A High-Resolution Warm-Up Simulation Model for a Gasoline Engine with Advanced Thermal Control

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ABSTRACT

A simulation model is presented for the prediction of the warm-up behaviour of a modern turbo charged 3-cylinder gasoline engine. The engine under investigation is equipped with advanced thermal management technology, namely split-cooling and variable thermostat control.

The presented simulation methodology is based on a high resolution, one-dimensional thermal and flow transport approach. For the prediction of heat rates due to combustion a detailed model for thermodynamic cycle simulation is added. Lumped masses and generic finite element models represent the engine structure. Special attention is paid to a strategy for a time and cost efficient model creation.

1 INTRODUCTION

During the last decade powerful simulation techniques for vehicle cooling and engine thermal management analysis have been developed. Most of them have been verified and enhanced in the scope of special validation projects. Some of them had proven their reliability in standard development processes such as for vehicle cooling systems design. However, for the special field of warm-up simulation of engines, simulation methodologies still need improvement to become more accurate with respect to fuel consumption forecasts and compatible with the virtual development process.

Reviewing recent publications [1, 2], the developments of two different main stream concepts for warm-up simulation can be observed. The first concept is based on a coarse representation of the engine structure, where the engine structure is represented by only a few lumped masses. Heat input to and heat exchange between the masses is calibrated by extensive vehicle tests on a dynamometer. This approach is often used by coolant systems suppliers without access to detailed engine data.

The second main stream concept is based on a high resolution representation of the engine structure, the water jacket and the lubrication system with less calibration effort and higher accuracy. The fine grain approach is used by engine developers to assess friction losses and fuel consumption potentials throughout the development process.

In the frame of this investigation a fine grain warm-up simulation model [3] has been developed towards higher resolution to investigate the warm-up behaviour of a three-cylinder gasoline turbo-engine with split-cooling functionality. Special attention was paid to the integration of existing simulation modules from a standard virtual development process and to the use of automatic CAD data processing.

2 THE GENERAL CONCEPT OF ENGINE WARM-UP SIMULATION

The prediction of fuel consumption for warm-up drive cycles is generally based on fuel consumption maps measured for the hot engine. If the fuel consumption is recorded over brake load, internal engine friction is included. For the engine in the process of warming up additional friction losses occur, which in the first instance are caused by lubrication friction.

Hydrodynamic lubrication is dominated by oil viscosity and consequently by the oil temperature. To predict the oil temperature during the warm-up process the heat rate through combustion to the walls of the combustion chamber and piston must be known or calculated. Further the thermal transport through the engine's structure and water jacket must be accounted for to finally predict the heat input to the oil and the oil temperature.

To deal with thermal management issues frictional losses must be broken down to the engine's different frictional groups which can be derived as a function of oil temperature from strip-down measurements. To consider thermal management concepts with an initial no-flow strategy for the coolant - split cooling in this case - the liner's structural temperature must be captured as well to correctly predict friction of the piston assembly. The use of empirical relations for the prediction of friction can be helpful, but needs calibration as well and thus cannot substitute strip-down measurements.

3 THE INTERACTION OF SIMULATION MODULES

To predict fuel consumption for different thermal management concepts a set of interacting simulation modules is needed for the heat input through combustion, for heat conduction through the engine's structure, for flow and thermal transport through the coolant and oil circuit. Besides a drive performance module is added to calculate the engine load for various drive cycles and vehicles.

The current simulation model is grouped into six modules, see Fig.1. The design of the modules is carefully chosen so that the overall model can be well organized and allows the integration of other modules like ECU control, hybrid drive-line or refrigerant circuit. Before some of the modules are described in detail, the interaction of the modules, i.e. the exchange of information is discussed in the following.

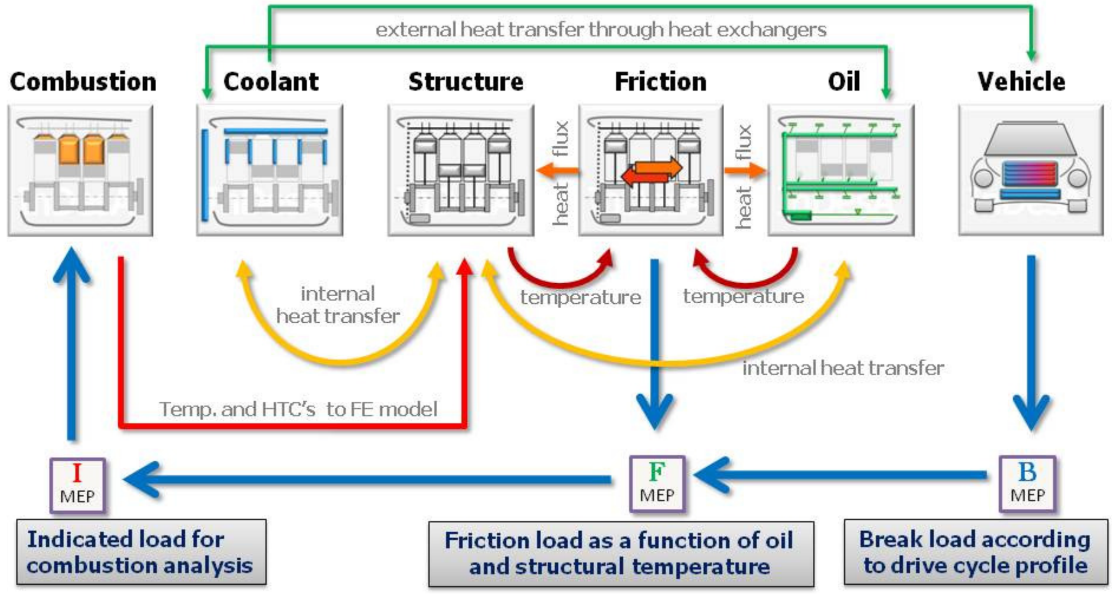

Figure 1: Interaction of simulation modules

As the drive cycle, the vehicle and drive-line are defined, the brake load of the engine can be simulated by either prescribing the vehicle speed and transmission shift profile or by a virtual driver actuating accelerator and brake pedal. For the simulation of conventional drivelines the first method can be used while for more complex configurations like hybrid drivelines the latter approach is more applicable.

Having calculated the brake load the friction load for the engine is predicted and added for the initial starting temperature. The frictional load is interpolated from maps derived from strip down measurements. Brake load and frictional losses add up to the indicated load, on which the calculation of the heat input to the walls of the combustion chamber is based. Unlike in most other known methods, the heat rates are not derived from measurements, but from engine thermodynamic cycle simulation.

For the integration of thermodynamic cycle simulation three methods have been investigated: direct coupling, a mean-value, and a map-based approach. For any approach a detailed 1D cycle simulation model is mandatory. In the first place, the cycle simulation model can be directly coupled to the warm-up model. However, due to the smaller time-step of the cycle simulation, the accumulated simulation time for the whole drive-cycle becomes extensive. Currently a fast running model is under investigation as well. Alternatively the cycle analysis can be performed beforehand for a matrix of operating points, i.e. over IMEP and engine speed, and the results are stored in a map to be looked up in the following warm-up simulation. Those look-up maps typically contain the average gas side heat transfer coefficients (HTC) and the combustion gas temperatures for different sections of the combustion chamber. This approach is the fastest in terms of compute time and is reasonable if the interaction of combustion and other components of the warm-up systems is week or not considered.

If the interaction is not negligible as is the case for the inclusion of EGR cooling and if direct coupling is not desired a third approach, the mean-value approach, can be chosen. Here a detailed cycle analysis model is simplified to allow higher computational time steps. The information lost due to the model simplification must be compensated by the use of neural networks, being trained to accurately predict the HTCs and gas temperatures. Like the detailed cycle simulation model the mean-value model can be directly coupled to the engine's warm-up model but is faster by approximately one order of magnitude.

Whatever approach is used, the HTCs and gas side temperatures are passed to the structural module to be mapped on to the inner surface of the combustion chamber, where instead of lumped masses a detailed parametric finite element cylinder model is used. It includes liner, head, valves, ports, and moving piston and rings for every cylinder. The gas side HTCs and temperatures generate the heat rate into the engine structure, where it is absorbed and distributed to the oil and coolant, allowing both fluids to warm up. The oil and structural temperatures are used to update the frictional losses for the different frictional groups, which consequently has an impact on the operating point of the engine, i.e. the indicated load.

The frictional load does not only change the operational point and hence the thermodynamic efficiency of the engine but also generates dissipated heat which is fed back to the structure and the oil. Coolant and oil can exchange heat through the structure, especially the cylinder head but also through heat exchangers included in the system.

As the main thermostat opens at the end of the warm-up drive cycle the radiator in the vehicle's front-end is finally involved. The vehicle module includes therefore a model for the air path through the car's frontend and radiator, to account for the heat release to the ambient.

4 HIGH-RESOLUTION SIMULATION MODULES

For the investigation of the three-cylinder gasoline turbo-engine the GT-SUITE software was chosen, where the described module structure was directly implemented (Fig 2). Most advantageous was the availability of a detailed GT-POWER part load model for thermodynamic cycle simulation. This model has been used to create the look-up map for the HTC's and gas side temperatures. It has also served as a basis for the mean-value approach, but otherwise will not be described in detail in the scope of this paper.

As the investigated engine includes a split cooling concept, where the coolant flow in the cylinder block is controlled by an additional thermostat, great care has been taken to model the water jacket, the engine structure and lubrication friction, leading to a high-resolution approach.

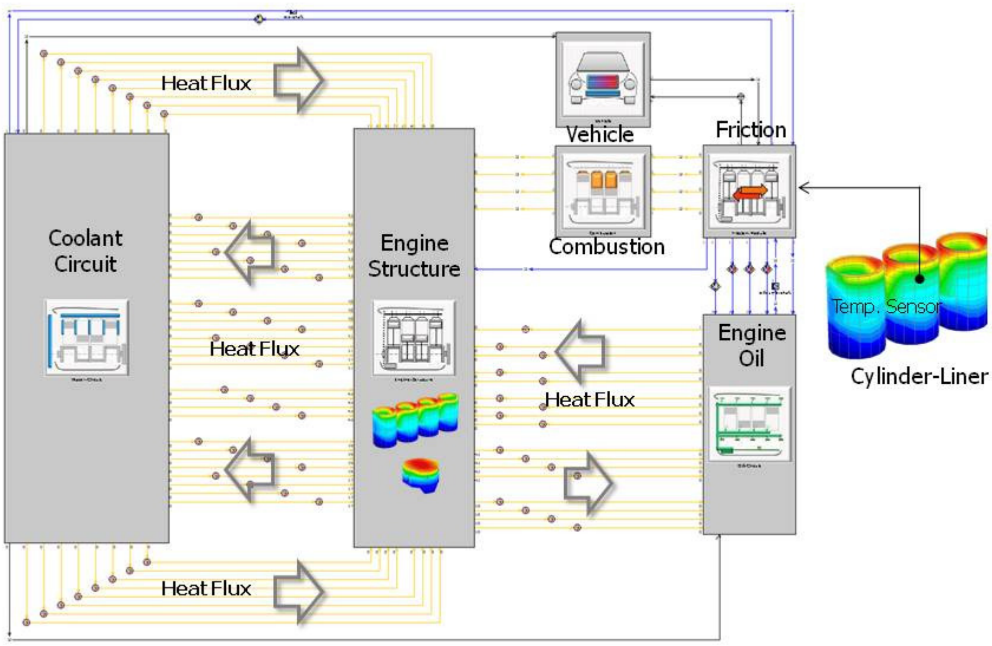

Figure 2: GT-SUITE Warm-up simulation model

4.1 The Coolant Module

The water jacket of the engine is split into two zones (split cooling). The first zone which is controlled by an additional thermostat consists of the crankcase water jacket and the connected cylinder head water jacket on the intake side. The thermostat which is placed right behind the water pump at the engine inlet shuts off the coolant passage to the crank case when the engine is cold. However, the coolant can pass directly to the second zone, the water jacket of the exhaust manifold which is integrated into the cylinder head [4]. From there the coolant approaches the exhaust ports and exhaust valve bridges. Until a defined coolant temperature is reached the coolant in the crank case is at rest, allowing the cylinder liner to warm up more rapidly, hence reducing frictional losses for the piston assembly.

The water jacket includes complex, three-dimensional flow passages, which are difficult to capture with one-dimensional flow objects. It was therefore decided to back up the analysis with a three-dimensional CFD flow calculation, which is generally available in a standard engine development process. The CFD analysis is used for the calibration of the one-dimensional model in terms of flow distribution, pressure losses and heat transfer coefficients (HTC).

However, the calibration effort was kept low by use of a rather detailed one-dimensional flow model backed up by the CAD data processing feature GEM3D included in GT-SUITE. With the help of GEM3D CAD data could be directly processed and the water jacket was cut into sections as shown in Figure 3. In a next step all sections were automatically converted to one-dimensional flow objects, preserving all flow connections in terms of cross-sectional flow area and direction, the volume and wetted surface, as well as different characteristic lengths of the section. That way the one-dimensional objects account for three-dimensional conservation of momentum. It was found that the one-dimensional results needed only little calibration at a few locations to match CFD results.

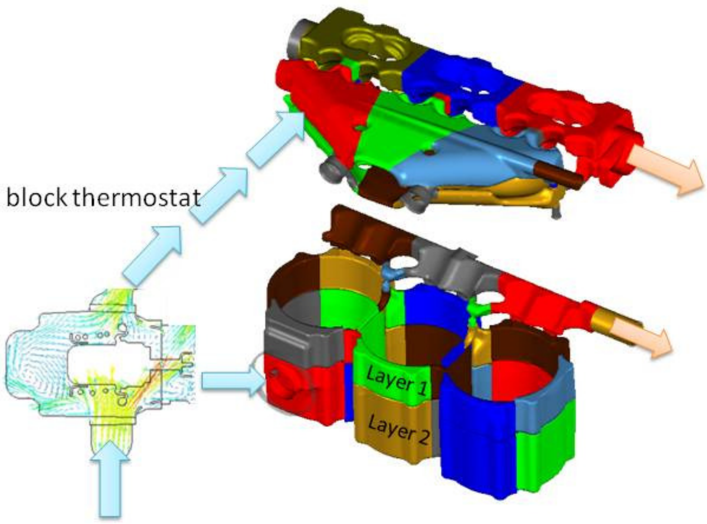

Figure 3: Sectioning of the water jacket for 1D analysis with GEM3D

The use of GEM3D for CAD data processing led to significant time savings for model building and calibration. Besides GEM3D processed flow models are more accurate and less prone to user errors. The only expertise needed is to reasonably split the water jacket into sections and place the cuts accordingly.

The coolant circuit is completed with a water pump, the main thermostat at the outlet of the engine, a heater and oil-coolant heat exchanger, the turbo charger water jacket and the radiator. All parts except the radiator are included during the warm-up phase of the engine.

4.2 The Engine Oil Module

Similar to the coolant circuit, the oil circuit is simulated as a closed circuit with a pump powering the system. For warm-up simulation there is no need for a detailed hydraulic model. It is sufficient to account for the proper thermal energy transport.

For the geometrical set-up of the model GEM3D is used, where with little user interaction all flow passages are automatically identified and transferred to one-dimensional flow elements. Since the flow rates for all oil consumers were known from a detailed hydraulic simulation, oil consumers of the same kind could be grouped together, e.g. all main bearings.

The engine under investigation is equipped with a volume flow controlled oil pump with modulation options for the control pressure during warm-up. Since the oil flow rates were prescribed for all consumers, the oil pump could be simplified to operate with a small surplus volume flow rate and a pressure relieve valve to maintain the target pressure in the main oil gallery.

On the whole eight different oil flow consumer or groups are included in the simulation model with oil return passages to the oil pan. On the high pressure side pressure losses were included for the oil filter module. The heat transfer in pipes is directly calculated in GT-SUITE, whereas the HTCs in the return passages are based on user Nu-Re-Pr correlations. The heat exchange between oil and water in the oil-water-cooler is calculated on the basis of supplied heat transfer maps.

4.3 The Structural Module

To predict frictional losses of the piston group the structure of the piston and the liner must be included. Since the temperature of the liner varies significantly over its height and is affected by the attached water jacket it is not reasonable to work with the concept of lumped masses. Instead generic finite element models are used for the structure close to the combustion chamber. The generic FE-cylinder model provided in GT-SUITE includes the liner, piston, piston rings, flame deck, intake and exhaust ports as well as valves. Predefined ports can be used to connect the FE-model to the coolant, oil and other structural elements. The gas side boundary conditions, i.e. the gas temperature and HTCs are mapped directly to the different sections of the FE-model. Since GT-POWER includes identical object, the mapping is done fully automatic.

For the current study, a FE-cylinder model was used for each of the three cylinders whereas the outer engine structure and all shafts were modelled by use of lumped masses. It was found that lumped masses need higher calibration effort and are not as accurate as finite element objects.

4.4 The Engine Friction Module

The frictional module is entirely based on experimental data. Strip-down measurements are analyzed to derive the frictional losses for the different frictional groups of the engine. Since the engine is motored, the resulting frictional losses for the piston assembly need to be corrected by a load factor. For the other frictional groups no correction is necessary since most warm-up drive cycles require only low part load operation.

Although the effort to perform those measurements is significant and time consuming it is often part of a standard development process. Thereby the following additional requirements must be met: The engine should be equipped with thermocouples at various positions in the liner to record the structural temperatures. Apart from the oil temperatures, the liner temperatures are used to define the frictional losses for the piston assembly. This is especially significant for the investigated split-cooling concept, where the temperature of the liner is the root cause for friction reduction.

To control the liner temperature the coolant temperature must be varied in addition to the oil temperature at least for the investigation of the piston assembly. If the coolant flow rate is high enough the liner temperature is equal to the coolant temperature. Figure 4 shows the frictional torque for the full engine and the piston assembly for different oil/coolant temperatures. As the frictional torque curves are almost identical for the temperature pair 30/30°C and 30/60°C it becomes evident, that the water/liner temperature is dominating friction of the piston assembly and not the oil temperature in the oil gallery or oil sump.

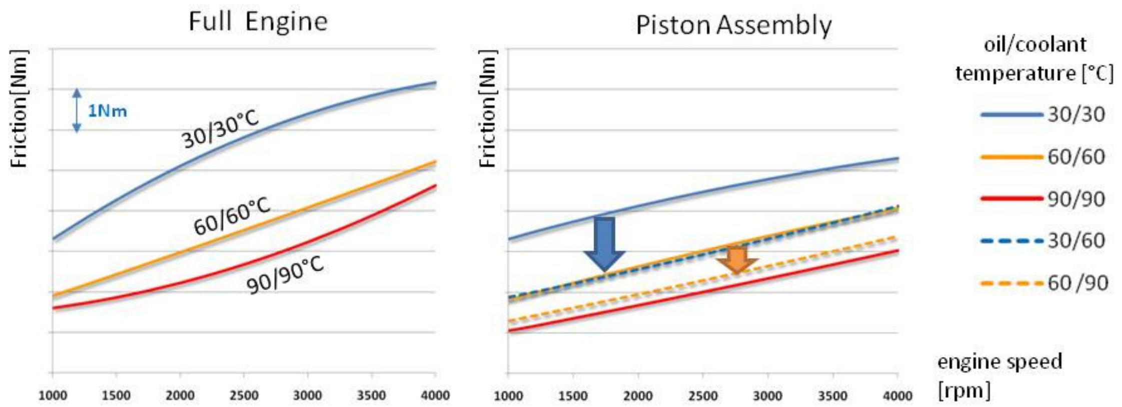

Figure 4: Strip measurements for full engine and for piston assembly

Power losses for the oil and water pump are directly derived from the hydraulic system simulation divided by the pump efficiency. Other accessories, like the generator need to be defined with respect to the envisaged impinged load for the test cycle.

5 MODEL CALIBRATION FOR STATIONARY OPERATION

As the simulation model is build up module-wise, calibration is done for each module before the assembly is being calibrated.

Due to the split-cooling concept of the engine special attention was paid to the water jacket. With the help of 3D CFD analysis the flow distribution in the engine was calibrated for the open and closed position of the block thermostat, i.e. with and without flow in the crankcase water jacket. For stagnant coolant in the crankcase, measurements on an engine test bench were used in addition to calibrate heat transfer. Two layers of 1D elements were inserted for the crankcase water jacket to account for the temperature stratification during the warm-up with stagnant coolant (see Fig. 3).

The assembled model was first checked against the hot engine’s fuel consumption rate measured for stationary operation to verify the overall correctness of the induced load. Subsequently for the same operation conditions warm-up tests were performed, to calibrate the instantaneous fuel consumption rate and fluid temperatures for the two cases, i.e. with open and operating block thermostat. For the latter case the block thermostat opens at a coolant temperature of 70°-75°C.

Figure 5 shows the calibration for fuel consumption of the engine for steady state low part load operation. This operating point is considered to be an average load representing the NEDC drive cycle, thus leading to similar warm-up behaviour. The results for the open block thermostat are calibrated and are therefore in good agreement with experiments. Without further calibration, the case with operating block thermostat was simulated showing good agreement with experimental data as well. In that case the model is to be expected fully predictable.

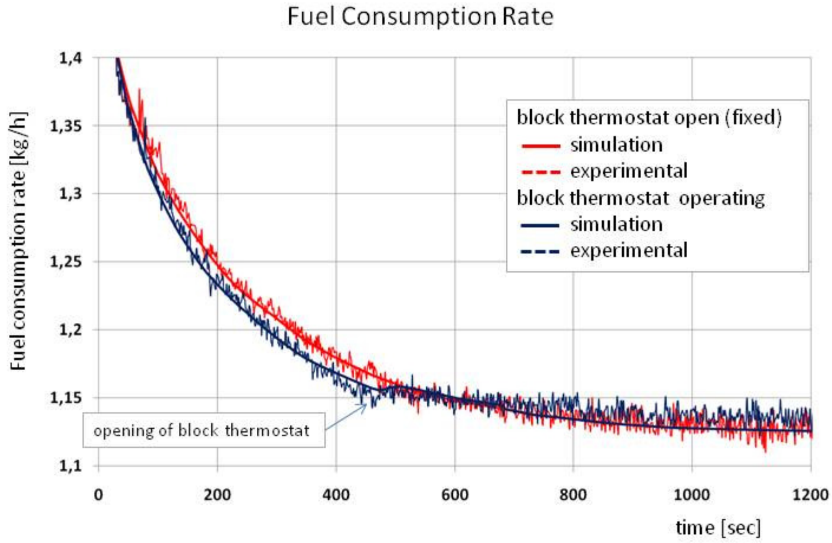

Figure 5: Fuel consumption rate for steady state operation with block thermostat open (blocked) and operating (opening at 70°C)

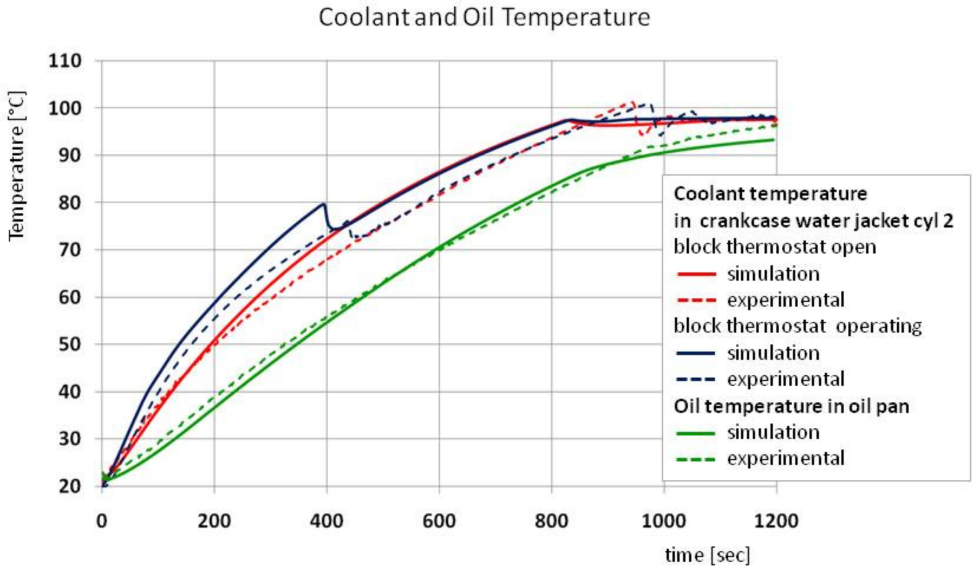

Figure 6: Coolant and oil temperature in crankcase water jacket for steady state operation

For Figure 6 the coolant temperatures are recorded at the second cylinder on the intake side as well as the oil temperature in the oil pan. Again good agreement is achieved between simulation and experiment for this operation point. It is advisable to calibrate further stationary operating points and check on the fluid temperatures before applying the simulation model to fuel consumption drive cycles.

6 WARM-UP SIMULATION FOR THE NEDC DRIVE CYCLE

As model calibration for different operating points is passed, the warm-up simulation model is ready to use for the prediction of fuel consumption for various part load drive cycles. The drive cycle under investigation is the NEDC cycle, performed on a cold vehicle at 20°C with four repeated urban driving profiles followed by one extra-urban profile. Although the simulation model can be used for the investigation of various thermal management features, the results presented are focused on the split-cooling concept.

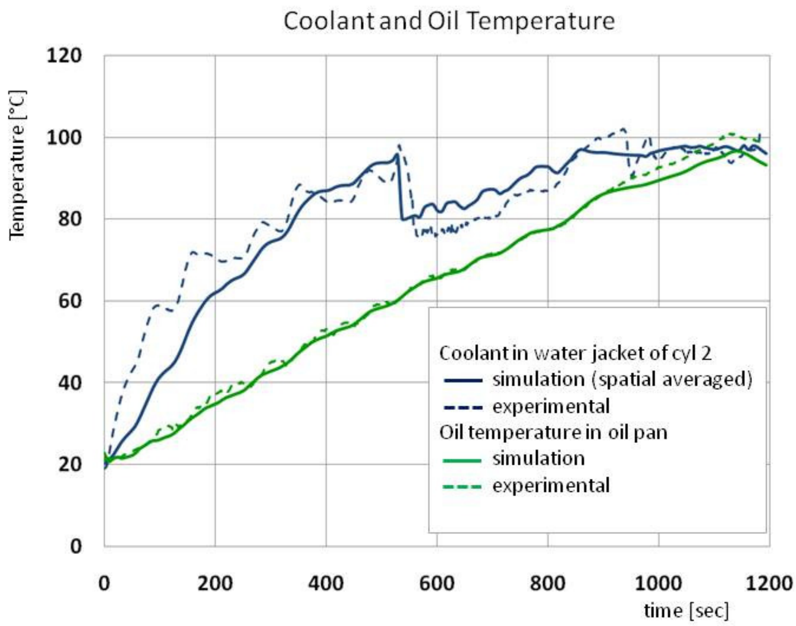

Figure 7: Coolant and oil temperature with operating block thermostat (split cooling) for the NEDC drive cycle

Figure 7 shows the temperature profiles for the coolant and the oil for split cooling operation, i.e. with the block thermostat opening at 70°C. The coolant temperature is monitored in the crankcase water jacket at the second cylinder. As the coolant remains stagnant in the crankcase for the first 550 seconds, the temperature rise is 15°- 20°C above the case with open block thermostat thus leading to a benefit in fuel consumption for the NEDC drive cycle of 0.8 %. As the resolution of the water jacket is restricted to two layers in height (Fig. 3) for the simulation, the temperatures are averaged values for that volume whereas the experiment shows the local temperature at the position of the thermocouple.

The main thermostat which opens at 90°C at the end of the driving cycle is simulated by an ideal PID controller, hence not revealing the typical oscillating flow behaviour caused by the real thermostat.

Figure 8 shows the benefit of frictional torque for the piston group. Until the opening of the block thermostat an advantage of 0.5 Nm can be achieved with split cooling. From the curve characteristics it can be concluded that the advantage of split cooling could be extended to a longer period of the NEDC drive cycle. However, this would require an additional temperature sensor in the crank case water jacket for boiling control and a faster reacting thermostat or valve to prevent overheating for sudden full load acceleration.

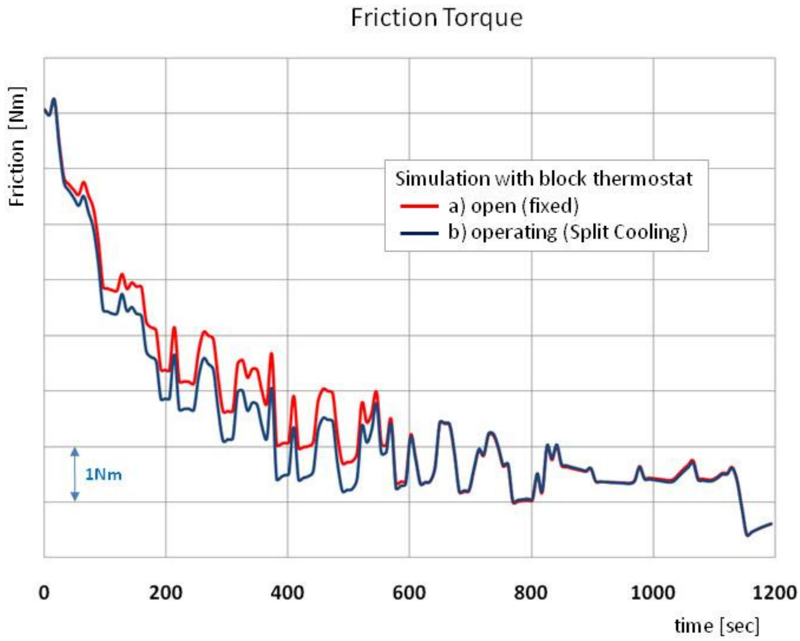

Figure 8: Friction torque of piston assembly for the NEDC drive cycle

7 CONCLUSION

A simulation methodology has been extended for the prediction of fuel consumption of a 3-cylinder gasoline engine with a split-cooling concept for warm-up drive cycles. As the investigated split-cooling concept acts on the cylinder liner temperature, a high resolution approach with integrated finite element models was used to accurately account for the thermal behaviour of the piston assembly.

The calibration effort was found to be reasonable since most of the required experimental data are derived from standard measurements carried out throughout the engine development process. Continuous calibration of the simulation model for consecutive development stages leads to good agreement with test bench results.

Finally the high resolution modelling approach can be used to derive and calibrate engine warm-up simulation models of a much coarser representation of the engine structure and fluid circuits. Such coarse grain models can be provided to coolant system suppliers reducing their effort of calibrating self made engine models by costly vehicle dynamometer tests.

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