Proceedings Paper:

Yuan, R., Price, C., Kasurkar, R. et al. (3 more authors) (2020) Numerical investigation of heat retention and warm-up with thermal encapsulation of powertrain. In: SAE Technical Paper. WCX SAE World Congress Experience, 21-23 Apr 2020, Virtual conference. Society of Automotive Engineers (SAE International) .

https://doi.org/10.4271/2020-01-0158

Numerical investigation of heat retention and warm-up with thermal encapsulation of powertrain

Ruoyang Yuan, Christopher Price, Roshan Kasurkar, Mark Spenley, Nilabza Dutta, Kambiz Ebrahimi

Loughborough University, Jaguar Land Rover, Ltd.

Citation: Yuan, R., Price, C., Kasurkar, R., Spenley, M. et al., "Numerical Investigation of Heat Retention and Warm-Up with Thermal Encapsulation of Powertrain," SAE Technical Paper 2020-01-0158, 2020, doi:10.4271/2020-01-0158.

Abstract

Powertrain thermal encapsulation has the potential to improve fuel consumption and CO_2 via heat retention. Heat retained within the powertrain after a period of engine-off, can increase the temperature of the next engine start hours after key-off. This in turn reduces inefficiencies associated with sub-optimal temperatures such as friction. The Ambient Temperature Correction Test was adopted in the current work which contains two World-wide harmonised Light duty Test Procedure (WLTP) cycles separated by a 9-hour soak period. A coupled 1D – 3D computational approach was used to capture heat retention characteristics and subsequent warm-up effects. A 1-D powertrain warm-up model was developed in GT-Suite to capture the thermal warm-up characteristics of the powertrain. The model included a temperature dependent friction model, the thermal-hydraulic characteristics of the cooling and lubrication circuits as well as parasitic losses associated with pumps. A 23°C WLTP cycle was run via the 1D model, key fluids and solids temperatures around the engine bay calculated at the end of the 1st WLTP cycle were then imported into a 3D heat retention model, in which the transient 3D computational fluid dynamics and heat transfer coupled simulation was initiated to model the full-geometry vehicle for a 9 hours static soak period. The cool-down behaviors of the coolant and oils were predicted from the 3D model and the temperatures at the end of the soak were fed back to the 1D warm-up model to carry out the second WLTP cycle simulation at 14°C ambient condition.

A coupled 1D-3D heat retention modelling method predicted both warm-up and cooldown characteristics to within circa ±3 °C of vehicle test data over the entire ATCT test. The impact of thermal encapsulation was clearly shown, whereby coolant and oil temperatures at the end of the 9 hour soak period were 6°C and 10°C higher with encapsulation respectively, which led to a fuel consumption improvement in the order 1% over the post-soak 14°C WLTP through retaining heat.

Keywords: warm-up, encapsulation, heat retention, fuel economy, heat transfer

1. Introduction

Thermal management and optimisation in modern vehicle design and delivery have real-world impact on climate change and can help improve air quality by reducing emissions from transport. To better represent real-life driving situations and average European temperature conditions, the Ambient Temperature Correction Test [1] is required for the type-approval of light-duty vehicles in Europe for the measurements of CO_2 emissions and fuel consumption, in which a minimum 9 hours vehicle static (engine key-off) soak is introduced after the 1st WLTP cycle at 23°C ambient temperature and followed by a second WLTP cycle at a cooler ambient temperature of 14 °C.

At cold-start condition (i.e. 14°C), engine fluids of increased viscosity at low temperatures cause increased friction and pumping losses. Additionally, engine component of low temperatures results in a lower thermal efficiency of the internal combustion engine [2]. These effects will lead to higher CO_2 emissions of the ATCT cycle compared with the standard 23 °C WLTP, but closer values to the real-life one [3].

In order to reduce the energy consumption and emissions though the engine warm-up process, engine and vehicle thermal encapsulation have been introduced [4-6] to help keep the heat within the engine bay for as long as possible to slower the cooldown effect during the vehicle static soak. Different coverage designs of the encapsulation may have various benefits on the heat retention [6], on the same time, the encapsulation material weight, cost as well as the producibility need to be considered to provide a sustainable heat retention solution.

Computer aided engineering (CAE) provides a cost-effective tool to analyse energy flow and efficiency of the powertrain as well as other subsystems of the whole vehicle system. A combined heat retention and powertrain warm-up modelling approach will be beneficial in the vehicle design process to obtain the evaluation on the encapsulation benefits on CO_2 emissions and fuel consumptions.

Powertrain warm-up is a key phase of current legislative drive cycles which can significantly contribute to tail-pipe emissions due to the powertrain operating away from its target operating point. Cold operating temperatures generate increased levels of frictional losses as well as less efficient combustion [7]. CAE techniques are able to model the transient warm-up and energy consumption characteristics of powertrains, including engine frictional losses, pumping hydraulic power consumption as well as fluid and structural temperatures over time [8]. The optimization of holistic powertrain thermal energy management offers large and potentially cheap efficiency gains, CAE techniques area valuable tool which allow fluid pumping strategies and thermal energy management strategies to be optimized. In the context of the ATCT, a powertrain warm-up model can be used to quantify the fuel consumption benefit of heat retention in the cooldown phase by studying the impact of initialization temperatures on warm-up.

To account of the potential heat retention effect for the underhood region during the 14°C vehicle soak period, numerical methods were developed for both simplified underhood model [9] and detailed full-geometry passenger vehicles [10-13]. The buoyancy-driven convection heat transfer especially at the early stage of the vehicle soak plays an important part in the combined heat transfer process of the engine bay [9,12]. However, to take account of the convection effect requires detailed computational fluid dynamics (CFD) simulation of vehicle front on a component level. The simulation is usually found computational expensive [9-11]. Recent work [12] successfully demonstrated the CAE capability of the heat retention modelling and its potential to be embedded with encapsulation design process [13]. The computing cost was significant reduced and can be achievable for industrial applications. However, the CO_2 emissions and fuel economy were not linked to the encapsulation study in the work.

In this paper, a modern approach was demonstrated combining two CAE environments, a 1D warm-up model in GT-Suite, specific to powertrain warm-up thermal energy system optimisation, and a 3D heat retention model predicting the engine-bay cool-down behaviour. The thermal energy of a passenger car fitted with gasoline engine was studied using the coupled/hybrid CAE approach. The test case examined including two WLTP drive cycles with a 9 hour static soak in between. The CAE results were compared with the experimental data and the thermal encapsulation effect on fuel savings was evaluated. The two CAE models coupled together enabled analysis of the thermal fluids behaviour and energy consumptions over legislative drive-cycles and real-world drive conditions, and the optimisation for sizing and thermal energy management.

Present Contribution

The benefit of thermally encapsulating automotive powertrains is not well documented in the literature and the difficulty associated with vehicle testing over the ATCT means a computational methodology capable of capturing the thermal characteristics and temperature dependent energy consumption would be a valuable contribution to CO2 improvements. The current work aimed to investigate the benefits of thermal encapsulation via numerical methods. The main objectives of the present work can be summarized as follows:

- To adopt a 1D powertrain warm-up model to predict powertrain warm-up characteristics and capture system-level energy consumption including engine frictional losses.
- To develop a 3D heat retention model to predict powertrain thermal behavior under static soak conditions.
- To investigate the potential temperature benefit of improved heat retention from thermal encapsulation via a couple 1D - 3D modelling approach.
- To investigate the fuel economy benefit over defined legislative drive cycles associated with improved heat retention via thermal encapsulation.

2. Numerical Methodology

In the current work, a 1D – 3D coupled modelling approach was adopted to predict warm-up and cooldown behavior of a Jaguar Land Rover powertrain. An example of the ATCT-WLTP drive cycle and the diagram of the coupled simulation process are shown in Figure 1. The 1D warm-up model ran in GT-SUITE to predict the fuel consumption, friction losses of the powertrain during the two WLTC drive cycles (23°C and 14°C, respectively). The end temperatures of the engine components and key fluids (oil and coolant) were imported to the 3D underhood heat retention model to simulate the thermal behaviour of the 9 hours static soak. The results from the 3D model were then imported to the 1D model for the second cold-start WLTC drive cycle simulation. The CO_2 emissions and fuel consumptions were predicted by the coupled process and linked to the specific design of the encapsulation layout.

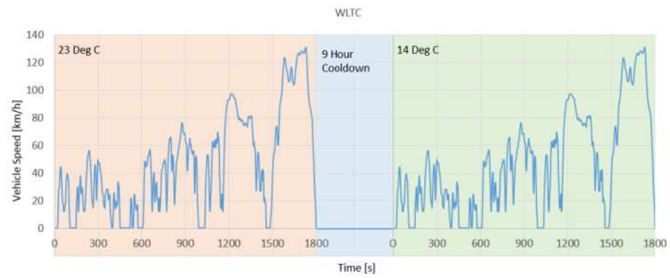

Figure 1. Top: the drive cycles examined and bottom: diagram of the coupled CAE approach.

In the following, detailed information on the model setup is discussed.

1D warm-up model

The one-dimensional CAE software GT-Suite was used for the warm-up modelling phase of the process. The computational tool was used to model all thermal aspects of the engine, including the cooling system, lubrication system, structural masses using a 3D finite-element methodology as well as the transmission oil circuit. Heat input was calculated using a pre-run combustion model over the full engine speed and load range and transferred into the combustion chamber of the structural model. Heat dissipation through the structure was modelled and heat transfer to the various fluids was captured using temperature and flow dependent heat transfer coefficients generated via 3D CFD techniques.

A friction model was incorporated based on pre-generated friction data via a GT-Suite friction model, which included temperature dependent characteristics stemming from oil viscosity and mechanical friction predictions (based on expansion/clearance changes). The data was incorporated as temperature, engine speed and engine load dependent maps for each friction consumer group (piston, cranktrain, bearings, valvetrain). The parasitic power consumption of cooling pumps was also taken into account via 3D CFD generated mapped data at varying temperatures, flow rates and variable pump mode. The isentropic efficiency was adopted to calculate power consumption (note, mechanical friction impact was not included however this was deemed negligible considering the small contribution to total engine friction). Fuel consumption maps were derived directly from engine test data at temperatures of 90°C (coolant) and 25°C (coolant) to allow a temperature dependent fuel consumption prediction to be made. A thermal flow diagram can be found in Figure 2 which highlights the modelled hardware and thermal paths. Three heat sources were captured, namely combustion heat, engine friction and transmission friction. Heat then flowed through the system as shown in Figure 2, ultimately returning to the ambient. Note that heat exchangers such as the transmission oil cooler and engine oil cooler were included to capture oil warming within the cold phase of the WLTC.

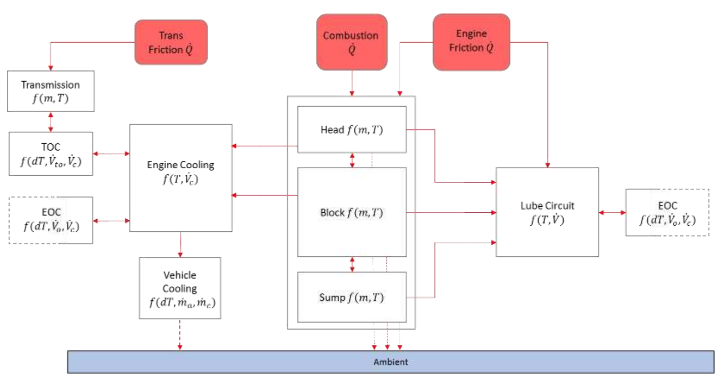

Figure 2. A schematic diagram of the 1D warm-up model architecture. Heat sources are displayed as red boxes, and heat transfer paths are displayed as red arrows.

3D heat retention model

The 3D heat retention model utilize a combined 3D CFD – heat transfer modelling method, in which the buoyancy-driven flow is resolved by the CFD using particle based Lattice-Boltzmann Method [15,16] provided by PowerFLOW, SIMULIA, and the resolved external (surrounding air flow to the solid surfaces) convection coefficients and flow temperatures are imported as boundary conditions for the detailed heat transfer calculation from the 3D heat transfer model using PowerTHERM, SIMULIA. The heat transfer processes in between the surrounding air, engine solid and the internal fluid, coolant and oil, are calculated including the convection, conduction and radiation effects. The resolved surface temperatures of the components were then fed back to the 3D CFD to calculate external flow behaviour of the next time interval. This process iterates for several cycles of customised time periods until the second stage of the 3D heat retention modelling in which a fast standalone heat transfer model is used to reduce the simulation time. The detailed information on the coupled method can be referred to the ref. [12].

The computing costs for the 1D warm-up model takes 8 (×1 CPU) hours to run for a WLTP cycle with the finite element structure, which is reduced to 1 (×1 CPU) hour with a lumped mass structural model. The 3D heat retention model takes 41 (×384 CPUs) hours to run for a complete 9 hours soak simulation.

3. Results and discussion

In this section, simulation results of thermal fluids behaviour over the first 23°C WLTC drive cycle by the 1D warm-up model were first discussed, followed by results of the heat retention modelling of the static soak. Next, the warm-up performances from the encapsulation heat retention case and from the 14°C baseline conditions were compared. Contribution to the CO_2 emissions and fuel consumptions was discussed in the end.

1D warm-up modelling – fluids and metal temperatures over the 23°C WLTP

The first WLTP of the complete ATCT was carried out using the 1D warm-up model. All temperatures were initialized at 23°C (coolant, engine oil, transmission oil and structure) and the WLTP cycle was completed under a 23°C ambient temperature condition. The results shown in Figures 3-4 display the fluid temperature profiles predicted from the numerical model, as well as a comparison to vehicle test data measured in a climatic wind-tunnel. The coolant temperature was measured at the 'engine out' coolant pipe comparable to that of test and was predicted to within 5°C over the entire WLTP cycle. The oil temperature was measured in the oil gallery, and CAE predicted temperatures were within 2°C of test data over the first 800 seconds, where friction is highly dependent on oil temperature. From 800 seconds onwards, oil temperature was overpredicted by ~ 8°C, but had little impact on friction due to temperature impact on oil viscosity diminishing at higher temperatures [17]. The transmission oil temperature followed a similar trend which can be found in Figure4, whereby the first 1000 seconds were predicted to within 3°C. This suggests the heat transfer coefficients to oil may have been over-predicted, but it was decided that further calibration was outside of the current scope of method development work. The spatial averaged metal temperatures are displayed in Figure 5, the end fluid and metal temperatures were transferred into the 3D heat retention model for initialization for the 9 hour soak period.

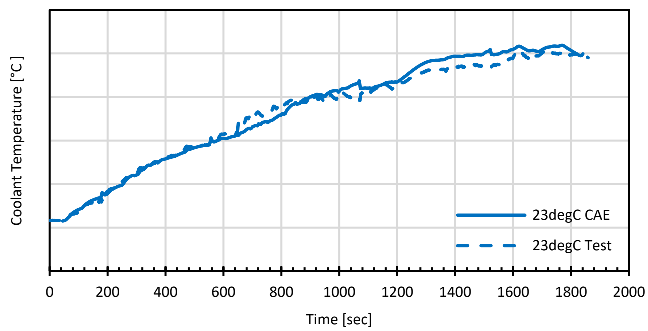

Figure 3. Coolant temperature over a 23°C ambient temperature WLTP cycle.

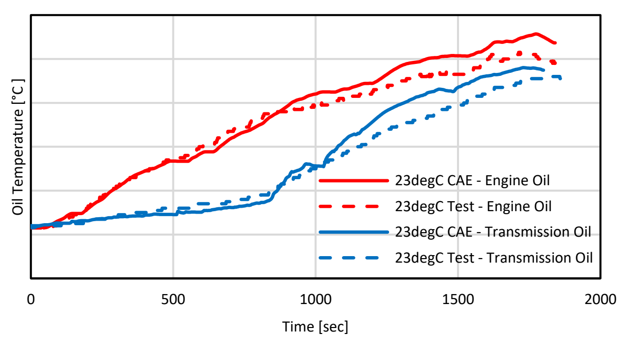

Figure 4. Oil temperatures (engine and transmission) over a 23°C ambient temperature WLTP cycle.

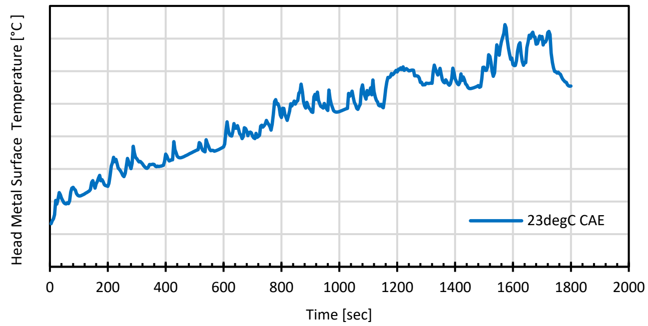

Figure 5. Engine head metal temperatures over a 23°C ambient WLTP cycle.

3D heat retention modelling – the buoyancy flow and engine fluids behaviour over the 14°C cooldown

The understanding of the flow development during early soak stage is vital to accurately predict the heat transfer coefficients for the heat retention modelling. Example of the resolved buoyancy-driven flow at the beginning of the soak is plotted in Figure 6. Three cross-sections of the engine bay are shown: one vertical transverse plane (X, Fig. 6. top row), two vertical longitudinal planes (Y1 and Y2, Fig. 6. mid and bottom rows). The superimposed velocity streamlines visualize the buoyancy flow around the engine bay components inside the underhood region. Heat from the turbocharger, exhaust manifold, engine (cylinder head, engine block, and engine oil sump), and the transmission unit (gearbox and transmission oil sump) generated after the first drive cycle, conducted to the surrounding air and induced the buoyancy-driven convection flow within the engine bay. The encapsulations at the top and side of the engine compartment and around the oil sumps provide insulations to the metal parts and the fluids within (coolant and oils). Increased convection coefficients were shown with and without the encapsulations respectively in between the external flow (air) and the engine solids by the simulations. Besides the temperature distribution, the CFD simulation helped visualizing the path of air leakages. For example, in Fig. 6 (right) considerable air flow likely occurred from the wheel arches and from the front cooling pack to the engine bay. These air flows are likely be reduced with specific encapsulation design such as the vehicle-mounted encapsulation concept shown in the previous work [13], where the engine parts are further insulated from the surrounding air and its convection heat transfer, thus leading to a slower cool-down process during the vehicle static soak.

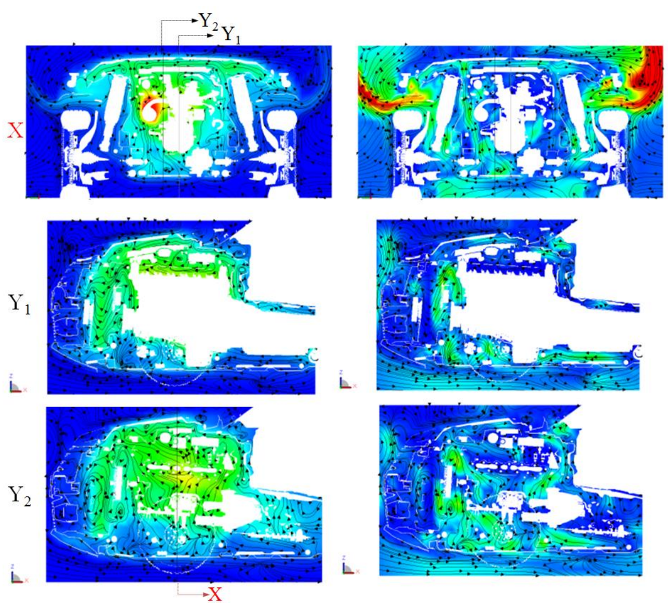

Figure 6. Images of the buoyancy flow within the under-hood region at 5 min of the soak process from a full-scale 3D CFD simulation using LBM method. Left - normalised flow temperature and right – velocity magnitudes (colour map range: 0 – 0.3 m/s), Superimposed is the velocity streamlines.

A 9 hour vehicle cool-down was simulated by the standalone vehicle thermal model and the key fluid (coolant, oil) temperatures were obtained and plotted in Figure 7. The absolute values of the temperatures (y scale) are removed in the figure as reasons as previously, to not reveal the confidential information on the specific vehicle. In general, cool-down trajectories of the coolant temperatures at cylinder head and engine block are correlated well in the simulation and the vehicle testing data. The mean deviation of the coolant temperature from the CAE compared with the test results is within 2% throughout the soak, with a maximum of 8% at early soak. For the oil temperature predictions, the modelling results show noticeable deviations for the engine and transmission oils during the first 0-4 hours cool-down, which decrease at the end of the soak period. The discrepancy of the oils' temperatures especially at the beginning of the soak is due to the fact that the model used single fluid-node to represent the oils bulk behaviour within the sumps, whiles the oil temperatures measured at the sumps plugs at the bottom were expected to have different temperature values from the bulk mean due to the buoyancy-driven temperature stratifications [14].This could be further investigated in future work. Here we concern the end temperatures of the oils and coolant fluids after the 9 hour soak. The coolant and oils end temperature differences were within 1°C and 3°C respectively. The end values from CAE were imported into GT to carry out the 14°C WLTC cycle simulation and evaluate the corresponding fuel consumption.

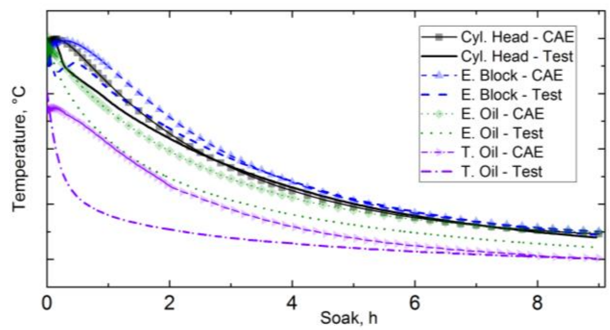

Figure 7. The cooldown curves of the engine coolants of the cylinder head and engine block, and of the oils in the engine sump and transmission oil sump from the CAE and the test of the 9 hours soak.

1D warm-up modelling – fluids temperatures and fuel consumptions over the cold-start 14°C WLTP

The temperatures at the end of the 9 hour soak phase were used as initialization for the second 14°C WLTP phase run using the 1D warm-up model. It is worth noting that the temperatures in the second WLTP were higher than the first WLTP even though the ambient temperature was colder at 14°C, due to the thermal encapsulation retaining heat over the 9 hour soak. The fluid temperatures can be seen in Figures 8-11 and were compared to the first 23°C WLTP as well as a standard 14°C WLTP where all temperatures were initialized at 14°C. A post soak vehicle test WLTP was also plotted for comparison. The warm-up time for the post-soak WLTP was in the order of 100 seconds faster than the 23°C WLTP. The metal temperatures were also initialized at temperatures warmer than 23°C which can be seen in Figure 9.

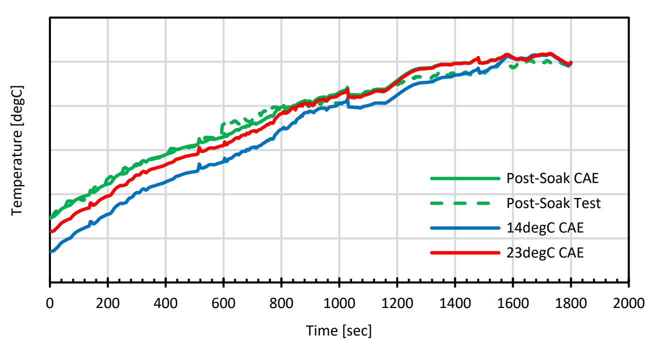

Figure 8. Coolant temperature over the 23°C and 14°C WLTP cycles as well as the post-soak WLTP cycle.

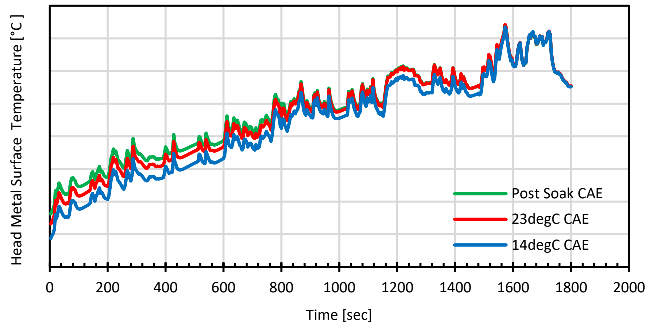

Figure 9. Engine head metal surface temperatures over a 14°C ambient post soak WLTP cycle.

Warm-up times of the oil (Figures 10-11) were also in the order of 80 seconds faster for the second WLTP, hence engine friction losses were less than the 23°C WLTP. The instantaneous piston friction can be seen in Figure 12 for both the 23°C WLTP and the post-soak WLTP. The post-soak WLTP clearly showed a lower level of friction generated by the piston (including rings and skirt). This in turn directly impacted the fuel consumption which can be seen in Figure 13.

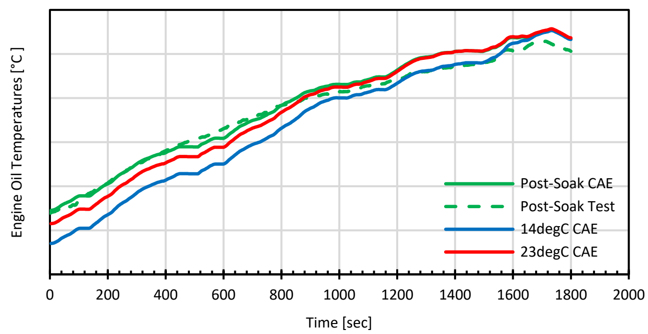

Figure 10. Engine oil temperatures over the 23°C and 14°C WLTP cycles as well as the post-soak WLTP cycle.

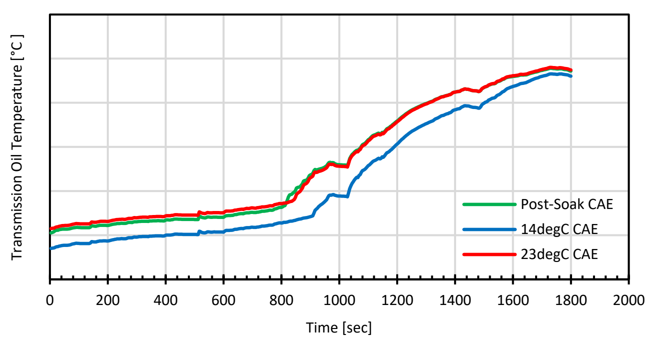

Figure 11. Transmission Oil temperatures over the 23°C and 14°C WLTP cycles as well as the post-soak WLTP cycle.

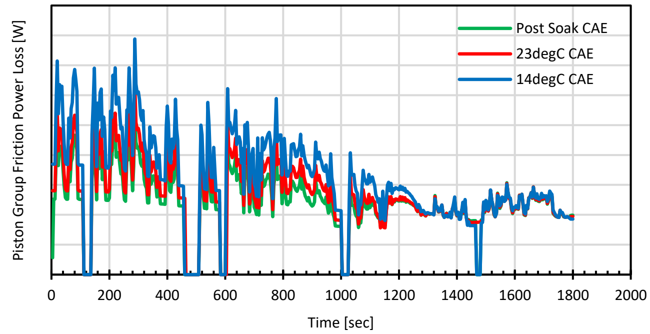

Figure 12. Piston group friction power loss over the 23°C and 14°C WLTP cycles as well as the post-soak WLTP cycle.

The cumulative fuel consumption trace of Figure 13 predicted at the end of the cycle the 23°C WLTP had the lowest fuel consumption, less than the post-soak WLTP. It is worth noting here that the post-soak WLTP had lower fuel consumption for the first 1000 seconds (displayed in the top-left zoomed image of Fig.13) during the warm-up period due to higher starting temperatures. During the temperature regulation phase of the WLTP cycle, the 23°C ambient WLTP consumed less fuel which may have stemmed from increased friction group temperatures due to the warmer ambient. Further work will be undertaken to understand further the impact of ambient temperature on local friction group temperatures and subsequent fuel consumption. Another important aspect to highlight is the lack of temperature dependent combustion characteristics captured in the current 1D warm-up model. More sophisticated and sensitive three-dimensional techniques are required to understand the impact of liner and head temperatures on combustion efficiency, as well as volumetric efficiency and variables such as spark timing.

The impact of heat retention and thermal encapsulation can also be derived from the work carried out. A fuel saving of 1% was seen when comparing the post-soak 14°C and the standard 14°C WLTP.

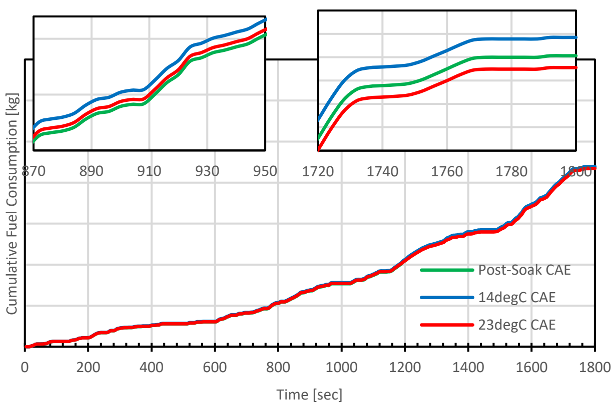

Figure 13. Cumulative fuel consumption over the 23°C and 14°C WLTP cycles as well as the post-soak WLTP cycle. Included are two zoomed images from mid-cycle (870 – 950sec) and end-cycle timing (1720 – 1800sec).

Summary

Driven by legislative and stringent fuel economy as well as real world driving emission (RDE), thermal encapsulation is becoming one of the key enabling technology. A holistic systems approach has to be taken towards thermal encapsulation and heat retention, with vehicle engine bay context being captured. This requires a CAE based design and verification method during the engineering process and for early failure mode detection. The CAE accuracy vs runtime needs to be considered. Transmission thermal modelling accuracy and its impact on the fuel economy assessment will be considered as next step. Engine oil modelling during key off soak is also an area that needs to be improved especially to understand the buoyancy driven engine oil stratification impact within a typical engine sump.

In this work, the ATCT CAE process was successfully modelled using a coupled 1D – 3D numerical approach. The 1D warm-up model was able to predict transient fluid temperature trends and was predicting to within 3°C of test for the warm-up phase of both 23°C and 14°C WLTP cycles.

3D heat retention modelling was applied on a detailed full-geometry model of a vehicle and engine bay encapsulation. The 9 hour static soak was simulated with the buoyancy flow characterized especially at the beginning of the soak. Possible air and heat leakages were identified around the engine bay. The coolant and oils cooldown trajectories were successfully predicted with good agreement to the test data. The discrepancy at the end of the soak was within 1°C for the coolant, transmission oil, and 3°C for the engine oil.

The impact of retaining heat within the powertrain was clearly seen, with a ~1% improvement in fuel consumption through retaining heat during the 9 hour soak period.

References

1. European Commission. REGULATION (EC) 2017/1151 of 1 June 2017 supplementing Regulation (EC) No 715/2007 of the European Parliament and of the Council on type-approval of motor vehicles with respect to emissions from light passenger and commercial vehicles (Euro 5 and Euro 6) and on access to vehicle repair and maintenance information, amending Directive 2007/46/EC of the European Parliament and of the Council, Commission Regulation (EC) No 692/2008 and Commission Regulation (EU) No 1230/2012 and repealing Commission Regulation (EC) No 692/2008. Official Journal of the European Union. L 175, 2017, pp. 1–643.
2. Roberts A., Brooks R., Shipway P., ‘Internal combustion engine cold-start efficiency: A review of the problem, causes and potential solutions.’ Energy Conversion and Management, ISSN: 0196-8904, 2014, Vol: 82, Page: 327-350
3. Fontaras G., Ciuffo B., Zacharof N., Tsiakmakis S., et al. ‘The difference between reported and real-world CO2 emissions: how much improvement can be expected by WLTP introduction?’ Transp. Res. Proc., 25 (2017), pp. 3933-3943,
4. Petley D., Jansen W., Wicksteed B., Caprioli D., et al., ‘Pre-development validation of an engine mounted encapsulation approach on SUV vehicle as ‘Eco-Innovation’ by means of EU Regulation 725/2011.’ In: Bargende M., Reuss HC., Wiedemann J. (eds) 14. Internationales Stuttgarter Symposium Proceedings. Springer Vieweg, Wiesbaden, 2014.
5. Green Car Congress, Oct. 2009. BMW outlines intelligent heat management applications for reducing fuel consumption and CO2. Place: https://www.greencarcongress.com/2009/10/bmw-outlines-intelligent-heat-management-applications-for-reducing-fuel-consumption-and-co2-newther.html
6. Powertrain Encapsulation for Low CO2 Emissions, in the press article ‘Ricardo shares insights on thermal optimization of electrified and conventional powertrains’, in Vehicle Thermal Management Systems conference (VTMS 14), London, 5-6 June 2019.
7. Shayler P J, Christian S J, Ma T A. Model for the Investigation of Temperature, Heat Flow and Friction Characteristics During Engine Warm-Up. SAE Transactions,Vol. 102, Section 3: (1993), pp. 1588-1597
8. Samhaber, C., Wimmer, A., and Loibner, E., "Modeling of Engine Warm-Up with Integration of Vehicle and Engine Cycle Simulation," SAE Technical Paper 2001-01-1697, 2001, https://doi.org/10.4271/2001-01-1697.
9. Chen K., Johnson J., Merati P., and Davis C., "Numerical investigation of buoyancy-driven flow in a simplified underhood with open enclosure," SAE Int. J. Passeng. Cars - Mech. Syst., vol. 6, no. 2, pp. 805–816, 2013.
10. Minovski B., Lofdahl L., and Gullberg P., "A 1D method for transient simulations of cooling systems with non-uniform temperature and flow boundaries extracted from a 3D CFD solution, " SAE Tech. Pap., 2015-01-0337, 2015.
11. Owen R., Price A., Boleto J. D. B., Sivasankaran S., and Jansen W., "Method development and application of thermal encapsulation to reduce fuel consumption of internal combustion powertrains," SAE Tech. Pap., 2019-01-0902, 2019.
12. Yuan R., Sivasankaran S., Dutta N., Jansen W., and Ebrahimi K., "Numerical investigation of buoyancy-driven heat transfer within engine bay environment during thermal soak", Appl Therm Eng., 146 (2020) 114525. DOI: 10.1016/j.applthermaleng.2019.114525
13. Yuan R., ‘Heat retention analysis with thermal encapsulation of powertrain under natural soak environment’, International Journal of Heat and Mass Transfer, 147 (2020) 118940. doi.org/10.1016/j.ijheatmasstransfer.2019.118940
14. Minovski B., Andrić J., Lofdahl L., and Gullberg P., "A numerical investigation of thermal engine encapsulation concept for a passenger vehicle and its effect on fuel consumption," Proc IMechE Part D: J Automobile Engineering, vol. 233, no. 3. pp. 557-571, 2019.
15. Bhatnagar P. L., Gross E. P., and Krook M., "A model for collision processes in gases. I. Small amplitude processes in charged and neutral one-component systems, " Phys. Rev. vol. 94. Pp. 511–525, 1954.
16. Exa Corp., PowerFLOW User's Guide 3.0 (Exa Corp., Lexington, MA, 1998).
17. Plumley, M., Wong, V., Molewyk, M., and Park, S., "Optimizing Base Oil Viscosity Temperature Dependence For Power Cylinder Friction Reduction," SAE Technical Paper 2014-01-1658, 2014, doi:10.4271/2014-01-1658

Acknowledgments

The authors would like to acknowledge the funding support from the Innovate UK and the Advanced Propulsion Centre (APC) for carrying out this work.

Definitions/Abbreviations

ATCT	ambient temperature correction test.
WLTC	world-wide harmonised light duty test procedure.
CAE	computer aided engineering.