible from 20.3 s to 21 s plotted in the first graph (solid line).

Besides the upshift behavior of the DCT, it is also important to examine downshift. Defined vehicle retardation tests are executed to identify the transmission’s downshift behavior. First, the vehicle coasts from 130 km/h with the gear lever in D position. Afterwards, the vehicle decelerates from that speed in the same gear lever position with 1.8 m/s² and 4.0 m/s² deceleration

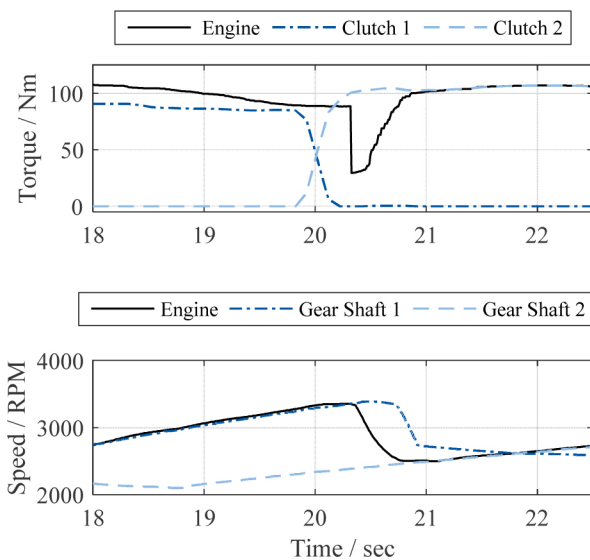


Figure 1. Gear change event from gear 3 to 4

SIMULATION

Based on the vehicle measurements, it is possible to create and parametrize a detailed simulation of all the vehicle dynamics. In the simplest case, only one simulation tool is used to develop and run a complete system simulation. This study makes use of the co-simulation approach [12]. Co-simulation allows for models to be developed, where each one represents a sub-system, with different simulation tools and to be executed together within a complete system simulation run by a master simulation tool. Through this approach models from domain-specific simulation tools can be integrated in a modular way. In the present study, the sub systems comprise a GT-POWER combustion engine model, a SimulationX transmission model and a dSPACE ASM vehicle dynamics model. All models are created and calibrated within their specific simulation tools. The simulation of the complete system is performed in VEOS®, a software for PC-based simulation of virtual ECUs and environment models from dSPACE. With this tool it is possible to integrate models via the FMI standard. This requires the definition of model inputs and outputs. Moreover, the interface for offline simulations has to be designed with the requirements of the intended coupling between real-time simulation and a test bench in mind. The most relevant coupling signals are shown in Figure 2. The automated driver in the vehicle simulation starts the engine and generates a desired accelerator pedal signal. Within the combustion engine model, the pedal position is converted to an effective crankshaft torque and sent to the input shaft of the transmission. Depending on the current state of the transmission, the drive shaft torques are transmitted to the vehicle model and the engine speed is sent back to the engine model. The shaft torques cause the vehicle to move resulting in four different wheel speeds which are fed back to the transmission.

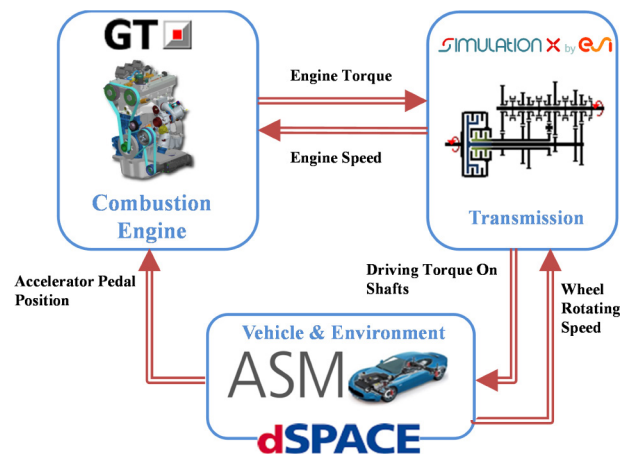


Figure 2. Simulation structure

Combustion Engine

For the MiL approach, an engine model was built using GT-Power, a 1D-CFD (computational fluid dynamics) tool for engine performance simulation from Gamma Technologies. While 3D-CFD mainly focuses on in-cylinder fluid kinematics and combustion processes, 1D-CFD is widely used to analyze the flow field throughout the whole intake and exhaust tracks and provides an accurate calculation of the engine gas-exchange process. A standard 1D engine model usually runs 100 times more slowly than wall-clock time. Such models are built for steady-state simulations during engine air and combustion system design process, such as intake manifold design, valve timing optimization and turbocharger matching. For transient simulation with other vehicle components and driver, a higher simulation speed is required while a certain decrease of simulation accuracy is acceptable. This can be realized by reducing the volume discretization level, simplifying combustion calculation, increasing the integration time step and minimizing unnecessary results output. These reduced models can run close to real time and are suitable for EIL simulations. GT defines such models as fast-running-model (FRM). [13]

As the project purpose is a closed-loop simulation with other powertrain components and controllers under dynamic real-drive conditions, the modelling process of the combustion engine as FRM focuses more on the real-time capability and the system coupling rather than simulation accuracy. As the model overview shown in Figure 3, the intake and exhaust tracks are reduced to 6 sub-volumes between the air filter, the compressor, the cylinders, the turbine and the environment. The turbocharger maps and the intake/exhaust flow discharge coefficient over valve lift are measured on a component test bench. The heat transfer inside the cylinders is modelled using heat transfer formula based on the classical Woschni correlation without swirl, but with small modifications. Wiebe parameters imposed as multi-dimensional look-up tables are used for combustion processes [14, 15, 16].

A high-level controller is used to determine the required fuel mass flow with respect to the current air mass flow through the throttle and the imposed air-fuel equivalence ratio.

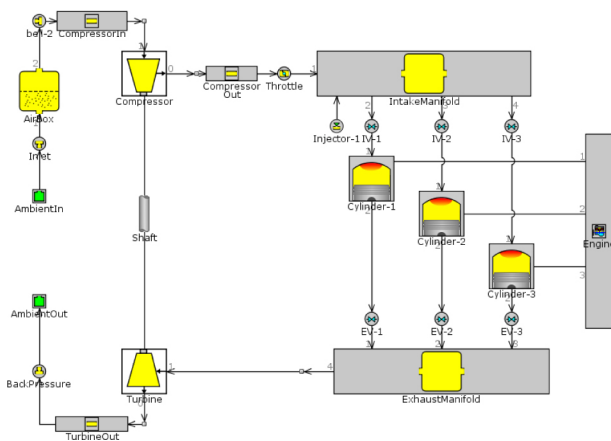


Figure 3. GT Fast-Running-Model of a three-cylinder turbocharged gasoline engine with port injection

Up to an engine speed of 7000 1/min, the model runs stable with the GT-RT solver using an integration time step of 0.25 ms. It runs two times faster than wall-clock time on a normal Windows laptop with an Intel Core i5 processor and can be executed on the present HiL simulator without any overrun problem.

The engine model is calibrated stand alone with the steady-state measurement data from the engine dyno test bench for combustion, pressure drop and heat transfer. In Figure 4, a comparison of simulated and measured brake specific fuel consumption (BSFC) is shown to represent the model accuracy at steady-state conditions. The simulation shows good agreement with measured data for the most relevant engine operating range during NEDC.

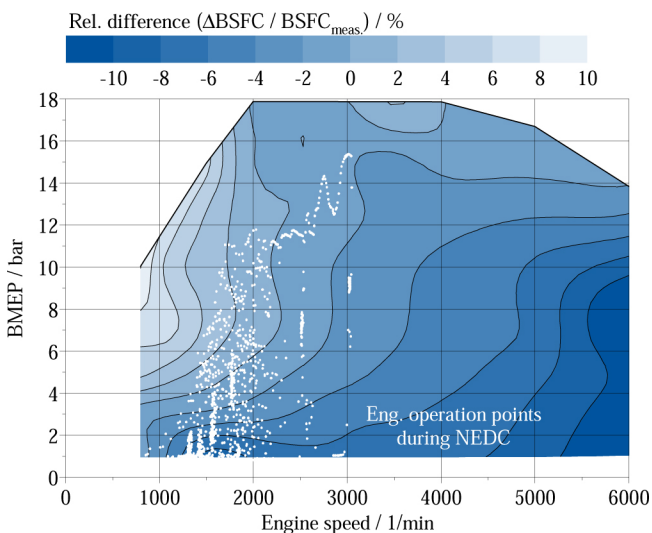


Figure 4. Percentage difference between simulated BSFC and measured data from an engine dyno test bench, with engine operating points during the NEDC from vehicle measurements on a rolling road

After the steady-state calibrations, the model is also tested on its own for simple transient operations, e.g. load step at constant engine speed. The accurate transient response of the turbocharger is vital for the dynamic behaviour of the vehicle and further influences the driver's reaction and the gear change. A comparison between the simulation results and the engine test bench measurements with an

accelerator pedal position step from 15 % to 100 % at 2000 1/min is shown in Figure 5. The engine model shows good congruence with measurement data here.

For the powertrain co-simulation, the engine model is exported from GT-Power and simulated together with the other models in VEOS. An idle controller is created in Simulink with feed-back and feed-forward control for the throttle and the ignition timing. It is coupled with the engine model to ensure a stable idling speed and a smooth launch process in the dynamic driving cycle.

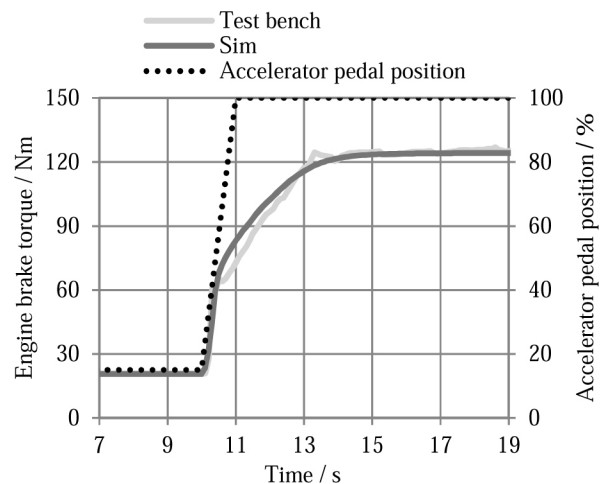


Figure 5. Comparison between a simulated load step at 2000 1/min and measured data from an engine dyno test bench

Transmission

The reference vehicle has a rear-wheel drive with a 6-speed automatic double clutch transmission. The transmission and the drivetrain are modelled with the commercial simulation tool SimulationX from ESI. SimulationX is a system simulation tool that supports signal oriented and acausal modeling using Modelica [17] as the modelling language. It comes with ready-to-use component libraries from different physical domains. For the transmission model, elements from the 1D Rotational and from the 1D Transmission Elements libraries were used. Figure 7 shows the model structure.

The model is stimulated by the engine torque signal; the computed engine speed is fed back to the engine. Stiffness, inertia, damping and friction losses of the dual-mass flywheel are included. The clutches are modelled as rigid friction elements which take sticking and slipping into account. Both clutches are controlled by the TCU through their maximum transferrable torque which is equivalent to the torque capacity or the clamping pressure. The gearbox model (see Figure 6) consists of the inertias, the gear ratios and real-time capable synchronizer elements.

The synchronizer elements do not include the contact forces based on the geometry of the teeth. This would result in higher dynamics and larger computational effort for the model which would affect its real-time capability. Such a high level of detail is needed for vibrational or shifting comfort analysis. In this MiL and EiL application, behavioral models based on elastic force elements