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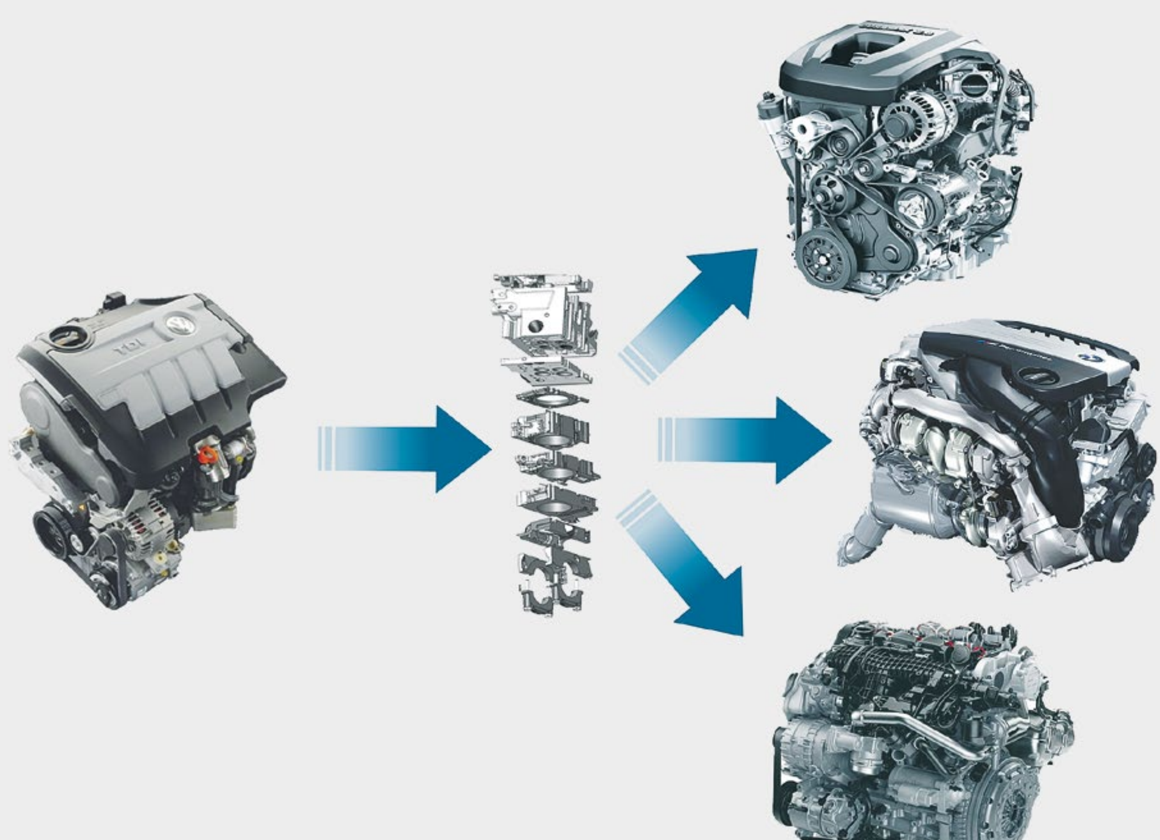
Simulation of Heat Transfer with Variation of the Operating Conditions

The future challenges in the area of the upcoming emission and consumption regulations for light duty engines can only be overcome with the help of an optimised warm-up phase. This requires an intelligent redistribution or use of the heat generated during the combustion process. All these measures, which are taken in order to optimise the warm-up phase or to improve the overall temperature distribution, are referred as thermal management. The aims of the thermal management are the optimisation of component temperatures, friction, emissions and fuel consumption. In the FVV project engine heat exchange, a thermal 1-D simulation model was created, which enables a preliminary calculation of the thermal behaviour by a diesel engine during the warm-up phase, thereby providing the groundwork for reducing consumption. Based on these results, at IMS the research project Engine Heat Exchange III was initiated to extend the model, perform thermal tests and examine a model transferability to other engines.



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1 MODEL EXTENSION TO CO-SIMULATION

The thermal network described in [1] allows heat distribution in the components, media and friction agent pressure to be determined. A combustion engine is predominantly heated by the heat input from the combustion or the friction processes. As both losses, wall heat and friction loss, change significantly over the course of the heating phase, this should be better considered in future. The heat input, in relation to load and speed, was realised using static characteristic maps in the initial model. This characteristic map approach is frequently used when forming thermal engine models, for instance in [2]. Parallel to the heat flows of the characteristic maps, the friction agent pressure was calculated according to Schwarzmeier, this being used to determine the frictional heat. In the original model, the friction did not have any effect on the load point displacement, especially at the beginning of the warm-up phase. If specific processes in the warm-up are to be examined at the beginning of the heating phase, a reciprocal exchange of thermal network and combustion model is required. It must be borne on mind here that the different time periods of the individual models (cycle-resolved in comparison to total simulation time) have to be linked numerically. Besides consideration of the load point displacement, it is possible to examine diverse injection strategies, catalytic converter heating, the influence of injection volume distributions, load change or mixture formation effects and emissions in the warm-up phase. In addition, all combustion process parameters can be evaluated as high resolution.

2 SIMULATION ACCURACY

Following the model extension, a renewed model validation was necessary. The stationary warm-up performance at constant operating points was analysed and compared with the results of the simulation for this. Taking load point 1500 rpm and 50 Nm as an example, **FIGURE 1** shows a comparison between the simulation results of both models with measured values from the test bench. A significant temperature rise was revealed by the measurements at the start of the test. A high heat input into the components and coolant occurs here due to the load point displacement. This behaviour can be observed most prominently at the heat sinks, which are closer to the combustion, for example the cylinder head, cylinder liner or the coolant. The engine oil, which is further from the actual heat source thermally, tends to be slow and subdued in response. The advantage of the coupled model approach was clearly apparent here. The co-simulation model can describe the

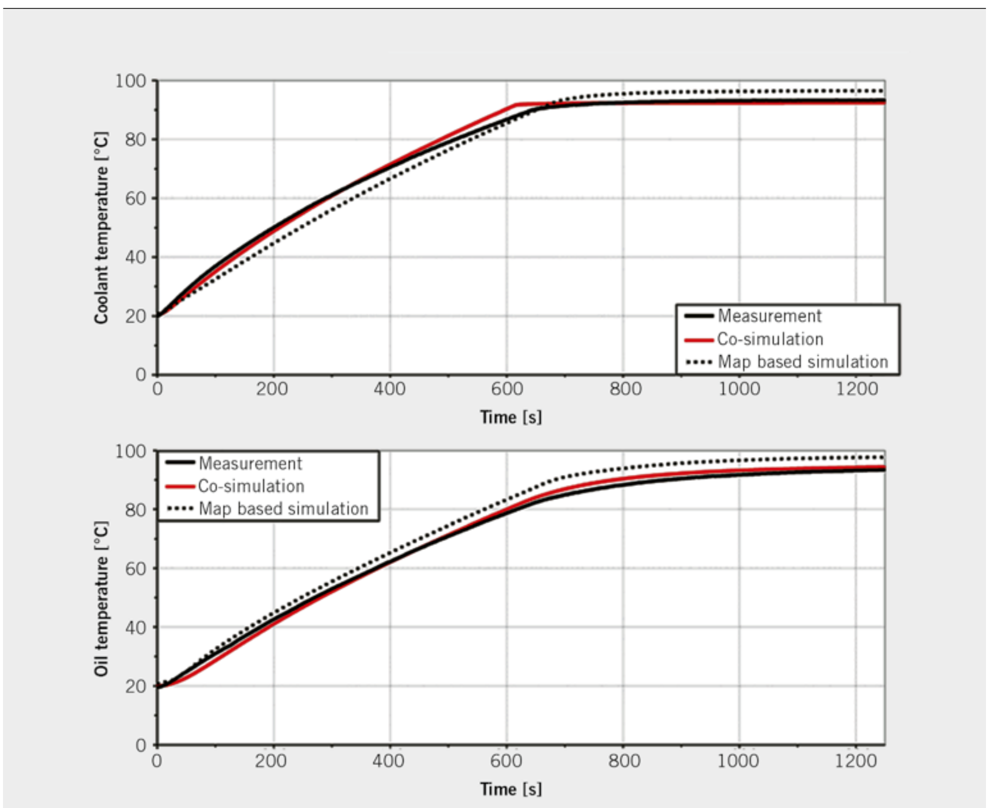

FIGURE 1 Comparison of media temperatures (measured values and simulation) (© IMS)

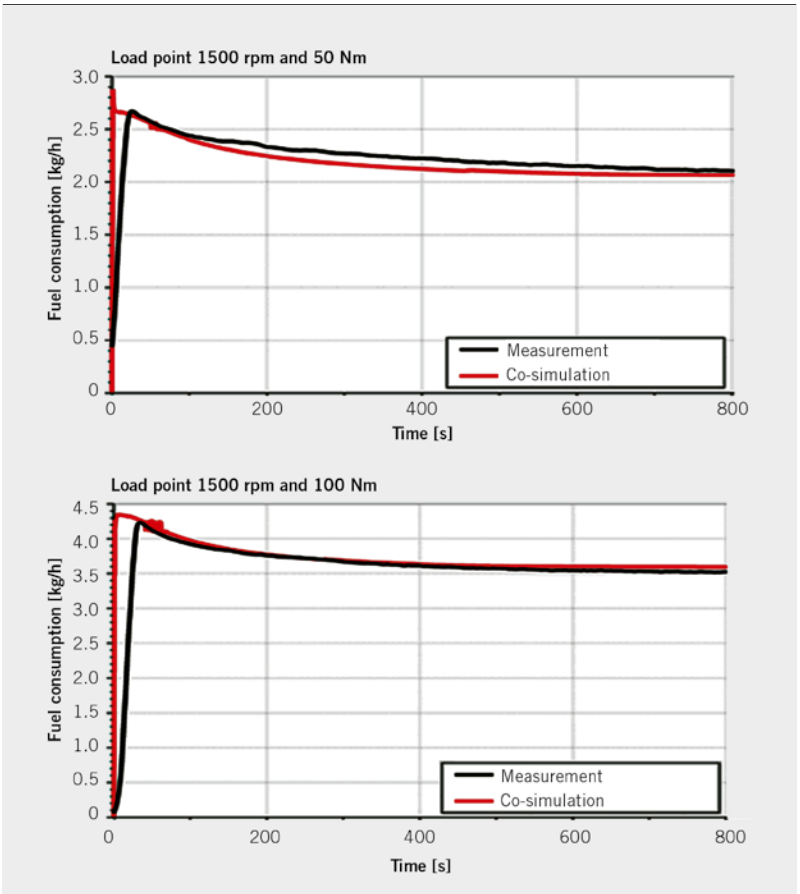

FIGURE 2 Comparison of fuel mass flow (simulation and measured value) (© IMS)

heating performance of the components and media much better in the low-temperature range, which improves the accuracy of the results considerably.

Comparison of the measured values for the fuel mass flow with the results of the simulation for two selected load points in **FIGURE 2** shows a good correspondence. A deviation only occurs at the beginning, which can be justified with the integration behaviour of the fuel scales however. The simulation model can immediately provide a result numerically here. Analyses of thermal management measures in respect to their effect on fuel consumption are possible with the coupled model both qualitatively as well as, with slight uncertainty in respect to the absolute volumes, quantitatively.

3 ENGINE ENCAPSULATION

Individual thermal management measures were examined on the test bench throughout the entire course of the project for the model validation. The measured and simulation results of an engine-remote thermal encapsulation are shown below. This serves for both storing residual heat for an engine restart as well as for reducing the dissipated heat during an engine cold start. Tests with an engine-close, acoustically-optimised encapsulation reveal very good results in respect to cooling performance [3]. Thus, temperature differences of up to 24 K are exhibited over a time period of 6 h. The advantages of an increased start temperature in relation to pollutant emissions and consumption are validated by different test series in many respects. An approach involving an engine-close encapsulation was avoided owing to the complex measuring technology on the test bench. The test vehicle was encapsulated with several 20 mm thick plates of the insulation material Theta-FiberCell 1330gsm made by Autoneum for the test.

3.1 TEST BENCH RESULTS

In the first step, the insulation effect was examined in relation to a reduction of the dissipated heat during a cold start (load point 1500 rpm and 50 Nm). It was discerned that the encapsulation had hardly any effect on the heating performance. Minor advantages only became manifest at the end of the heating phase with greater temperature gradients on the engine exterior. However, the early heating phase with its highly pollutant-forming effect could not benefit from this. Following this, subsiding tests from the operationally warm state were performed. With reference to the oil temperature, which has the greatest effect on the friction, a temperature advantage of approximately 16 K can be determined after 4 h. Similarly, all other thermal masses revealed a significantly higher persistence level after 4 or 8 h.

3.2 RESTART AFTER 4 H WITH INSULATION

Advantages due to the insulation could be determined in the subsiding tests. A restart after four hours shutdown time with and without insulation is compared in **FIGURE 3**. The delayed cooling means that all thermal masses have an even higher start temperature. A comparison of the oil temperatures at engine start reveals a temperature potential by almost 17.5 K. The same also applies to

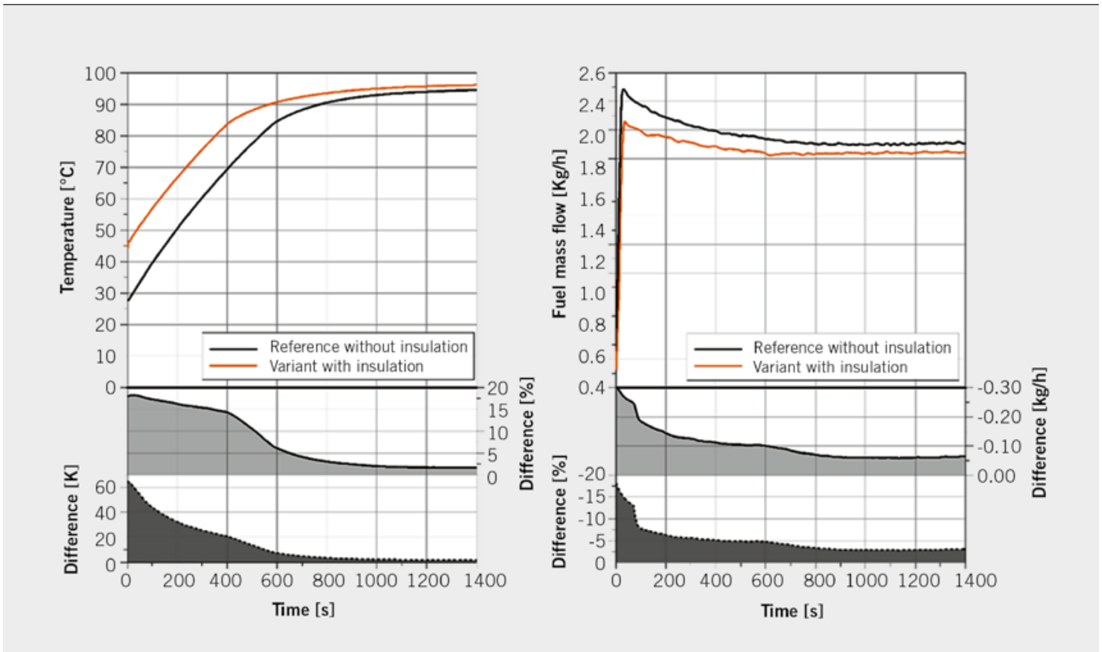

FIGURE 3 Oil temperature (left) and fuel mass flow (right) in comparison (load point 1500 rpm and 50 Nm) (© IMS)

the coolant temperature. Advantages of around 20 K can be ascertained here. The temperature difference in the remaining components behaves in a similar way. With approximately 4.7 % consumption reduction, these large temperature differences can exhaust a large part of the total theoretically attained savings potential of 6 to 9 % in the warm-up phase [4]. Good advantages also result after 8 h due to the degressive properties associated with the viscosity of the oil. Even if the advantage is considerably less here on average with 10 K temperature difference compared to the non-insulated tests, 3 to 3.5 % fuel savings are still discernible.

4 SIMULATION RESULTS

Parameterisation of the heat transfer within the simulation model proves to be extremely complicated owing to the heat transfer coefficients on the complex surface structure and heat distribution with different materials. Reference values from physical tests were used for the non-insulated state. In contrast, no quantifiable measured values were available for the encapsulated test, which meant that several calculations with variation of the heat transfer coefficient were conducted and then compared with the measured results. An averaged theoretical heat transfer coefficient could therefore be determined approximately for the encapsulation, **FIGURE 4**. A comparison of the tendencies (measurement and simulation) at 600 s reveals that the heat transfer coefficient with insulation is around 10 to 20 W/m²K. The effect of insulation on the warm-up phase is very low. Even in case of theoretical full-insulation (unbroken line), discernible advantages only become manifest at the end of the heat-up phase. This coincides with the results of the measurements (dotted lines).

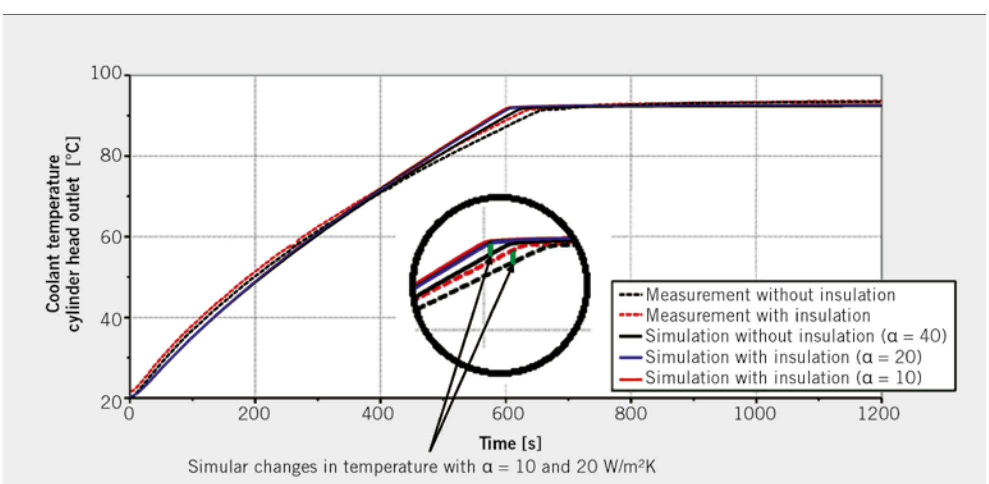

FIGURE 4 Temperature progression of the coolant for various heat transfer coefficients (© IMS)

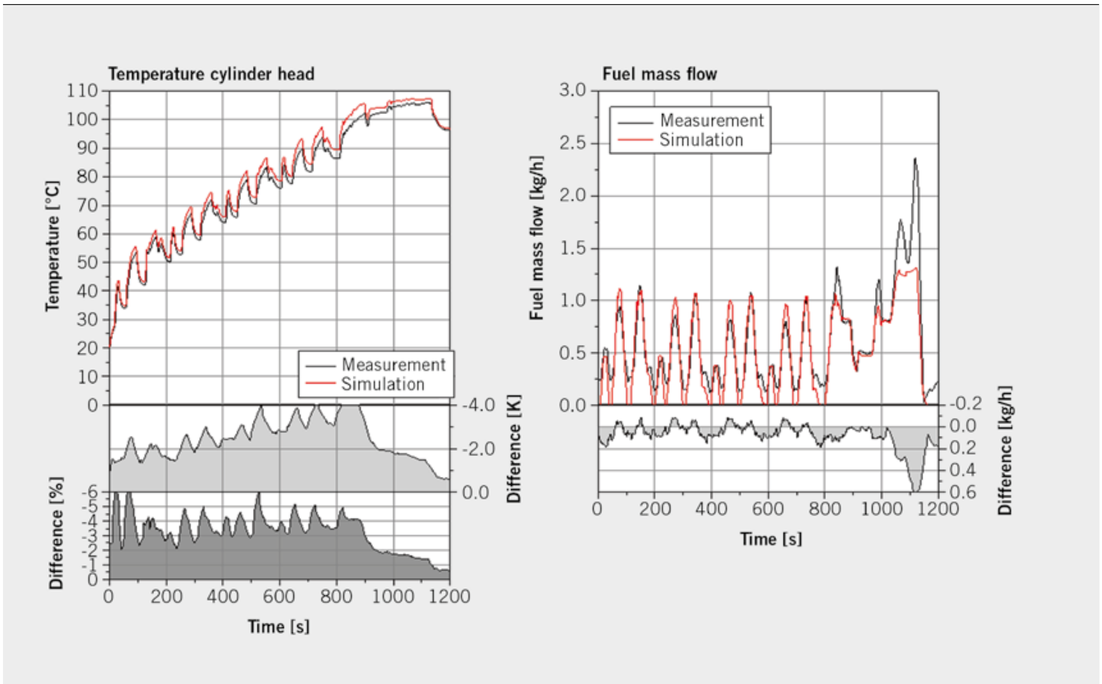

FIGURE 5 Temperature of the cylinder head (fire deck) and fuel mass flow (Ford 2.2 l in the NEDC) (© IMS)

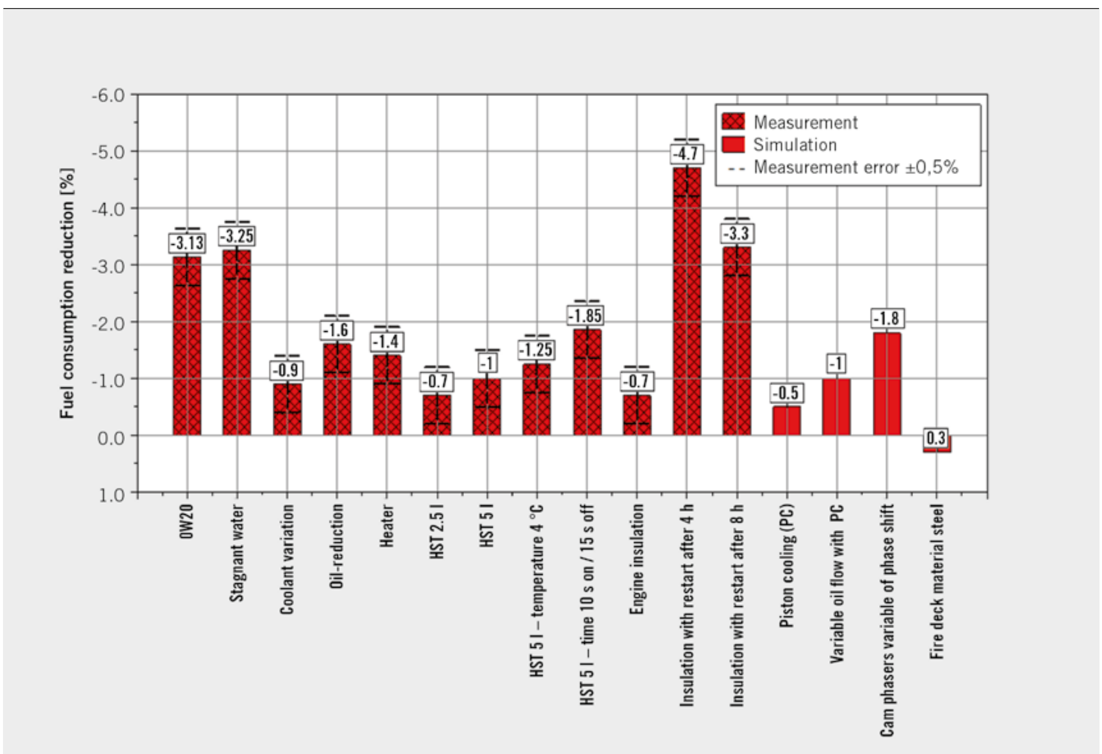

FIGURE 6 Comparison of all examined and simulated measures at 1500 rpm and 50 Nm (© IMS)

5 TRANSFERABILITY OF THE MODEL APPROACH TO OTHER ENGINES

Two further engines (Ford 2.2-l Duratorq TDCI and Volvo 2.4-l R5 diesel engine) were examined during the model validation. **FIGURE 5** shows results for the simulation of a Ford 2.2-l Duratorq TDCI engine in the NEDC. The measured results necessary for this analysis were made available by the respective manufacturer. A comparison of measurement and simulation in **FIGURE 5** shows a good correspondence of the measured and simulation results. A deviation of 3 K from the measured value results on average. It is apparent that the jumps in the heat input can be simulated well by the load change in the model. In particular, the temperature progressions of the fire deck show the good prognostic capability of the model, also during dynamic load changes. **FIGURE 6** shows an overview of all measures examined in the project by direct comparison in relation to a uniform load point. The fuel mass flow in the dynamic cycle could be mapped very well. Only the areas with high transient behaviour through to higher loads are not fully mapped by the model. The reasons for this lie in the simple mapping of the injection characteristics. This comprises the individual strategies of static load points, and is therefore only of limited suitability for highly transient operating conditions.

6 SUMMARY

For a direct continuous calculation of the fuel consumption, it was necessary to couple the thermal network with a cycle-resolved combustion model. Besides a successful revalidation of the simulation model with enhanced prognosis accuracy in the cold start phase, various thermal measures were examined as simulations and also on the test bench. To conclude, a test for the model transferability was conducted. To this end, the warm-up performance of two other engines was calculated and compared with measured results made available. It is apparent that a basic transferability with an average deviation of 4 K can be assumed for similar engines.

REFERENCES

[1] Zeitz, V.; Varga, P.; Tschöke, H.; Schmidt, J.: Simulation des Wärmetransports in Verbrennungsmotoren. In: MTZ 75 (2014), No. 07–08, pp. 74–81
[2] Beulshausen, J.: Thermomanagement als Maßnahme zur Reibungsreduktion von Verbrennungsmotoren. Aachen, RWTH, dissertation, 2014
[3] Duvigneau, F.; Luft, T.; Hots, J.; Verhey, J.: Thermo-acoustic performance of full engine encapsulation – A numerical, experimental and psychoacoustic study. In: Applied Acoustics 102 (2016), pp. 79–87
[4] Richter, R.; Tohnhauser, J.; Schreiner, M.; Steinberg, P.: Wärmemanagement des Kraftfahrzeugs IV: Gesamtfahrzeug – Wärmemanagement mit einem neuen Wärmeübertrager-Prüfstand. Renningen: expert Verlag, 2004

THANKS

The research project "Simulation des Wärmetransports in Verbrennungsmotoren zur Reduzierung der Reibung und CO_2-Emission" (project number 1147) was funded by the Bundesministerium für Wirtschaft und Energie (BMWi) and supervised by the Forschungsvereinigung Verbrennungskraftmaschinen e.V. (FVV). The authors would like to thank for the financial funding and the support of the supervising working group under the chairman Andreas Rundkowski, Volkswagen AG.