

Sepp Steiner

# Development of Real-Time capable VTMS Model

Diploma Thesis

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## Abstract

The thesis concentrates on the development of a thermal model for use in real-time capable Vehicle Thermal Management System (VTMS) simulations. The thermal model is based on the AVL L.I.V.E. reference vehicle and developed using the software tool AVL BoostRT. First there is a general description of VTMS simulations, the AVL L.I.V.E. project and the state-of-the-art thermal model at AVL. After this description the thesis deals with the topics discretization, active mechanisms in thermal models and calibration of the thermal model. Finally the BoostRT model developed in scope of this work is compared to measurements and to results of the state-of-the-art approach. This comparison displays promising results, especially regarding calculation time.

## Kurzfassung

Diese Diplomarbeit beschäftigt sich mit der Entwicklung eines thermischen Modells zum Einsatz in echtzeitfähigen "Vehicle Thermal Management System" (VTMS) Simulationen. Das Demonstratorfahrzeug des AVL L.I.V.E. Projektes dient als Referenz für das mit dem Softwarepaket AVL BoostRT erstellte thermische Modell. Zu Beginn der Arbeit befinden sich allgemeine Beschreibungen zum Thema VTMS Simulationen, zum AVL L.I.V.E. Projekt und dem etablierten Ansatz zur Erstellung von thermischen Modellen im VTMS Einsatz. Danach folgen die Themen Diskretisierung, aktive Mechanismen in thermischen Modellen, Modellierung und Kalibrierung des thermischen Netzwerks. Zu Ende wurde das mit BoostRT entwickelte Modell mit der Messung im Fahrzeug wie auch mit den Simulationsergebnissen des etablierten Ansatzes verglichen. Das Resultat ist vielversprechend, besonders unter Betrachtung der Rechengeschwindigkeit.

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# 1 Vehicle Thermal Management System Simulations

## 1.1 Introduction

The discipline Vehicle Thermal Management System (VTMS) deals with all energy fluxes (mechanical, thermal and electrical) in a vehicle. In other words VTMS captures the energy fluxes in form of energy balances. At first it is a tool to evaluate and assess the usage of primary energy. On the other hand this assessment of the energy usage provides the engineer information on the possibilities and potential to optimize the system by means that change the thermal behavior of the vehicle and reduce losses and energy consumption. This measures might be variation of the layout, components and control strategy. VTMS simulations provide a tool to solve this task in a virtual environment.

## 1.2 Targets and Usecases

Figure 1.1 displays the interests in VTMS simulations as well as the relations in between. These interests are on the one hand side technical-oriented or based on legislative regulations, and on the other hand side there is the comfort and the safety of the passengers. In order to comply these interests, VTMS simulations should provide the engineer a tool to determine and balance the energy fluxes at the individual systems and components to reduce the losses and deliver the required energy at time.

The prime target of the VTMS simulation is to reduce the CO<sub>2</sub> emissions of a vehicle by measures that change the thermal behavior of the system. These measures can be separated in three categories:



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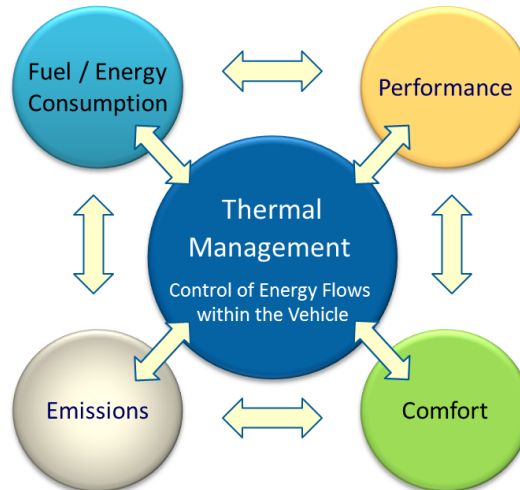


Figure 1.1: Balance of interests in VTMS simulations. (Source: AVL internal)

- Measures to ensure a faster warm-up of the engine and its auxiliaries. [3, chapter 1.2 p. 18,19]
- Measures to reduce the friction at the engine and the auxiliaries and therefore the power consumption at warm-up [3, chapter 1.2 p. 21,22] and warm engine conditions.
- Waste Heat Recovery, re-use at first run wasted heat and recuperation of losses.

Some examples, commonly used, are the introduction of the following measures.

**Reduction of coolant volume:** By reducing the coolant volume the overall thermal inertia gets reduced and as a consequence the temperature is rising faster towards the desired operational temperature.

**Split cooling:** In order to reduce the friction at engine cold conditions, split cooling is a concept to ensure faster warm-up of the engine block with in contrast still sufficient cooling of the cylinder head. For this reason the cooling system at the engine is split into at least two parallel paths. There is even the possibility to control these paths by valves as an additional stage of extension. [3, chapter 1.2 page 19]

**Electrical Coolant Pump:** An electrical coolant pump makes the coolant mass flow rate independent of the rotational speed of the engine. The coolant mass flow rate can be controlled on demand, e.g. the flow rate can get

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reduced at high speed but low torque driving conditions and increased at the opposite. Additionally the volumetric flow rate can be even reduced to zero to ensure a faster warm-up behavior but this measure also introduces the requirement of additional sensors to measure the system temperature at the engine since the coolant temperature at engine outlet is not available in this condition. [3, chapter 1.2 page 18]

**Map-controlled Thermostat:** A map-controlled thermostat can be used to increase the temperature at part load condition and therefore reduce friction. This type of thermostat features a electrical heating element in addition to the regular wax pellet to provide the ability of fast reaction to alternating load conditions.

**PWM controlled fans:** Similar to electrical coolant pumps on the coolant side, PWM controlled fans are used to provide the air mass flow on demand and therefore reduce power consumption.

This is no complete list of measures but it gives an overview on measures which may get used, to get the best operating temperatures, as fast as possible, in order to reduce friction. This reduction of friction in contrast to an engine with a regular cooling concept will also result in a lower fuel consumption and in lower emissions.

### 1.3 Overview on VTMS Models

In order to provide the required details to simulate and evaluate the thermal behavior, a VTMS model usually consists of several sub-models. These sub-models can be categorized by their functionality. VTMS simulations have to capture:

- the sources and sinks of energy fluxes,
- the path through the vehicle of these fluxes,
- optional elements to control these fluxes
- and finally the operational conditions of the vehicle in order to determine the fluxes.

This can be achieved by the introduction of an engine model (in this thesis a map based model), a model of the mechanical drivetrain, the electrical

## 1 Vehicle Thermal Management System Simulations

consumers and sources as well as the control strategy. Additionally the VTMS has to contain the model of the thermal inertia and thermal transports in order to include the paths of the thermal energy fluxes. These models are complemented by a virtual driver on a virtual road profile required to drive the virtual vehicle and therefore to determine the operational conditions of the vehicle.

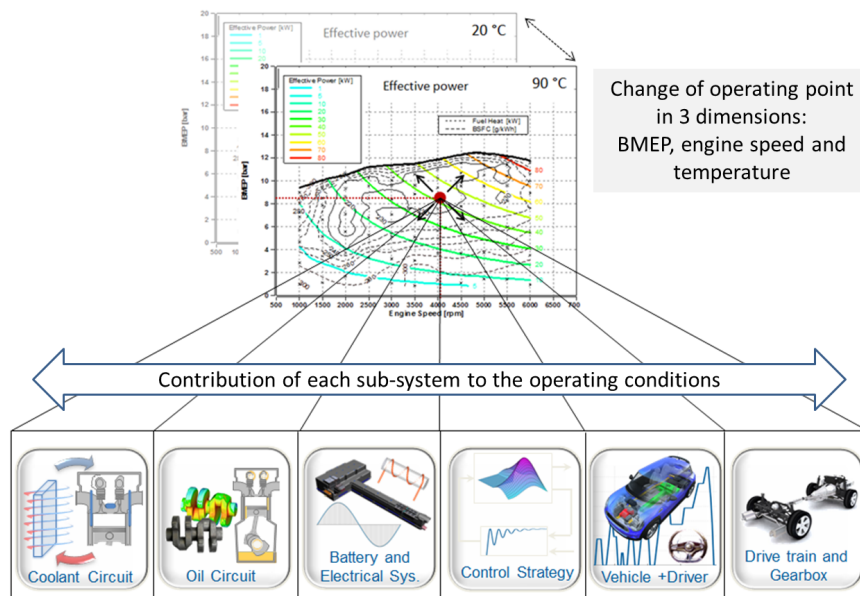


Figure 1.2: Sub-models in VTMS simulations. (Source: AVL internal)

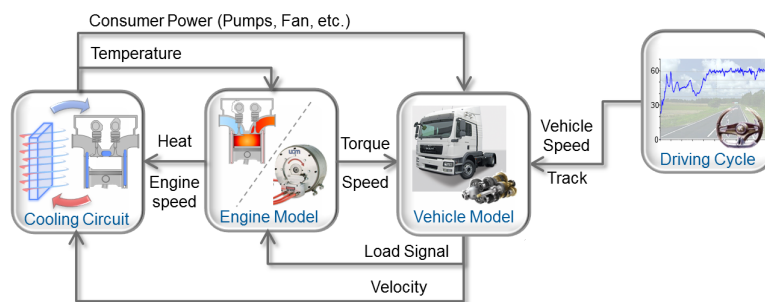


Figure 1.3: Signals and control in VTMS simulations. (Source: AVL internal)

Figure 1.2 gives an overview on the sub-models and their contribution to the operating point of the engine. At first these are calibrated separately and at a second step connected by the energy fluxes and control signals at their system boundaries. Figure 1.3 displays a base set of relevant signals at the subsystem boundaries, additional ones may be required in order to setup more sophisticated arrangements.

### 1.3.1 Engine Model

In a VTMS model the engine consists of at least two models. On the one hand side the model of the combustion process and on the other hand side a model of the thermal inertia of the engine, see section 1.3.3.

The combustion process in this thesis is modeled by using a map based approach. This approach is sufficient for thermal simulations because of the low dynamics of the thermal process. BMEP, rotational speed of the engine and a reference temperature determine the point of operation of the engine and therefore the fuel consumption, the process losses and all wall heat fluxes. If the direct evaluations of emissions are a target of the simulation a more detailed approach has to be used.

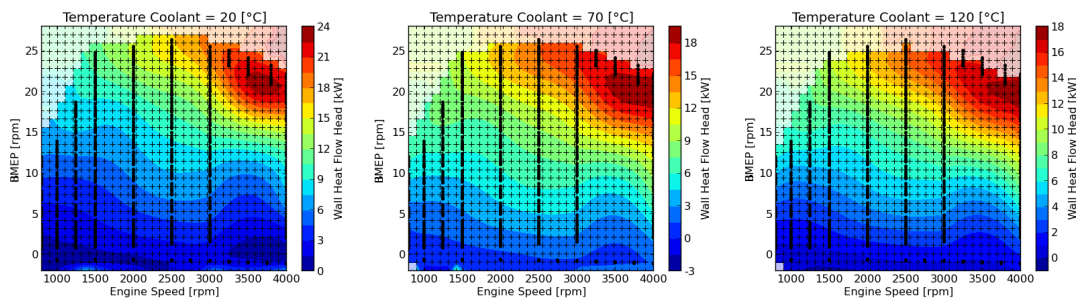


Figure 1.4: Map based engine model: exemplary wall heat flows at engine head for 3 different temperatures. (Source: AVL internal)

In case of this development the map based approach was chosen to setup the engine. Figure 1.4 displays exemplary excerpts of the map of wall heat flows at the engine head for three different temperature levels. All relevant heat fluxes at components and at the engine, heat sources for the thermal simulation, as well as the fuel mass flow for each operating point are modelled using this map based approach. Heat sources of the hybrid system are directly setup in Cruise<sup>1</sup> to provide the related heat fluxes.

### 1.3.2 Drivetrain Model - Virtual Vehicle

Additionally to the engine, the drivetrain with all its losses is required in the simulation to complete the vehicle at the mechanical side. In the AVL

<sup>1</sup><https://www.avl.com/cruise1>

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simulation environment, this model is setup by using the software tool Cruise. In addition Cruise acts as the master controller of the co-simulation, coupling all signals of the main model with its sub-models. Cruise also provides the driver, the ambient conditions and the road profile to the simulation.

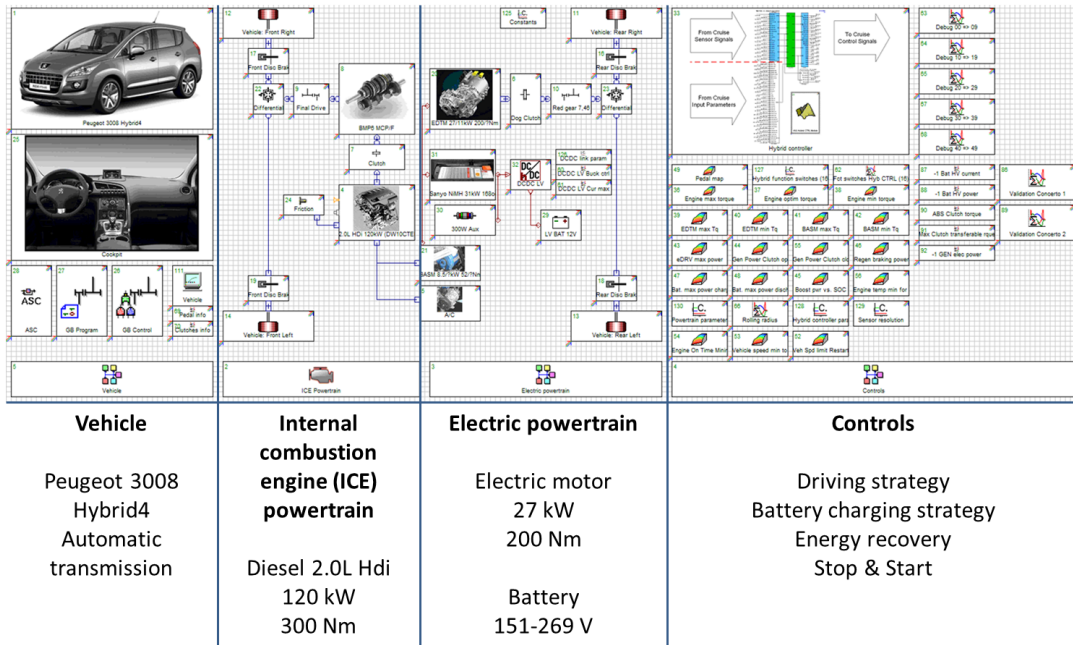


Figure 1.5: Cruise vehicle model. (Source: AVL internal)

### 1.3.3 Thermal Model

Although the vtms model is almost complete, the most imported part, and scope of the work, - the Thermal Model - is still missing. The thermal model contributes the thermal inertia of the system to the vtms model and as a consequence returns the actual temperatures needed to calculate the point of operation of the engine, the losses, the friction and fuel consumption.

Figure 1.6 displays the thermal model with its sub-circuits, the cooling, lubrication and air circuit as well as the thermal masses of the engine and all included auxiliary components.

# 1 Vehicle Thermal Management System Simulations

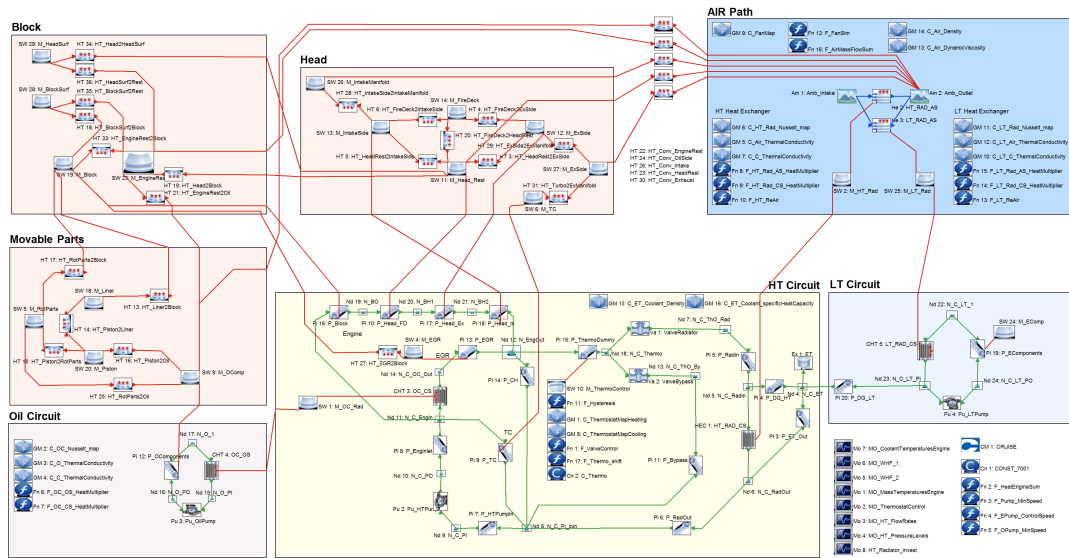


Figure 1.6: Thermal model of engine and cooling, lubrication and air circuit as created in the work using BoostRT.

## 1.3.4 Model of Control Strategy

In order to control all by the Engine Control Unit (ECU) adjustable parameters and components the control strategy is either modelled in Cruise or Matlab Simulink<sup>2</sup>. This Control Strategy Model may also get used as the baseline for the functional development of the ECU or directly for integration of submodels into the ECU. The optimization of this sub-model, under the assumption of all components are already fixed, is usually one of the most important but final parts of the VTMS simulation.

## 1.4 The State-of-the-Art Approach

The state-of-the-art approach at AVL uses the software tools Cruise<sup>3</sup>, Flowmaster<sup>4</sup> and Matlab Simulink<sup>5</sup>. Cruise is used to model and simulate the Driver, the driving conditions, the parts of the control strategy, the road profile, the drivetrain and it acts as the master controller of the coupled co-simulation.

<sup>2</sup><http://www.mathworks.de/>

<sup>3</sup><https://www.avl.com/cruise1>

<sup>4</sup><http://www.flowmaster.com/>

<sup>5</sup><http://www.mathworks.de/>

## 1 Vehicle Thermal Management System Simulations

The thermal model of the engine, the coolant and lubrication circuits and the air side of the thermal process is simulated by using Flowmaster. In addition the rest of the control strategy, especially the part taken directly from an ECU is usually modeled in Matlab Simulink. This approach is state-of-the-art and got optimized throughout the last years.

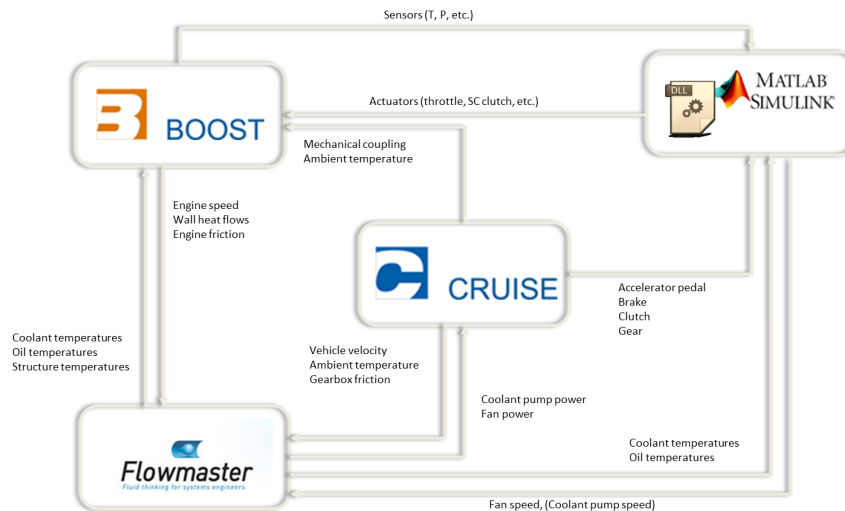


Figure 1.7: Overview on state-of-the-art approach. (Source: AVL internal)

Although this approach is reliable, it has one draw back, the coupling of all software products, especially by including Flowmaster into the simulation, to one big heterogeneous simulation. As a consequence external interfaces have to be used to get all involved software products to work together.

### 1.5 Real-Time Capability in VTMS

The main target of the real-time approach is to provide decreased calculation times and to merge the office topic simulation with the real-world application. In future this approach should also provide the ability to run the simulation “online” in a vehicle or include sub-models in the ECU in order to for example reduce the number of sensors and therefore the cost required to measure the operating conditions of the vehicle.

Another target for real-time applications are hardware-in-the-loop or software-in-the-loop tasks. In this case either real components, a whole engine or a real controller may run in the virtual real-time VTMS environment. This

environment may get used to develop and pre-calibrate a test item under virtual real-world conditions. Other applications are mentioned in the section “Benefits...” (section 5.3) at the end of this thesis.

### 1.5.1 What is real-time capability?

There are slight variations in the definition of the word real-time. In general real-time capable simulations and processes deliver the results at fixed deadlines. Therefore it is more a feature of the used software than a feature of the model or type of calculation. The used software has to respond to events at given deadlines. [4, Chapter III]

In the case of vtms simulations, the simulation has to be faster in any condition than real processes, in order to provide a response, the result, at the deadline. This constraint includes two aspects. If the simulation is coupled to a real process the software has to fulfill the defined deadlines regarding response times, but without the connection to a real world process, it can do its calculation as fast as possible and hopefully faster than the state-of-the-art approach in order to improve the calculation speed of offline non-real-time calculations.

Additionally the deadlines, in simulation context usually called time-steps, have to be defined in a manner to ensure that all important events of the simulated driving cycle get caught and the reaction of the simulation is the same as it would be in the real world application.

### 1.5.2 Cruise and BoostRT - A Way to get Real-Time Capability

In order to achieve the target of real-time capability, the software products AVL Cruise and AVL BoostRT are used in this approach. The Cruise model stays the same as in the state-of-the-art approach but the thermal calculations of Flowmaster are replaced by AVL BoostRT. BoostRT, short form of Boost Real-Time, is on the one hand side, at least technically, real-time capable [2] and on the other hand side BoostRT and Cruise share the same solver and for this reason the models will get united to a single monolithic model, without



## 1 Vehicle Thermal Management System Simulations

the need of external coupling. Therefore the overhead of the communication of all participants in the state-of-the-art approach co-simulation should become reduced to zero. As a consequence faster calculation times are expected.

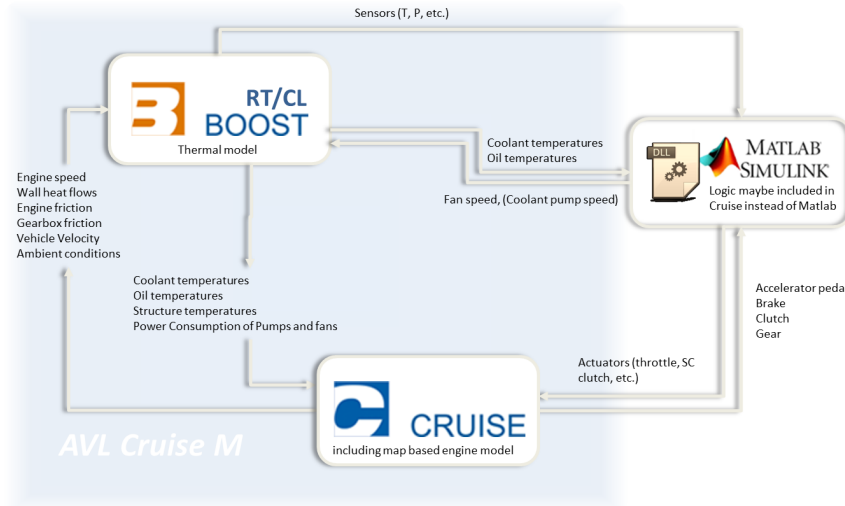


Figure 1.8: Overview on real-time approach. (Source: AVL internal)

### 1.6 Scope of Work

The target of this work is the condensation of a state-of-the-art VTMS model into a real-time capable one on the base of a reference vehicle (section 1.6.1). Therefore all sub-models created and used by the software tool Flowmaster are replaced by BoostRT. The sophisticated thermal mass model is simplified as much as possible model, described in section 2. In addition to this thermal mass model, the cooling, lubrication and air circuits are rebuilt by means of BoostRT. Since the feature-set of BoostRT regarding cooling and lubrication is quite basic, many features, standard in Flowmaster, for example the fully featured thermostat, had to be rewritten from scratch. Chapter 3 illustrates the calibration process and finally chapter 4 provides an overview on the results of the simulation.

### 1.6.1 Reference for Simulation - AVL L.I.V.E.

The AVL L.I.V.E., short form for “Live In Vehicle Energy balancing”, is the real world application reference for this work. AVL L.I.V.E. is a research and development project with the target to improve the thermal benchmarking of vehicles as well as to improve the processes to setup, run and post-process vtms simulations. The reference vehicle at the AVL L.I.V.E. project is a Peugeot 3008 Hybrid4 with the key data:

|                           |  |
|---------------------------|--|
| Manufacturer              | Peugeot  |
| Type                      | 3008 Hybrid4   |
| Engine type               | R4 Diesel 1997 cm <sup>3</sup><br>120 kW @ 3750 rpm, 300 Nm @ 1750 rpm |
| Electrical engine         | Synchronous Electric Engine<br>27 kW and 200 Nm                        |
| Emission legislation      | EU5  |
| CO <sub>2</sub> -emission | 104 g/km ( 4.0 l/100km )<br>4WD capable full hybrid                    |

Efficiency increase by measuring, analysis and adjustment of vehicle energy fluxes




| <b><u>EFFICIENT BENCHMARKING &amp; MODELING</u></b>  | <b><u>DEMONSTRATOR ENERGY BALANCE VEHICLE</u></b>   | <b><u>L.I.V.E. (Live In Vehicle Energybalancing) OPTIMIZATION</u></b>  |
|--|---|--|
| <p>Methodology to depict the improvement potential in existing vehicles using CAE simulation tools:</p> <ul style="list-style-type: none"> <li>▪ Fast, efficient and cost-effective vehicle benchmarking</li> <li>▪ Reverse-Engineering in view of the vehicle energy management system</li> <li>▪ Specific detail models of existing vehicle systems</li> </ul> | <p>Demonstration of the ability to depict the vehicle structure with acceptable accuracy using the simulation:</p> <ul style="list-style-type: none"> <li>▪ Modularity &amp; Variability of Systems / Functions</li> <li>▪ Visualization of consequences resulting from various operating strategies</li> <li>▪ Automatic calibration of vehicle models using L.I.V.E. (Live In Vehicle Energybalancing)</li> </ul> | <p>Optimization of all systems in vehicle by applying defined energy management measures:</p> <ul style="list-style-type: none"> <li>▪ Optimum energy management measures using CAE supported optimization</li> <li>▪ Prediction of the improvement potential using AVL VTMS simulation approach</li> <li>▪ Validated improvement potential using state of the art-hardware</li> </ul> |
|   |    |   |

Figure 1.9: AVL L.I.V.E. - Targets of Energy Balance Research platform. (Source: AVL internal)

## 1 Vehicle Thermal Management System Simulations

The main targets of the AVL L.I.V.E. project are displayed in figure 1.9. In general these targets can be summarized:

- Reduction of sensor count and therefore cost to reverse engineer the thermal balance of a vehicle.
- Research on improvements in the setup of vtms models.
- Automation of vtms model setup, based on live in vehicle measured input data.
- Improvement of vtms simulation accuracy and predictability.
- Research on optimization potentials of vehicle thermal control strategies.

### Reference Circuits

There are two cooling circuits in the reference vehicle, one for the regular combustion engine and its auxiliaries, the high temperature (HT) circuit and the other for its hybrid components, the low temperature (LT) circuit.

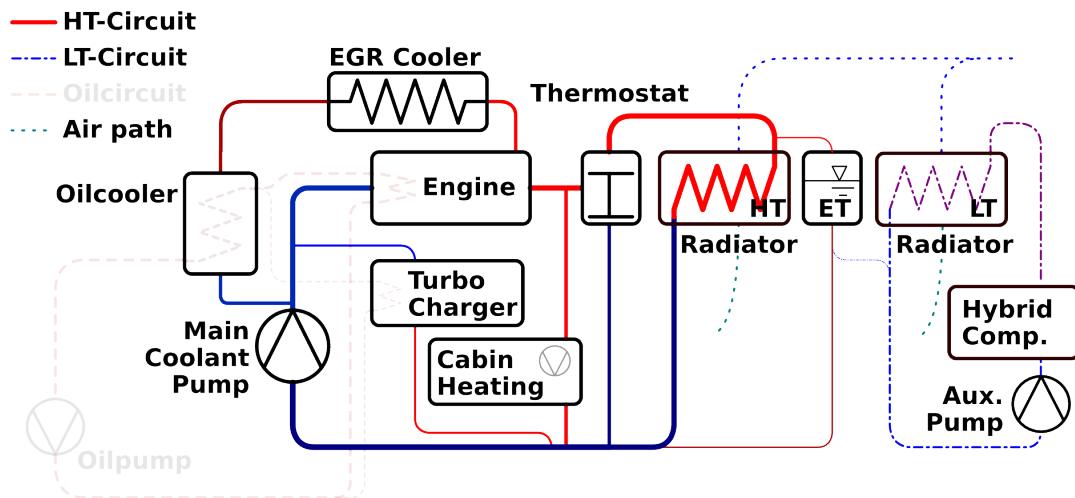


Figure 1.10: Cooling circuits of reference vehicle. (Source: based on AVL internal)

The HT circuit is cooled by the medium Water/Glycol 50/50 and consists of a mechanically driven coolant pump feeding the coolant to the engine, block and head in serial. The oilcooler in series with the EGR-cooler is located in parallel to the engine. After the junction of oilcooler/EGR-cooler path with the engine path the coolant is split into the turbo charger, the cabin heater path and towards the thermostat. The turbo charger path as well as the cabin heater path lead directly to the suction side of the coolant pump.

## 1 Vehicle Thermal Management System Simulations

The thermostat controls the coolant flow rate towards the bypass path and the radiator path, both joining simplified at the suction side of the coolant pump. In parallel to the radiator there is an expansion tank to provide a volume for the expanding coolant while heating up. Additionally the expansion tank defines the pressure level of the system by pressurizing it (air spring).

All components of the LT path are located in series with the LT radiator and the electrically driven LT coolant pump.

Finally, in order to complement the fluid circuits of the thermal model, there is the lubrication circuit (figure 1.11) and an air path to transport the heat to the ambience. The lubrication system can be simplified as a serial arrangement of oilpump, oilcooler and solid structure part of the lubrication circuit.

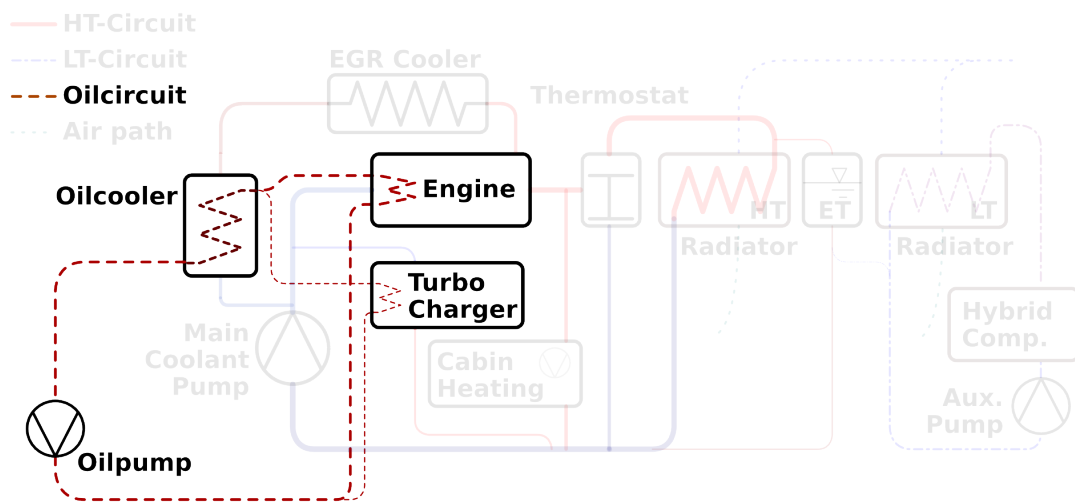


Figure 1.11: Lubrication circuit of reference vehicle. (Source: based AVL internal)

The air path consists of the ambient condition before the grill, the grill, a fan and both, LT- and HT radiator in parallel. After these components the incoming cooling air will have to find its way through the engine compartment.

### Reference Model used in state-of-the-art Approach

There is a thermal model developed in parallel to this work by the means of the software tool Flowmaster. Flowmaster, a software package to conduct 1D fluid and thermal simulations is used in this state-of-the-art approach. It features a fairly complex setup of all the thermal relevant components

## 1 Vehicle Thermal Management System Simulations

as displayed in figure 1.12. A detailed view on the mechanisms, active in the thermal model, is provided in chapter 2.4 of this work but in general it connects all thermal relevant circuits, (cooling, lubrication and air) and all heat effective masses by heat fluxes to the overall thermal model.

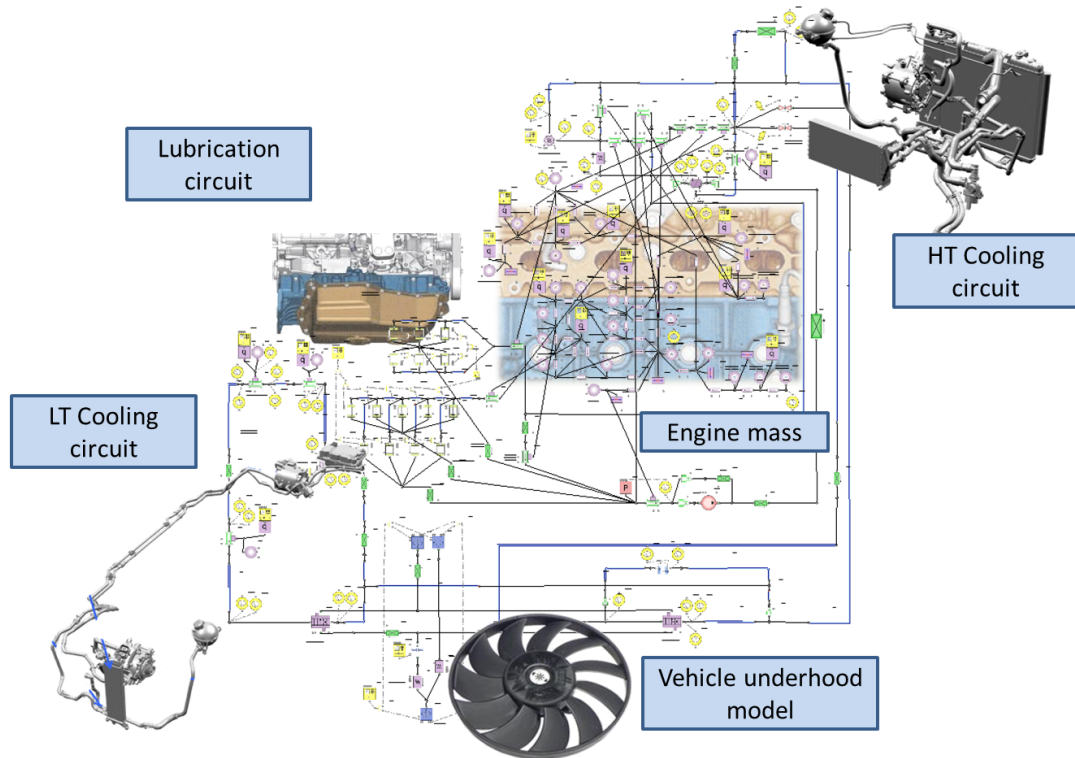


Figure 1.12: Flowmaster reference implementation of the thermal model, using the state-of-the-art approach. (Source: AVL internal)

The Flowmaster model features:

- The main high temperature cooling circuit, with all elements of its piping.
- The low temperature cooling circuit with all its components modeled separately.
- A detailed lubrication circuit.
- The air path with all its relevant components modeled separately.
- A sophisticated mass model of the engine and all thermal relevant components based on a benchmark engine.

In comparison to the Flowmaster model (figure 1.12) the BoostRT model (figure 1.6) is clearly simplified. Especially the lubrication circuit and all piping is reduced to minimum number of components.

## 2 Modeling of the VTMS - Thermal Circuit

This chapter deals on the topics discretization from point of view of measurement and simulation and the resulting consequences on modeling of the thermal simulation in general, the later sections (2.4) should point out the chosen assumptions and rules to simplify the model.

### 2.1 Laws Valid for Measurement and Simulation

The following section should contain the most important mathematical background to review results, measurements and to describe the used models. The reference for all equations is the VDI Wärmeatlas [5]. The most important equations in the thermal model of the VTMS is the first law of thermodynamics (equation 2.1) and the law of conservation of mass (equation 2.7). Both have to apply on all nodes and all thermal relevant components in the thermal model and will get used to determine the discretization of both the measurement and the simulation.

#### First Law of Thermodynamics

$$\frac{dE_{sys}}{dt} = \sum_i \dot{Q}_i + \sum_j \dot{W}_{tj} + \sum_k \dot{m}_k \cdot \left( h_k + g \cdot z_k + \frac{1}{2} \cdot v_k^2 \right) \quad (2.1)$$

$E_{sys}$  ... the inner energy of the system  $J$

$t$  ... time in  $s$

$\dot{Q}$  ... heat flow at system boundaries in  $W$

$\dot{W}$  ... work flow at system boundaries in  $W$

## 2 Modeling of the VTMS - Thermal Circuit

- $\dot{m}$  ... mass flow at system boundaries in  $\frac{kg}{s}$
- $h$  ... specific enthalpy of fluid at system boundaries in  $\frac{J}{kg}$
- $z$  ... geodetical height of location at which mass flow crosses system boundaries in  $m$
- $g$  ... gravity in  $\frac{m}{s^2}$
- $v$  ... velocity of fluid at system boundaries in  $\frac{m}{s}$
- $i, j, k$  ... iterators to capture all mass flows and energies at the system boundary

Either the setup and the first steps in the calibration process rely on steady state conditions:

$$\frac{dE_{sys}}{dt} = 0 \dots \text{in case of steady state results} \quad (2.2)$$

In contrast to the kinetic energy, the potential energy is negligible in the cooling system of a vehicle, due to low differences in geodetical height.

$$\sum_k \dot{m}_k \cdot g \cdot z_k \cong 0 \quad (2.3)$$

Furthermore if a single component or section is evaluated and the inlet and the outlet piping have the same diameter and a negligible difference in density the sum of the kinetic energy at the heat exchanger boundaries becomes 0. In a general case this simplification is not valid.

$$\sum_k \dot{m}_k \cdot \frac{1}{2} \cdot v_k^2 = 0 \quad (2.4)$$

Any component except fans and pumps exchange no technical work at their system boundaries.

$$W_j = 0, \text{ for all } j \text{ and any } t \longrightarrow \sum_j \dot{W}_{t_j} = 0 \quad (2.5)$$

## 2 Modeling of the VTMS - Thermal Circuit

The First Law of Thermodynamics is now reduced to:

$$\sum_i \dot{Q}_i + \sum_k (\dot{m}_k \cdot h_k) = 0 \dots \text{for heat exchangers} \quad (2.6a)$$

$$\sum_i \dot{Q}_i + \sum_j \dot{W}_{t_j} + \sum_k (\dot{m}_k \cdot h_k) = 0 \dots \text{for pumps and fans} \quad (2.6b)$$

### Law of Conservation of Mass

Additionally to the First Law of Thermodynamics all components and nodes part of a fluid circuit have to respect the Law of Conservation of Mass.

$$\sum_k \dot{m}_k = 0 \quad (2.7)$$

### Pressure Losses and Heat Transfer

The models used to determine heat transfer coefficients and pressure losses are noted in their respective sections in the chapter 2.4 “Details on Modeling”.

## 2.2 Discretization in Simulation and Measurement

In general every attempt to look at a process, involves some kind of discretization. Therefore both the measurement and the simulation can only catch an image of the real process as good as their boundaries are. Usually both have different origins, targets and level of detail.

Figure 2.1 displays an overview on the correlations between measurement and simulation. Generally speaking, in simulations the local discretization is driven by the underlying mathematical correlations, in measurements on the other side the driving forces are usually the costs, the convenience and accessibility of the location. As a consequence in-vehicle measurements, like done in the AVL L.I.V.E. vehicle, are limited in order to maintain a fully functional vehicle but provide an image, as good as possible, of all heat fluxes



## 2 Modeling of the VTMS - Thermal Circuit

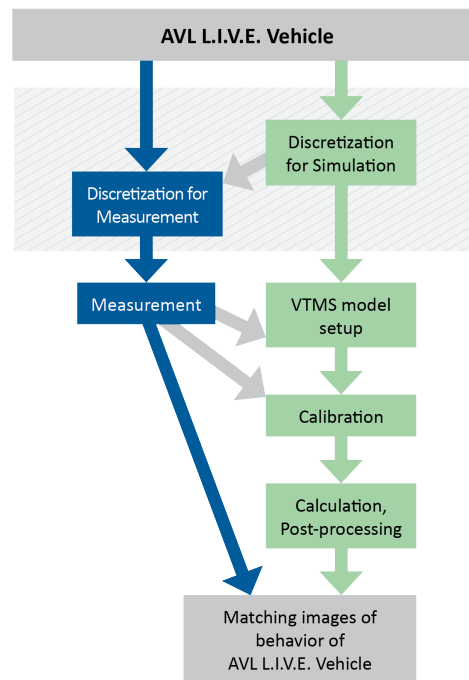


Figure 2.1: Schematic image of workflow of discretization, simulation and calibration

to the coolant and lubrication circuit in the most cost effective way and with as few points of measurement as possible.

### 2.2.1 Measurement

The target of measurements is to provide an image of a real process and the discretization is chosen to either provide data to control or verify the process itself, to get a better understanding of the process or even, like in this case, to provide input data for simulation tasks.

In order to fill all parameters in the equations listed in section 2.1 to characterize a section of the circuit, usually a component, the following data is determined through measurements.

**The pressure loss of the section:** needed to setup the pressure loss characteristic of the section depended on the measured volumetric flow rate.

**The volumetric flow rate at the section:** in order to calculate the mass flow rate in conjunction with the density of the fluid.

**The temperature difference at the section:** required finally to calculate the enthalpy difference, in other words the heat input at this section, by using the thermal capacity of the fluid, its temperatures at inlet and outlet and its current mass flow rate.

**The type of fluid or material:** in order to get access to its properties, like density, thermal capacity and thermal conductivity.

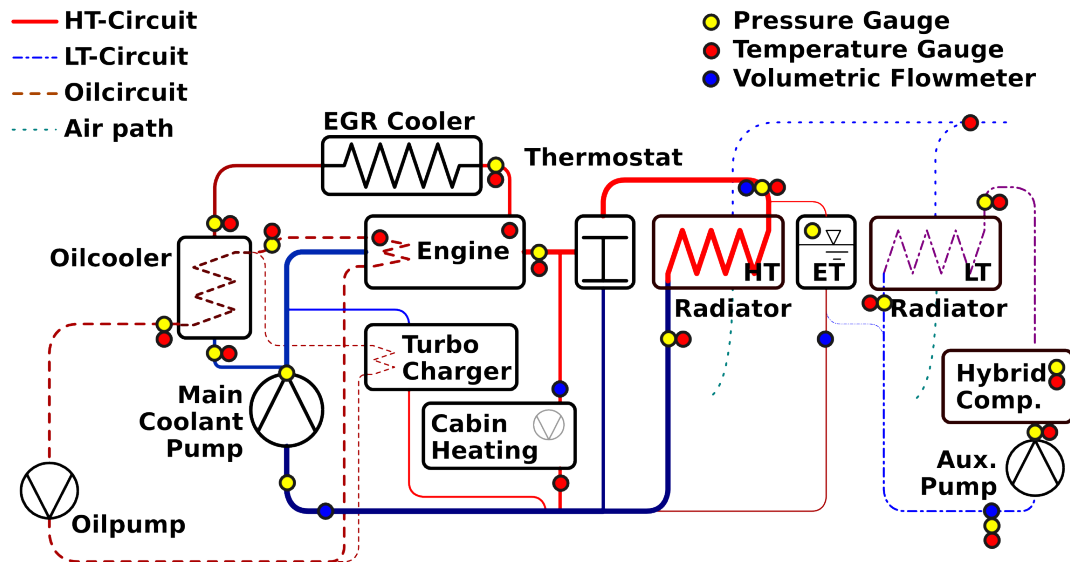


Figure 2.2: Excerpt of the measurement plan of the AVL L.I.V.E. project. (source: AVL internal)

In order to get reliable input data under exclusion of the influence of time dependencies, most of the measurements should be done at steady-state conditions or at least near those. The resulting input data is used for a baseline setup and furthermore extended and calibrated by the use of additional transient measurements, taken in a controlled environment.

In case of this work, most of the measurements were already done, before this work started. Therefore the discretization of the measurement was already done before. An excerpt of the measurement plan is displayed on figure 2.2. It is clearly visible that the before stated measuring parameters are used.

In order to provide data to setup and validate vtms models the measurement plan contains the following cycle runs and conditions:

**Steady-state and constant load measurements** to provide input data to setup the submodels of the vtms simulations. In contrast to warm-up cycle

## 2 Modeling of the VTMS - Thermal Circuit

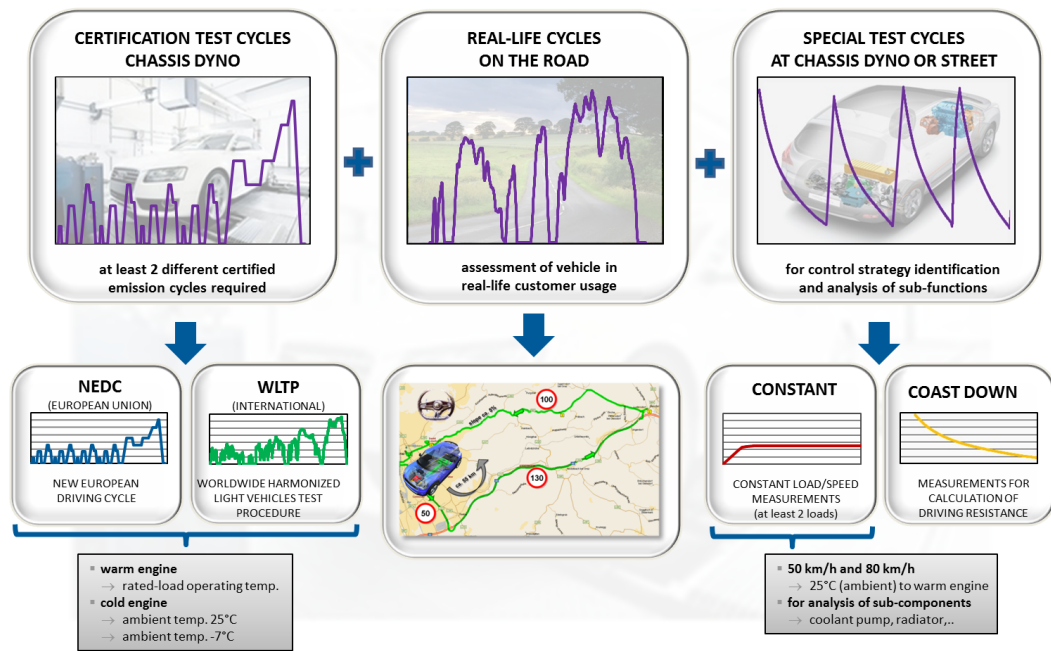


Figure 2.3: Overview on cycle runs in measurement plan. (source: AVL internal)

measurements, these measurements are done at various loads and vehicle velocities, to provide insights into the pressure loss characteristics of all components.

**Warmup cycles** used to calibrate the thermal behaviour of the system. Warmup cycles are done at constant speed and constant load, with conditioned cold initial operating conditions and end at a fully heated up engine. In other words the measurements end if all temperature measurements including the temperature of the solid masses, especially the engine, reach a constant steady state condition.

**NEDC measurement runs** to provide comparable benchmark data.

**Real world driving cycles** to provide data for evaluation purposes of the models, especially at transient conditions.

**Rigtest measurements** used to capture data characterizing single components.

**Coast Down measurements** in conjunction with all ECU readable parameters used to calibrate the Cruise model especially the friction and losses of the drivetrain.

### Limitations and Resulting Consequences of Measurement

Usually all kind of measurement is limited to its sampling rate and the resolution of the measurement equipment. Additionally there are limitations and therefore consequences for the post processing due to the local discretization of the measurement. In other words, especially if in-vehicle-measurements are done, the points of measurement are usually not at the component in- and outlet but at some more easily accessible locations or even at junctions, and therefore they not only include the component itself but also most of its piping.

Additionally there are some reasons of failure in measurements. One reason might be measuring range of the equipment. This usually applies to flow rate measurements. If the measured flow rate is at the lower range of the sensor, under the assumption the sensor is a turbine, temperature dependent friction may cause corruption of the measurement result. Even the insertion of a turbine increases the pressure drop at the section as well as the overall system pressure drop and therefore may cause a shift of the flow distribution. For this reason, the flow measurements in conjunction with pressure measurements to determine the pressure loss characteristics of a component are often done off vehicle at the rig-testbed at specific coolant temperatures. In the vehicle only the pressure loss and coolant temperature, with exactly the same sensors and positioning as used on testbed, are measured and the volumetric flow rate is calculated at post-processing from the components pressure loss characteristics.

Another parameter prone to failure is the pressure. The measured pressure is usually the static pressure without knowledge of the velocity related dynamic part of the total pressure. This lack of knowledge can be circumvented by equaling the dynamic part of the total pressure at inlet and outlet by choosing the measuring point in way, that the pressure is measured at points with the same velocity level. This is done by choosing locations at inlet and outlet with the same cross-sectional area. For this reason any difference at inlet and outlet has to be well documented in order to make no mistakes on the review of the results.

Finally even the measurement of temperatures, or better the post-processing of temperatures, may cause serious errors. For example, if the accuracy of the

## 2 Modeling of the VTMS - Thermal Circuit

temperature sensors and measurement equipment is  $\pm 0.1$  K and a temperature difference, not unusual, of 2 to 3 K is evaluated, the error is between 5 to  $\sim 3$  %. Additionally the temperature is usually dependent on the location of the sensors in a pipe and therefore the position of the sensors may introduce additional deviations. If those measured temperatures are used to calculate e.g. a heat input to a fluid and the fluid, like the usually used coolants in vehicles, have a high thermal capacity, a slight error in the temperature measurement may cause a big error on the calculated heat result.

Due to the fact that not every parameter of an environment can be easily determined in measurements, for example the velocity of the air at the air-cooled components, these parameters rely on benchmark based assumptions or at least some physical plausible relations, in this case an assumed pressure loss and the velocity of the vehicle as velocity of the cooling air. Some other parameters, like the temperature and therefore heat distribution between solids, for example in the engine, are not always accessible without destroying or at least modifying the component to the point of no return to the original state.

### 2.2.2 Simulation

In the simulation model all the physical components are replaced by simplified mathematical correlations. These constructs may be setup on input based on the physical component specifications or as in the case of this work on the measured input data. The discretization has to be chosen based on this input and the target of the simulation. The target of the simulation model in this work is on the one hand side to achieve, as fast and as constant as possible, calculation speeds and on the other hand side at least the control strategy of the whole energy fluxes in the model should remain variable and editable.

**fast and constant calculation speeds:** To maintain this target, the number of elements in the circuits should be kept as low as possible. In other words each path of the circuits should contain either one component to define heat rejection and pressure loss or if there are more than one heat effective components in a path, the maximum number of elements in the model should not exceed the number of heat effective components in the path.

**fully adjustable control strategy:** All components that control the energy fluxes and mass flow rates have to be kept fully functional in the model.

### 2.3 Workflow and Model Condensation for Real-Time Approach

The workflow to setup the vtms model contains two steps. First the setup of the model itself and as the second step the calibration of the model. The details on modeling are mentioned in chapter 2.4 and the calibration in chapter 3.

The following rules were applied to the simulation model to create the real-time capable vtms model:

- The thermal inertia, fluids and solids, have to match the physical setup.
- The system should be recreated in the model with as few components as possible.
- All elements that control the energy fluxes should remain fully functional.
- As many locations of measurement as possible should be kept in the simulation model.
- The discretization regarding time, in other words the timestep of the simulation should be chosen to be as big as possible to gain the calculation speed but still delivering good results.

## 2.4 Details on Modeling

This section has the target to demonstrate the ideas behind the modeling of all components and signals in the thermal model. It does not deal from the components point of view but should provide insights into the active mechanisms in the thermal model of a VTMS simulation and a possible solution, how to model these mechanisms by using BoostRT.

Although the simulation type itself is transient, all input data, as well as all equations to generate this input data from the measurements or datasheets of the components, are based on steady-state conditions. Therefore all derivatives of time are negligible at least regarding model setup and the first steady-state runs of calibration process.

### 2.4.1 Components used in BoostRT

BoostRT offers the following components, at least those got used, to setup the thermal model. Each component is listed with a short description of its function as well as the title of its related mechanisms described in the following subsections.

**Solid:** Element to store heat. *Active mechanisms: thermal mass (section 2.4.2)*

**Heat Transfer:** Conductive heat transfer between solids. *Active mechanisms: thermal mass (section 2.4.2)*

**Pipe:** Element to form fluid circuit, with characteristic pressure loss and able to transfer heat from or to a connected solid. *Active mechanisms: Mass transport (section 2.4.4), heat transfer (section 2.4.3) and thermal mass (section 2.4.2)*

**Heat Exchanger:** Same as pipe but used to model the thermal interaction between two fluids. *Active mechanisms: Mass transport (section 2.4.4), heat transfer (section 2.4.3) and thermal mass (section 2.4.2)*

**Pump:** Source to raise pressure and move fluids. *Active mechanisms: Sources and sinks (section 2.4.5)*

**Ambient:** Defines the conditions at air path inlet as well as outlet. *Active mechanisms: Air Path (section 2.4.10)*

**Expansion Tank:** Defines the pressure level in the fluid circuits. *Active mechanisms: Expansion Tank (section 2.4.7)*

**Map:** General element to store maps used to setup user defined correlations, for example, fluid properties, used in thermostat and fan.

**Function:** General element to provide user defined control functions, used for example in thermostat and fan.

**Cruise Interface:** Interface to the software Cruise. *Active mechanisms: Signals (section 2.4.8)*

### 2.4.2 Thermal Mass

There are two kinds of thermal masses considered in the model, on the one hand side the solid mass of the components and on the other hand the mass of the fluids in the cooling and lubrication circuits. These masses directly affect the warm-up behavior of the vehicle.

Each mass has the ability to store heat content, the enthalpy ( $E_{th_{stored}}$  in J in reference to 0K) [5, chapter A 1.1]:

$$E_{th_{stored}} = c \cdot m \cdot T \quad (2.8)$$

$c$  ... the thermal capacity in  $\frac{J}{kg \cdot K}$

$m$  ... the mass of the fluid (gas or liquid) or solid in kg

$T$  ... the temperature of the fluid or solid in K

and the rate of heat flow ( $\dot{Q}$  in W) between connected solid masses is defined with the equation [5, chapter A 2.2.1 eq. 19a & b]:

$$\dot{Q} = k \cdot A \cdot \Delta T \quad (2.9)$$

$k$  ... the heat transfer coefficient in  $\frac{W}{m^2 \cdot K}$

$A$  ... the heat effective cross-section area in  $m^2$

$\Delta T$  ... the temperature difference between two solid masses K



## 2 Modeling of the VTMS - Thermal Circuit

The heat transfer coefficient ( $k$ ) between solids is determined by:

$$k = \frac{\lambda}{s} \quad (2.10)$$

$\lambda$  ... the thermal conductivity in  $\frac{W}{mK}$   
 $s$  ... the distance between two solid masses  $m$

Additionally BoostRT provides the equations of Gnielinski [5, chapter Ga] to calculate the average heat transfer coefficient (instead of the  $k$  for conductive heat transfer,  $\alpha_m$  for convective heat transfer) between solid mass and a fluid at a circular pipe. At fluid to fluid heat exchangers, like the oilcooler and the radiators of the HT and LT circuit, a benchmark based approach is used. The topic heat exchangers and heat transfer to fluid will be captured separately in the subsection 2.4.3 “Heat Transfer”.

### Summary of Modeling in BoostRT

The solid masses of the components, especially of the engine, are based on geometrical information and material properties. All data is derived from a similar engine. The initial filling of all fluids at ambient temperature can be used to determine the mass of the fluids and is used to model the inner volume of all components containing fluid.

Affected components:

**Solid masses** defined by type, material properties, and mass.

**Heat transfers** defined by heat effective area and distance to each connected solid mass.

**Pipes and heat exchangers:** only their fluid volume is part of the thermal mass, all other functions and parameters will follow in the next sections.

## 2 Modeling of the VTMS - Thermal Circuit

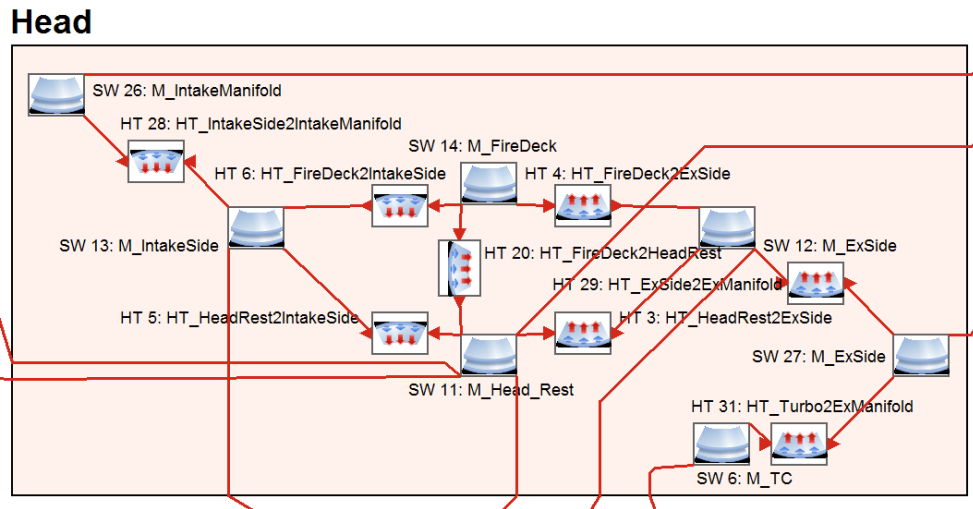


Figure 2.4: Excerpt of engine mass model in BoostRT.

### Requirements for Setup and Calibration

- The masses have to be the same, type and quantity, in the model as well as in physical vehicle.
- The overall sum of the masses must be kept constant while calibration, especially if two masses are simplified to a single one.
- All relevant heat transfers, based on conduction and convection must be captured in the model, solid to solid, solid to fluid and fluid to fluid.

### 2.4.3 Heat Transfer

There are two types of heat transfer in the thermal model, conduction of heat (see section 2.4.2), convective heat transfer between solid and fluid as well as the transfer between two fluids at a heat exchanger. The later two rely on the same base of equations to simulate the transfer [5, chapter Ga].

$$\dot{Q} = \alpha \cdot A \cdot \Delta T_{ln} \quad (2.11)$$

$\dot{Q}$  ... transferred heat in  $W$

$\alpha$  ... the heat transfer coefficient in  $\frac{W}{m^2 K}$

$A$  ... the heat effective area in  $m^2$

$\Delta T_{ln}$  ... the logarithmic temperature difference  $K$

## 2 Modeling of the VTMS - Thermal Circuit

$$\alpha = \frac{Nu \cdot \lambda_{(T_m)}}{d} \quad (2.12)$$

- $\alpha$  ... the heat transfer coefficient in  $\frac{W}{m^2 K}$   
 $Nu$  ... the dimensionless Nusselt number  
 $\lambda_{(T_m)}$  ... the thermal conductivity in  $\frac{W}{m K}$   
 dependent on the average temperature of the fluid  
 $d$  ... diameter of the pipe in  $m$

$$\Delta T_{ln} = \frac{(T_w - T_i) - (T_w - T_o)}{\ln \frac{T_w - T_i}{T_w - T_o}} \quad (2.13)$$

- $\Delta T_{ln}$  ... the logarithmic temperature difference in  $K$   
 $T_w$  ... temperature of the wall of the component in  $K$   
 $T_i$  ... fluid temperature at inlet of the component in  $K$   
 $T_o$  ... fluid temperature at outlet of the component in  $K$

The difference between the heat transfer between solid and fluid and the heat transfer between two solids in the BoostRT thermal model setup in this work is only the calculation of the Nusselt number.

At heat exchangers, heat transfer between two fluids, a three-dimensional Reynolds vs Reynolds vs Nusselt correlation like shown on figure 2.5 was used. This characteristic is the same as used in the state-of-the-art approach and due to lack of complete performance data, all heat exchanger characteristics used in the thermal model are based on internal benchmark data of a similar engine taken from a similar vehicle.

On the other hand the heat transfer between solid and fluid, at the pipes, is modeled by using the equations of Gnielinski [5, chapter Ga], already built into BoostRT.

## 2 Modeling of the VTMS - Thermal Circuit

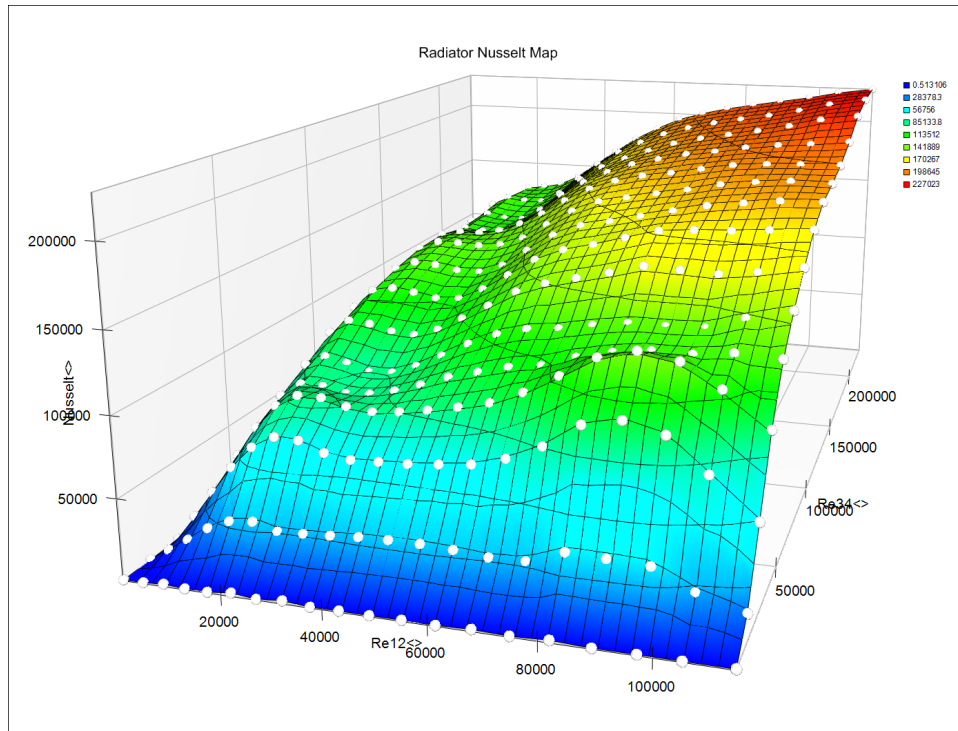


Figure 2.5: Exemplary heat transfer characteristic and implementation in BoostRT of the HT Radiator: Re vs Re vs Nu

### Summary of Modeling in BoostRT

The heat transfers are based on benchmark data of similar engines, with the exception of all heat transfers in pipes. This approach is used due to the fact that there is no complete heat transfer performance characteristics available for the actual components. This benchmark data got scaled towards the actual measured performance by using either, the BoostRT provided heat transfer multiplier, to adjust the calculated heat transfer coefficient and the heat effective area. Additionally several parameters, like the actual fluid properties, not user accessible by BoostRT at runtime have to be modeled from scratch, in order to use the characteristics in the same manner as in Flowmaster.

### Requirements for Calibration

- The Reynolds vs Reynolds vs Nusselt characteristics used to setup heat exchangers are dimensionless and based on benchmark data but still require either scaling and recalculation to meet the dimensions of the

actual components or the components have to be setup the same as the benchmark component.

- The geometrical input data, required to setup the heat exchangers must either fit to reality if the actual performance characteristics is available or it has to match the values of the benchmark component.

### 2.4.4 Mass Transport - The Fluid Circuits - Pressure Losses

In addition to the heat transfer between masses, energy can also be transferred, or better transported by the movement of masses, in other words by the heat content of the mass flow. The mass transport, locally limited by pipes forms the fluid circuits of the thermal model. Any component, part of a fluid circuit, has a specific pressure loss at a given volumetric flow rate, its pressure loss characteristic.

The pressure loss characteristic of a section or component can be determined by measuring the static pressure, at inlet and outlet at several volumetric flow rates and at fixed temperatures. Usually the location of measurement at inlet and outlet has the same pipe diameter and same velocity level and the difference in geodetical height is negligible. Therefore the difference in static pressure is the same as the total pressure difference, usually directly accessible in software packages.

In the state-of-the-art approach, there are usually three ways used to specify the pressure loss characteristic, either by geometrical dimensions and surface roughness, usually in pipes and bends, by defining a pressure loss vs volumetric flow rate table or by calculating a dimensionless pressure loss coefficient ( $\zeta$ ) using the Law of Bernoulli.

The Law of Bernoulli for a single component with constant diameter, including in and outlet conditions [5, chapter L]:

$$p_1 + \rho_1 \cdot g \cdot z_1 + \rho_1 \cdot \frac{v_1^2}{2} = p_2 + \rho_2 \cdot g \cdot z_2 + \rho_2 \cdot \frac{v_2^2}{2} + \rho_{12} \cdot \frac{v_{12}^2}{2} \cdot a \cdot \zeta \quad (2.14)$$

$p$  ... the static pressure in  $Pa$

$\rho$  ... the density of the fluid  $\frac{kg}{m^3}$

## 2 Modeling of the VTMS - Thermal Circuit

- $g$  ... the gravity constant in  $\frac{m}{s^2}$
- $z$  ... the geothetical height in  $m$
- $v$  ... the velocity of the fluid in  $\frac{m}{s}$
- $a$  ... bodyfactor see equation 2.15
- $\zeta$  ... the dimensionless loss coefficient
- 1 ... the index 1 indicates the condition at the inlet
- 2 ... the index 2 indicates the condition at the outlet
- 12 ... the index 12 indicates that the parameter is based on the average between condition 1 and 2

Simplified by the assumption of at least negligible velocity level differences and geodetical heights between inlet and outlet.

$$\Delta p_{12} = \rho_{12} \frac{v_{12}^2}{2} \cdot a \cdot \zeta \quad (2.15a)$$

$$a = 1 \dots \text{for instruments, components and adapters} \quad (2.15b)$$

$$a = \frac{1}{d} \dots \text{for circular pipes} \quad (2.15c)$$

$$\zeta = f(Re) \quad (2.15d)$$

$$Re = \frac{\rho \cdot v \cdot d}{\eta} = \frac{v \cdot d}{\nu} \quad (2.15e)$$

- $\eta$  ... dynamic viscosity in  $\frac{kg}{s \cdot m}$
- $\nu$  ... kinematic viscosity in  $\frac{m^2}{s}$
- $d$  ... the inner diameter of the pipe in  $m$

The loss coefficient  $\zeta$  for circular pipes can be determined by drawing the conditions in iterations in the Moody diagram [5, chapter Lb, figure 1], or by software using the corresponding equations [5, chapter Lb].

Due to the assumed discretization, to use only as few components in the circuit as possible and due to the target of the simulation, to concentrate on energy balance and not on piping optimization, there is at least one component, one pressure loss, for each path of the circuits required to setup the hydraulic model.

## 2 Modeling of the VTMS - Thermal Circuit

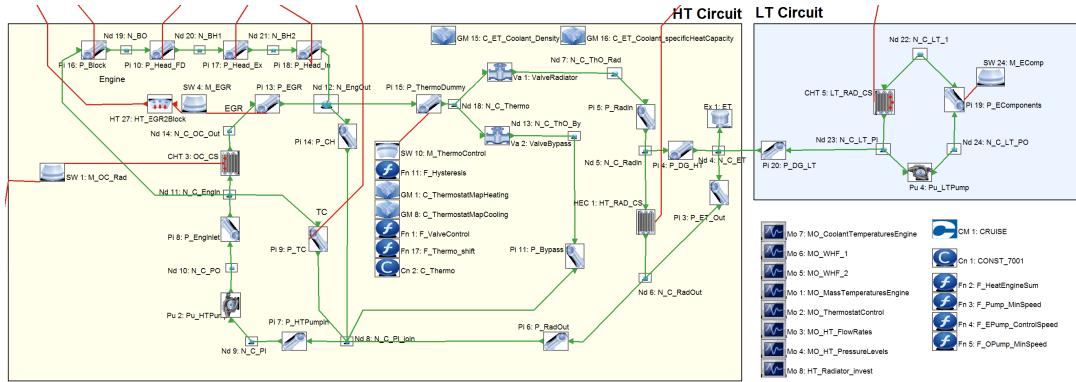


Figure 2.6: The coolant circuits used in BoostRT.

In case of this work all components including their piping are replaced by a single pressure loss element, the pipe. This pipe represents the sum of all pressure losses in the path. In BoostRT pipes are capable to transfer heat at their walls. If a path contains more than one heat source, pipe got split into same number of parts as heat sources in it. The pipe is setup the same as the corresponding elements in the real circuit, regarding volume and diameter. Since the hydraulic diameter of the real components, as well as the local velocity level, is unknown, the diameter of the corresponding piping was taken for setup. This is a rough estimation of the inner diameter and results in a specific virtual length of the pipe as well as in a properly not matching velocity level in the components, but both can be compensated by multipliers during calibration.

Additionally all fluid to fluid heat exchangers got setup not by pipes but by dedicated heat exchangers with basically the same hydraulic model with the addition of entry and exit loss coefficients.

### Requirements for Setup and Calibration

- Piping diameter and volume have to match those of the real circuits.
- The number of pipes used in the model is one for each path or heat source.

### 2.4.5 Sources and Sinks - Coolant Pumps and Fans

There is the requirement of sources and sinks, in order to move the fluid in the circuits. These sources and sinks are represented by the pumps and fans, the pressure sink at the suction side and the pressure source at the outlet side. This pressure difference drives the fluid in the circuit. Again as for all components the first law of thermodynamics and the law of conservation of mass has to be fulfilled.

In general the pumps are modeled in the same way as in Flowmaster. A pressure head vs flow rate vs rotational speed map provides the characteristic of the pump. Additionally the efficiency can be setup in the data sheet of the components.

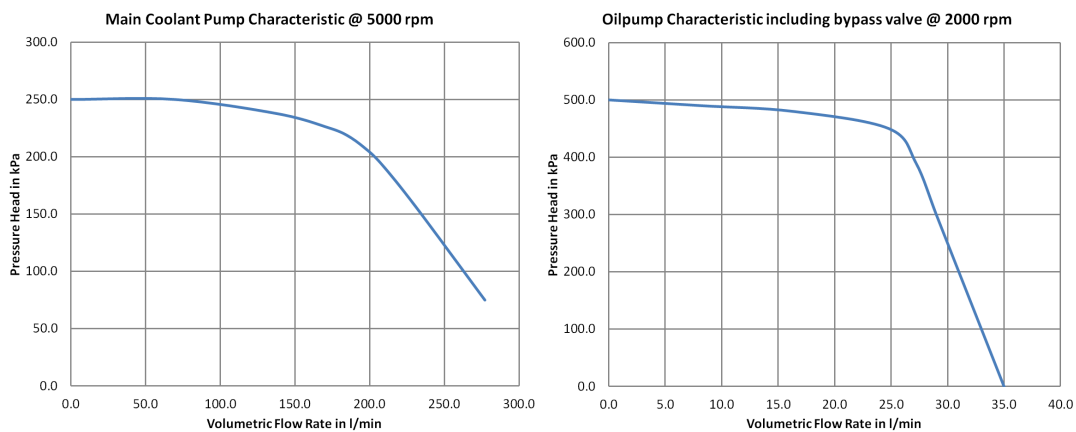


Figure 2.7: Exemplary pump characteristics.

$$P_{pump} = \frac{\Delta p \cdot \dot{V}}{\eta} \quad (2.16a)$$

$$\eta = \eta_{mech} \cdot \eta_i \quad (2.16b)$$

$P_{pump}$  ... overall power consumed by the pump in  $W$

$\Delta p$  ... pressure head of the pump in  $Pa$  or in SI units  $kg/(m \cdot s^2)$

$\dot{V}$  ... volumetric flow rate in  $m^3/s$

$\eta$  ... overall efficiency of the pump

$\eta_m$  ... mechanical efficiency of the pump

$\eta_i$  ... inner efficiency of the pump



## 2 Modeling of the VTMS - Thermal Circuit

In order to capture all loads in the vehicle model, the consumed power of the pumps has to be sent to Cruise, if their power consumption is not included in the engine friction maps. Either by sending the power calculated by equation 2.16 or by a direct mechanical coupling.

Although there are such components part of BoostRT, the fans in the air path are modeled by the use of functions from scratch, because there seems to be some issues, at least in the used development snapshot of the solver, regarding the connecting element, the plenum, in the air path. Either because of wrong or incomplete input data or software issues the plenum did not behave as expected. Therefore the same approach as used in the coolant and lubrication circuits was used to model a fan. A pressure head vs volumetric flow rate vs rotational speed characteristic is used in a map to change the inlet pressure of the air path and therefore the mass flow rate of the air. This simulates the same behavior like the fans act in Flowmaster.

### Requirements for Setup and Calibration

- The characteristics of the pumps have to meet the measurements.
- The operational points of the pumps can be taken from measurement although in some cases the shape of the characteristic has to be taken from benchmark, since the full shape of the characteristics is not always, as in case of this work, available through measurement.
- The efficiency of the mechanically driven pumps is at least in this work not needed, since their power consumption is already included in the engine friction maps. An update of the power consumption has only to be considered if these pumps get exchanged by different ones, for example electrical pumps.

### 2.4.6 Thermostat - Control of Energy Distribution

In general a thermostat is a component that ensures a desired system temperature by sensing a system parameter, usually a temperature, and controlling the mass flow and, as a consequence of the layout, the energy flow in the system. In vehicles the simplest and state-of-the-art implementation (figure

## 2 Modeling of the VTMS - Thermal Circuit

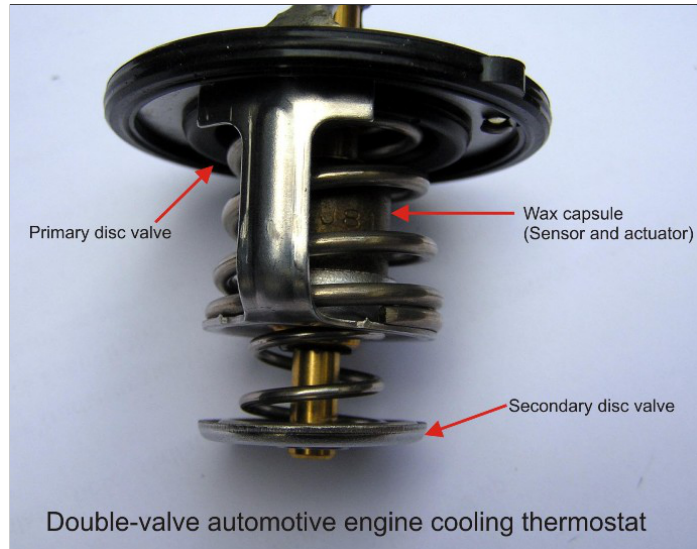


Figure 2.8: Thermostat of automotive cooling circuit (Source: Dougsim 2010).

2.8<sup>1</sup>) consists of a sealed wax pellet, which is directly connected to two spring-loaded plates. A volumetric expansion of the wax pellet will cause a positive translation of one plate and a negative one of the other. These plates and the housing of the thermostat form two valves controlled by the expansion of the wax according to the temperature of the pellet and the state of aggregation of the wax inside.

### Characteristic of Thermostat

The thermostat is characterized by its pressure loss vs opening vs volumetric flowrate characteristic or the loss coefficient vs opening characteristic for both paths and by its thermal behavior, the opening ratio vs temperature characteristic.

There are two opening ratio vs temperature characteristics, one for heating up and one for cooling down. Those two form the characteristic hysteresis. Additionally there is a time delay, in Flowmaster modeled as time constant, caused by the thermal inertia of the wax pellet.

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<sup>1</sup>Source: Dougsim 2010 [http://en.wikipedia.org/wiki/File:Double\\_valve\\_automotive\\_thermostat.jpg](http://en.wikipedia.org/wiki/File:Double_valve_automotive_thermostat.jpg) under licence CC-SA 3.0

## 2 Modeling of the VTMS - Thermal Circuit

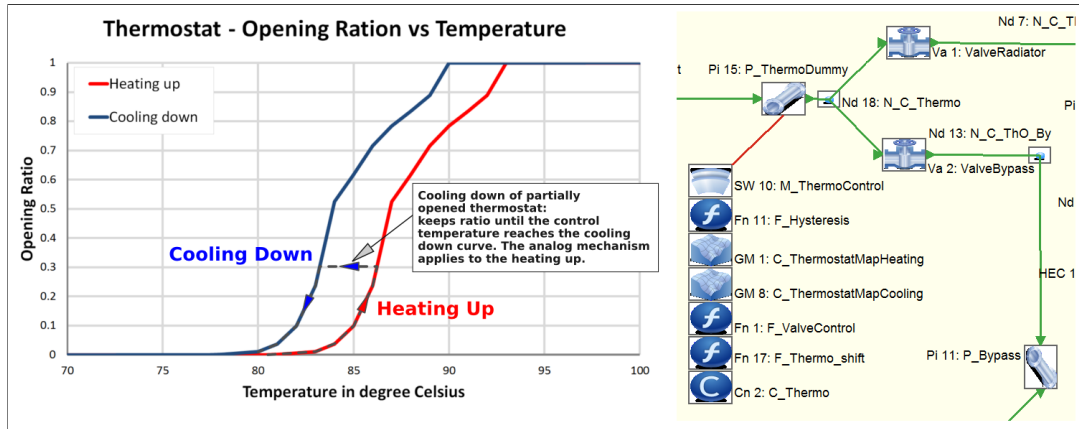


Figure 2.9: Thermostat opening characteristic in the BoostRT model.

### Summary of Modeling in BoostRT

In the simulation model the thermostat is a sub-model by itself, consisting of two valves, a dummy pipe and a solid mass connected to the dummy pipe. The temperature of the mass is used to control the thermostat position by looking at the heating-up and cooling-down curves, considering the actual behavior of the system. In other words, if the system is heating up, the thermostat is opening, and the control temperature is not rising any more, it stays at the actual opening ratio until either the temperature is rising again and the operation point is on the heating-up curve or the temperature is low enough, so the operation point of the thermostat at the actual opening ratio can be found at the cooling-down curve (figure 2.9). The time constant of the thermostat is modelled by the dummy pipe, the solid mass and the heat transfer between the fluid and the solid mass at the dummy pipe.

Since there is no datasheet of the thermostat available, the input data for it, especially the opening behavior, is based on internal benchmark data.

### Requirement for Setup and Calibration

- The Start-to-Open-Temperature (SOT): The coolant temperature at which the thermostat starts to open. The SOT is taken from measurements.
- The resulting steady state coolant temperature of the system.
- Hysteresis and opening vs temperature curves: These characteristic curves have not been available in this project and have to be taken from

internal benchmark data. Furthermore this curves can only be calibrated if the heat exchanger to ambient is proven to be fully calibrated and the overall flow distribution at rated speed, at thermostat fully opened and fully closed condition, meets the reference.

- The pressure loss vs opening ratio characteristic, assumed in the beginning and has to be calibrated to meet the measured flow distribution.
- The data to setup the solid and the heat transfer to simulate the time constant of the wax pellet.

### 2.4.7 Expansion Tank

The expansion tank provides room for the expanding fluids and defines the resulting pressure level of the system. In BoostRT there is just a single arm expansion tank available based on its geometrical dimensions and the initial liquid level. More complex setups have to be modeled at this stage of development by using pipes.

#### Requirement for Setup and Calibration

- The geometrical dimensions of the tank.
- The overall fluid volume must match the fluid volume in the physical vehicle. The model will be prone to failure, if fluid volume and expansion tank dimensions do not match and it either becomes full or empty while the simulation.

### 2.4.8 Signals at the Boundary of the Thermal Model

As described in the last sections the thermal model rely on the following signals in order to work as expected and delivers the following outputs. These connections have to be made in BoostRT and Cruise to unite these two software packages. Figure 1.3 displays these signals graphically.

Incoming Signals:

**Rotational speed of the engine** used to drive all components mechanically connected to the engine. In case of this work, the oil and coolant pump. In other words, the rotational speed is the variable boundary condition of the sources and sinks.

**Heat fluxes** caused by the internal combustion process at the engine, the heat input to the thermal network.

**Additional heat fluxes** caused by friction, directly connected to the source of heat.

**Signals to control auxiliaries** like the rotational speed of the fan.

**The ambient temperature and velocity of the vehicle** to setup the inbound conditions of the air to cool the system.

The thermal model returns the following results and signals to the vehicle especially to the engine maps in Cruise:

**The coolant, oil and engine mass temperature** used to update the input temperature of the map based engine and therefore as result, the heat fluxes, fuel consumption and the control strategy will get updated.

**The power consumed by mechanical and electrical consumers** to update and adjust the load of the engine. In case of the actual model, the mechanically connected pumps, the oil pump and the main coolant pump are already included in the engine maps. The fan and the electrical coolant pump, same as the whole hybrid system, is fully considered in Cruise and only the signal of activation is exchanged at the model boundaries.

### 2.4.9 Material Properties used in the Thermal Model

The coolant circuits are modeled using a volumetric mixture of Water/Glycol 50%/50%, the oil circuit is based on a 5W30 engine oil and the air path is modeled on air as real compressible gas. Additionally alloy, steel and cast iron was used to setup the solid masses. The wax pellet of the thermostat is modeled based on oil since there is no data of chemical composition of the wax pellet available. All material properties are taken from the internal database of BoostRT . This database is based on the properties provided by the "VDI Wärmeatlas" (chapter D, 7. edition 1994).

### **2.4.10 The Air Path of the Cooling System - Boundary to Ambience**

Finally the air path, transporting heat by mass transport across the system boundaries, consists in reality of several restrictions, the grill, the heat exchangers and engine compartment as well as a fan. The mass flow is driven by the ram pressure at the vehicle front. Additionally the fan, if not overblown, is used to actively increase the mass flow whenever required for the cooling requirements at the radiator.

In order to use as few components as possible but still keep the relevant functionality intact, this system is reduced to two heat exchangers, one for HT circuits and one part of the LT circuit. Both sides of the heat exchangers are connected to ambient and at the inlet the air mass flow is controlled by fan condition, velocity and resulting pressure loss. This path is mostly coded from scratch by using functions and maps. The basic model is basically the same as for pipes and pumps but the fluid properties are based on air as real compressible gas.

### 3 Calibration of the Thermal Model of the Real-Time Capable VTMS Model

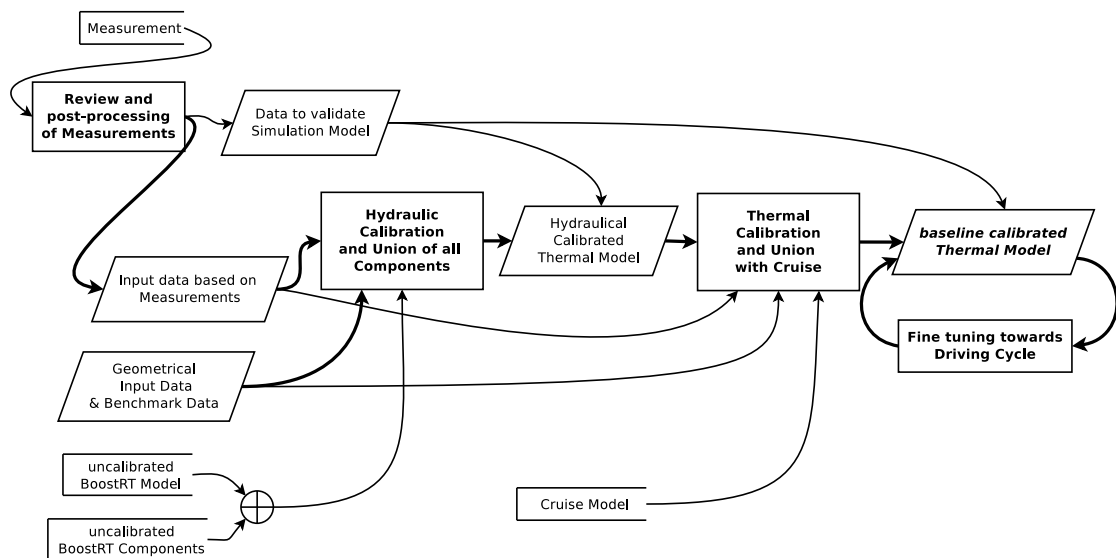


Figure 3.1: Overview on calibration process.

The next sections should demonstrate the workflow of the iterative process of calibration. The models of all components will get calibrated and connected to the fully featured thermal model of the VTMS simulation. The steps to this model are:

1. the hydraulic calibration of all components of the fluid circuits and the circuits itself (section 3.2),
2. the calibration of the thermal transports (section 3.3),
3. the fine tuning towards a driving cycle (section 3.4).

**Note:** All calibration is based on the limited in-vehicle measurements and possibly can be accelerated by more detailed rigtest measurements.

## 3.1 Input Data - Sources and Review

Before the start of calibration all available input data, especially the data provided by measurements, has been reviewed and checked for plausibility. This was done either by comparing the measurements with benchmark data, searching for anomalies and artifacts and comparing the measurements, as far as comparable to each other. This reviewed and approved input data is the base for calibration.

Additionally, since a target of this work is the condensation of a Flowmaster model, this model also became source of input data at that stage in the development process, when the Flowmaster model reached AVLs internal approval and replaced some assumptions and benchmark data.

As a conclusion all input data can be sorted by its validity (the most reliable input data at first).

1. Directly measurable parameters, for example easily accessible geometrical information and volumes.
2. Input data provided by reviewed measurements, for examples volumetric flow rates, pressure levels and temperatures.
3. Input data based on benchmarks, for example the thermal network of the engine.
4. Assumed input data.
5. All software provided multipliers, for example friction multipliers.

This enumeration of sources provides also the order of the parameters used in the calibration process, beginning at the bottom of the list. The most reliable data on top is only changed if there is a good reason, for example the review of the data proves wrong and new insights change the validity of the data.



## 3.2 Hydraulic Calibration - Pressure Losses

The hydraulic calibration ensures the correct flow rates in all fluid circuits and therefore the fluid mass distribution at steady state conditions.

### 3.2.1 Preparations

First all sets of vehicle measurements were scanned for steady-state conditions with constant flow rates and almost constant temperatures at thermostat fully opened and fully closed conditions. Those results got scaled to the rated speed of the engine and compared to internal benchmark data to ensure a plausible flow distribution. Two operating points at higher than idle rotational engine speed got chosen to avoid errors due to the limitations of flow meters at lower than their design flow rates.

Additionally the pressure levels got reviewed on artifacts and anomalies. For example, one pressure gauge showing, although downstream the oilcooler, a higher pressure level than at the inlet. Investigations prove this gauge to be wrong or at least measuring the static pressure with a portion of the dynamic pressure. A not fully perpendicular towards the flow direction oriented sensor may introduce influences of the dynamic pressure to the measurement signal. Luckily this wrong gauge is not crucial and could be compensated by other gauges in the path.

This reviewed input data provides the pressure heads of all pumps at a given volumetric flow rate as well as the pressure loss characteristics of all measured sections in the circuit. In order to complete the hydraulic setup the fluid volumes were taken from the specification of the reference vehicle.

### 3.2.2 Workflow

Usually, when using the software Flowmaster, all measurements become post-processed and translated into a pressure loss characteristic either pressure loss vs. flow rate or loss coefficient vs. Reynolds Number. These characteristics and the fluid volume can be directly used in Flowmaster, because the fluid volume is independent of the pressure loss characteristic.

### 3 Calibration of the Thermal Model of the Real-Time Capable VTMS Model

In contrast to the state-of-the-art approach in BoostRT, at time of writing this document, the losses are modeled as pipes and rely on geometrical information to model the friction in the pipes. Therefore the piping diameter and the coolant volume of each section of the circuit was used to setup the pressure losses, as described in section “Details on Modeling” (2.4.4). The pumps are fitted towards the measured performance and pressure head characteristics. Since the pressure loss of the simplified and on geometrical data based pipes will not behave as measured in the circuit, the BoostRT provided friction multipliers were used to calibrate the flow distribution for both thermostat fully opened and fully closed condition at the measured temperature without any heat transfer in the model. The same was done for all heat exchangers and all components got collected to build up the thermal network.

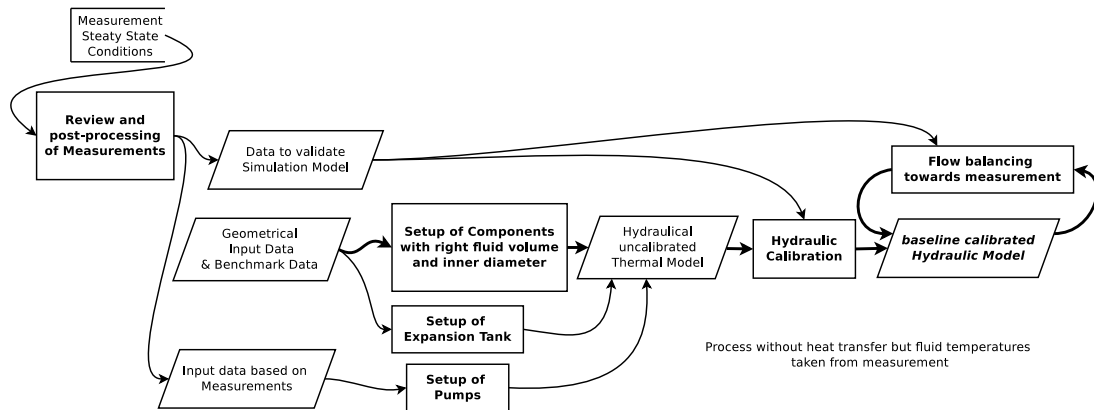


Figure 3.2: Hydraulic calibration process.

Summarized steps of hydraulic calibration in the simplified real-time approach, either for each component or united for the whole circuit:

1. Input of all geometrical information to create all relevant components, including heat exchangers, pipes, valves and expansion tanks of the circuits.
2. Setup of expansion tank(s) based on geometrical dimensions.
3. Modeling of the pumps according to measured input data.
4. Calibration of pressure losses to provide measured flow distribution by means of friction multipliers at measured temperatures without considering any mechanisms of heat transfer in the simulation.

### 3 Calibration of the Thermal Model of the Real-Time Capable VTMS Model

5. If not already done, connection of all components to build up the fluid circuits and review of calibration.

#### 3.2.3 Result

After this step the thermal model is able to calculate the fluid mass flow rates, therefore the coolant/oil distribution, at thermostat fully opened and fully closed condition. Additionally it is capable to calculate the power consumed by the pumps.

### 3.3 Calibration of Thermal Transports

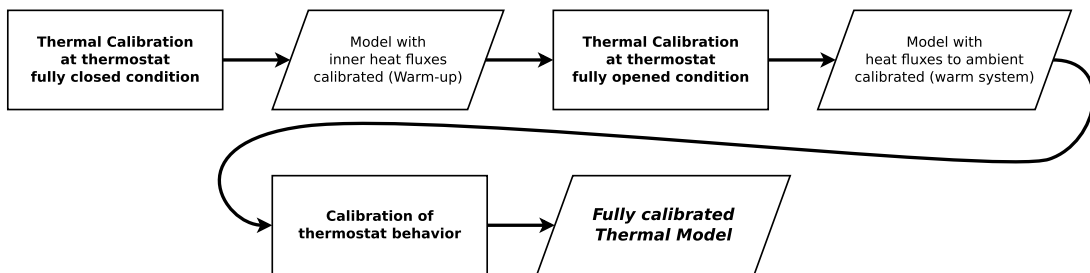


Figure 3.3: Overview on calibration process of thermal transports.

In this section the rest of the components will become part of the thermal network and it will become integrated into the Cruise environment and the thermal behavior of the model will get calibrated to meet the measurements. The thermal calibration is split in three steps:

1. The calibration of the heat distribution before the thermostat starts to open, therefore without the influence of the main heat exchangers to ambient and the air path.
2. The calibration of the main heat exchangers at in best case thermostat fully open condition.
3. Finally the thermostat characteristics itself to ensure a proper flow distribution even in interim positions.

### 3.3.1 Preparations

The preparations for the thermal calibration follow the same guidelines as the hydraulic calibration but with attention to different aspects in the circuit. At least from this point on the calibration is done by using transient heat transfer simulations of the thermal model. In order to ensure matching initial conditions in measurement and simulation, only measurement data captured on a well conditioned vehicle can be used as reference for calibration.

The best way to calibrate the thermal transports is to use warm-up measurements with constant load or at least with as less transients as possible. In order to validate the incoming heat fluxes, if possible, several measurements at different loads are preferred. These measurements have to contain the full heating up process at a preferable constant load until all temperatures reach a steady-state condition and therefore can be used for all three steps of the thermal calibration.

If there are no warm-up measurements available, a measured New European Driving Cycle (NEDC) [1] may get used for the first step of calibration, up to the point of opening of the thermostat. The rest of the NEDC measurement may get used for validation purposes. In addition to the NEDC at least one measurement at thermostat fully open has to be used in the second step of the thermal calibration process to get matching steady state temperatures. In this case again the NEDC, more specific the last roughly 300 seconds of the NEDC at thermostat partly open condition may get used to calibrate the opening characteristic of the thermostat.

Before going into the details of the calibration loops, the heat fluxes between the locations of measurement were calculated in order to not just compare resulting temperatures but also the resulting heat fluxes to get a fast convergence between simulation and measurement.

### 3.3.2 Workflow

From this point on, the simulation is done, for the first time as transient cycle simulation. In order to run this type of simulation, the thermal model has to be extended by the air path, the thermal masses and the Cruise interface,

### 3 Calibration of the Thermal Model of the Real-Time Capable VTMS Model

using the conceptions from chapter “Details on modeling” (2.4). At first this fully featured thermal model was integrated, by compilation, in the existing Cruise environment and configured to run cycle simulations.

If the fully featured VTMS simulation model, consisting of Cruise coupled with BoostRT, is setup the right way and capable to run cycle simulations, the calibration process containing its three steps is started.

1. Calibration of the warm-up behavior, in other words the heat distribution from input towards thermal masses, with inactive heat exchange to ambient (thermostat fully closed).
  - a) Calibration of heat transfer of auxiliaries.
  - b) Calibration of heat fluxes from engine to coolant.
  - c) Calibration of heat fluxes from engine to oil.
  - d) Calibration of heat fluxes between coolant and oil.
2. Calibration of the steady-state resulting at thermostat fully open condition.
  - a) Calibration of Heat transfer to air path and therefore to ambient.
3. Calibration of thermostat behavior in interim positions.

#### Warm-up Calibration

The first step of the thermal calibration deals on heat fluxes in the system, from the sources to the thermal masses. The period of interest ends with the opening of the thermostat. Therefore Cruise was adjusted to use the measured BMEP, rotational speed and vehicle velocity as well as the ambient temperature from the measurements and the coolant and engine mass temperature of BoostRT, in order to determine the operational point of the engine and as a consequence the heat fluxes sent to the thermal model.

Firstly the heat transfer at the auxiliaries, single masses connected to a pipe (e.g. the hybrid components in LT circuit as well as turbo charger and EGR cooler), were calibrated to come near the transferred heat fluxes of the measurement by adjusting the heat effective area and the heat transfer multipliers. The problem at this step is that the heat fluxes are driven by temperature differences but not

### 3 Calibration of the Thermal Model of the Real-Time Capable VTMS Model

all temperatures are known from measurement. Especially the temperature of the solids are missing.

For example at the turbo charger, the heat input to the solid mass, is delivered by the engine maps, but the resulting temperature of the solid and the heat transfer coefficient to the coolant are unknown. Additionally the whole process of heat transfer is transient. In contrast to the structural side of the system the volumetric flow rate, therefore by including the density the mass flow rate, as well as the inlet and outlet temperatures are known from measurement. The best way to calibrate the heat transfer at such a component is to fix the temperature at component inlet and ensure that the flow rate, calibrated in the hydraulic calibration process meets the measurement. Under these boundary conditions the transferred heat to coolant at the component as well as the outlet temperature can be compared to the measurements. For convergence the heat transfer multiplier provided by BoostRT was used, to scale the heat transfer coefficient based on the equations of Gnielinski, or the heat effective area can be adjusted to get matching outlet temperatures between simulation and measurement. As a consequence the transferred heat at the component will be the same as measured.

All components represented by one pipe and one connected solid mass were calibrated by this process, but the engine as well as the oil circuit contain much more uncertain parameters.

The next step was the calibration of the oil cooler (heat exchanger between oil circuit and coolant circuit) based on the same principle as before but with fixed temperatures both inlets, oil and coolant. At the oil cooler the unknown parameters are the mass flow rate of the oil, based on benchmark data, and the heat transfer coefficient as well as the heat effective area.

Finally the calibration process for the engine is still missing. In the thermal model the engine is based on the state-of-the-art implementation of the thermal masses and gets heat inputs provided by engine maps in Cruise, from friction as well as from the combustion process. This incoming heat fluxes have to pass and heat up, several masses before reaching the boundary to coolant or oil. Since all measurements at the AVL L.I.V.E. vehicle were done in a manner to keep all components intact, the temperatures of the solids representing the engine are not available in measurement. To make matters worse the

### 3 Calibration of the Thermal Model of the Real-Time Capable VTMS Model

engine thermal mass model is based on a split heat transfer to coolant into four parts, block, intake, head/firedeck and exhaust. The calibration can only be done for the overall heat input to coolant or oil at the engine, due to this modeling concept and the uncertainty regarding the local temperatures as well as the heat effective areas and local velocity levels. For this reason it is a heavily iterative process by adjusting heat effective areas, multipliers as well as shifting small amounts of mass from the coolant heat effective regions to the masses representing the rest regions of the mass model and vice versa.

This process ends as the transferred heat to coolant as well as the mix temperatures converge towards the measurements without any manually applied boundary conditions included in the thermal model and the heat distribution is calibrated in the thermal network but up to this stage all energy stays in the system.

#### **Hot Calibration of Steady State System - Heat Transfer to Ambient**

This step deals on the calibration of the heat exchangers to ambient, or in other words it is the calibration of the air path. At the coolant side the heat exchangers are already calibrated regarding flow rates and the calibration regarding heat transfer was done in a similar way as before.

The known values from measurement at the heat exchangers at the liquid side are the temperatures at inlet and outlet, the mass flow rate, calculated from volumetric flow rate and density of the fluid, therefore the transferred heat. Additionally the velocity of the vehicle as well as the ambient temperature are known from measurement. Due to the limited in-vehicle measurements and no complete benchmarking measurements of the radiator performance characteristics at the component-testbed, the heat transfer characteristic between coolant and air could not be determined only through measurement and for this reason the characteristic is based on benchmark data.

The calibration is done by isolating the heat exchangers and fixing the inlet boundary conditions at coolant side and air side as good as possible. At the coolant side, the volumetric flow rate should already match to the measurements, due to the hydraulic calibration. Additionally the coolant and air inlet temperatures as well as the vehicle velocity are fixed to measurement. The

### 3 Calibration of the Thermal Model of the Real-Time Capable VTMS Model

pressure losses of the air path as well as the cross section area and therefore the mass flow rate is used to calibrate the heat exchangers to meet the measured temperature at coolant outlet and as a transferred heat.

After this first loop the air path is assumed to be calibrated but the heat transfer characteristic of the heat exchangers has to be checked and adjusted for several operating conditions to ensure a proper thermal behavior. By this step in the calibration process the thermal system should be preliminary capable to behave in the same way as in reality with exception of the thermostat being in interim positions.

#### **Thermostat Calibration**

Finally the last step of the thermal calibration sizes the thermostat behavior. This step is chosen to be the last one because the thermostat behavior relies heavily on the coolant temperature and the coolant temperature depends strongly on the heat exchanger performance. Again it was an iterative process to calibrate the thermostat. A process that may have to be fine tuned even in future as the results (chapter 4) demonstrate.

In this step the most important part is not the thermostat itself, but the thermostat and the heat exchanger to ambient. The thermostat controls the volumetric flow rate through the heat exchanger and the bypass by its opening ratio and its opening ratio is controlled by the temperature controlled expansion of the wax pellet.

The best way to calibrate the opening ratio is to fix the coolant temperature to the measured reference coolant temperature and adjust the opening ratio vs temperature characteristic to make the volumetric flow rates in bypass and heat exchanger path matching the measurement. This method is the preferred way, if a full warm-up measurement is available. The hysteresis offset as well as the time constant is best calibrated using a NEDC cycle or a similar cycle with the thermostat in interim condition. Important in this case is a non-constant load that makes the thermostat to raise and to lower its opening ratio over time.

Although this step can be done much earlier in the calibration process it is named at last, because of the type of input data needed and its function as



### 3 Calibration of the Thermal Model of the Real-Time Capable VTMS Model

indicator for wrong calibrated circuits. If the reference temperature is not fixed to measurement any more but there is still either a wrong temperature or flow distribution in the circuit, both the heat inputs to coolant as well as the heat rejected at the heat exchanger to ambient have to be crosschecked and the steps 1 and 2, calibration of heat distribution to and from coolant have to be repeated.

#### 3.3.3 Result

The result of these calibration loops is a fully featured thermal model of the VTMS simulation capable to run cycle simulations. At this stage it is already embedded into the virtual vehicle and ready for additional calibration loops to fine-tune the simulation.

## 3.4 Additional Calibration Loops

The preparations and the workflow are almost the same as in the section before but without the process of adding components and connections. Instead of a specific heating up cycle at constant load and speed a New European Driving Cycle (NEDC) [1], if not already done, is taken to calibrate the thermal model. Additionally a real world driving cycle may get used to get a deeper insight in the behavior of the fan and the air path, resulting in a fully calibrated VTMS model. At this stage the attention should be directed even more on heat distribution. Without respecting the heat distribution most likely the thermostat and its opening behavior may get suspected to be wrong, but in most cases, due to missing parameters, the air path and therefore the radiator is the source of this misbehavior. The thermostat react only to the wrong heat rejection at the radiator.

## 4 Results and Comparisons

Finally the results of the simulations are compared to the state-of-the-art approach as well as to measurements.

### 4.1 Hardware and Software Setup used for Simulation

All simulations are done using a Windows 7 environment hosted in a virtual machine on a Workstation running Red Hat Enterprise Linux 5.9. The workstation is powered by two Xeon X5670 CPUs running at a clock speed of 2.93 Ghz. There are 4 CPU cores and 4 Gb of RAM assigned to the virtual machine. This setup applies to both the real-time approach as well as the state-of-the-art approach. Additionally to increase the performance of Flowmaster used to run the state-of-the-art approach, the required Flowmaster database runs on the same machine to reduce performance losses due to network activity.

The time steps used in Cruise for both thermal networks, Flowmaster and BoostRT, are fixed at 0.5 seconds. The Cruise master simulation runs with a time step of 0.1 seconds in case of the state-of-the-art approach. In the real-time approach this time step was one tenth of the time step in BoostRT, in other words 0.05 seconds.

### 4.2 Input/Boundary of Simulation - cold NEDC

A fully featured VTMS simulation has usually only the ambient conditions as well as the road profile as boundary condition. The road profile has to contain at least the desired vehicle velocity profile. The driver simulated in

## 4 Results and Comparisons

AVLs environment by Cruise should use these boundaries and drive the vehicle as described in the profile.

In order to compare the BoostRT model with its state-of-the-art counterpart as well as the measurement, the relation between the driver, the virtual vehicle and the thermal model is reduced to a minimum. Instead of the before mentioned boundary conditions, the simulation runs totally on measured input parameters, to eliminate errors due to wrong decisions of the control strategy or the driver.

Figure 4.1 displays the boundary condition of the simulation, taken from measurement.

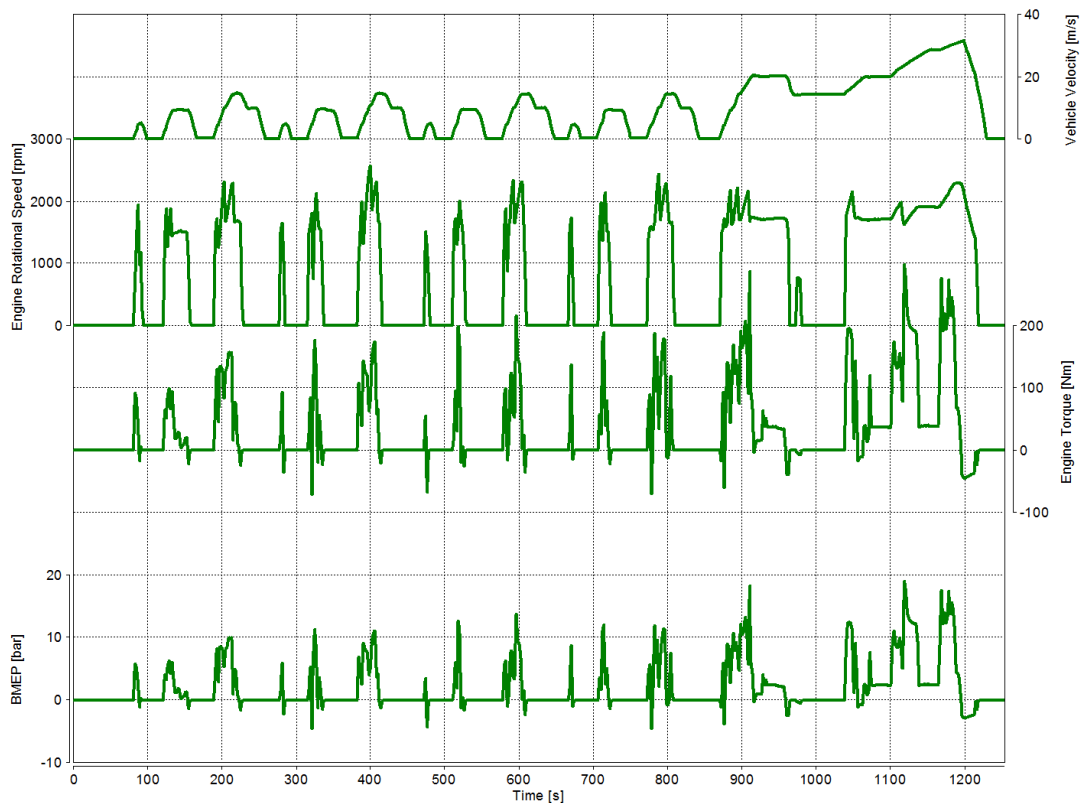


Figure 4.1: Input signals to run NEDC from measurement.

- The vehicle velocity, same as the desired velocity to define the road profile,
- the rotational engine speed, to ensure the use of the right gear as well as the right hybrid strategy.
- the torque to eliminate possible inconsistencies of the drivetrain model,

## 4 Results and Comparisons

- the BMEP to get the right operational point in the engine maps
- and finally not listed the ambient temperature, fixed to 20 deg celsius.

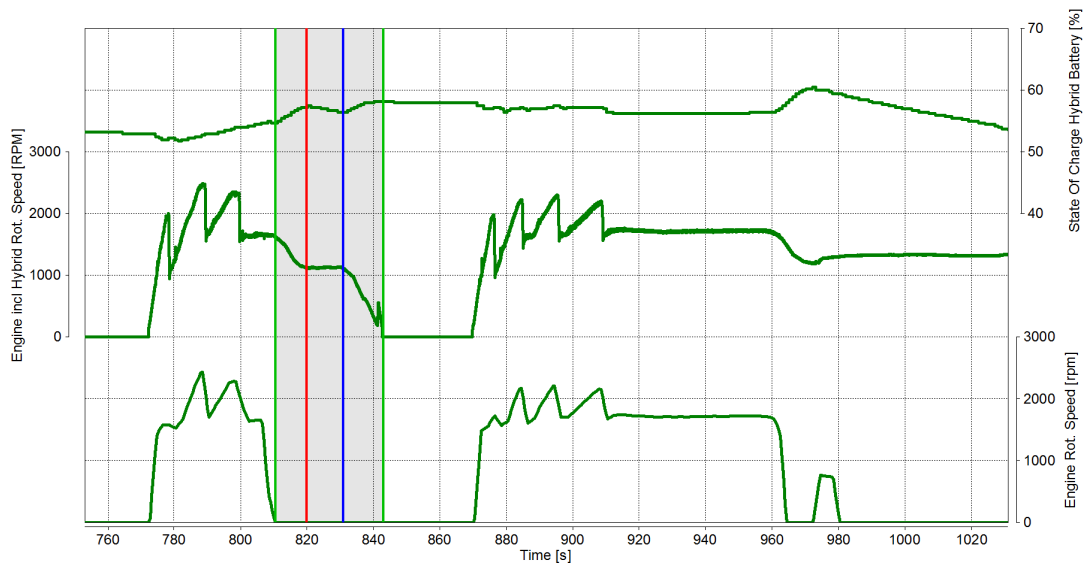


Figure 4.2: Close look on driving condition while NEDC - activated hybrid system.

Figure 4.2 displays a closer look on the driving condition. This data is no input data but should provide a view on the details. It is clearly visible that, for example between second 810 and 843, the combustion engine is turned off and the electrical motor is used either to drive the car or the recuperate energy and charge the batteries. At this stage of development, the Cruise hybrid control and the driver is not capable to provide the same behavior. For this reason the above directly measured input data was used to run the calculation. Furthermore the heat rejection of the hybrid system is still in development in Cruise and not scope of this work, it is not connected to the thermal model. The hybrid system affects only the heat fluxes of the LT cooling circuit but has no influence on other parts of the thermal model. Due to this fact it has to be evaluated in future when the hybrid controllers are ready.

### 4.3 Real-Time Approach vs State-of-the-Art Approach - cold NEDC

An excerpt of the results of the vtms simulation is displayed in this section. It should provide an overview on the results with detailed views on those parts,

## 4 Results and Comparisons

which are the most difficult components to calibrate: the thermostat and the radiator.

The following legend applies to all graphs. Any divergence to this legend is noted separately at the diverging graph.

**BoostRT Results** are marked by a black solid line.

**Flowmaster Results** are displayed using a dashed blue line.

**Measurement Data** is marked by green solid line.

### 4.3.1 Hydraulic Conditions of the HT Coolant Circuit

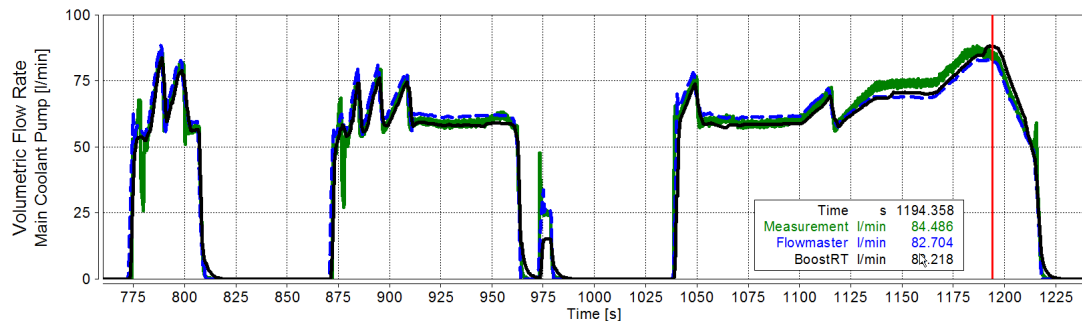


Figure 4.3: Volumetric Flow Rate at the Main coolant pump in the last 300 seconds of a cold NEDC.

At first step figure 4.3 displays the well matching volumetric flow rate at the HT coolant pump. The inertia of the pump used in BoostRT is a slight bit too high and therefore the following of the transients is not as good as in the Flowmaster based state-of-the-art simulation, but the overall performance seems to fit.

In contrast to the rest of the circuit the volumetric flow rate of the radiator path depends on the opening ratio of the thermostat. Figure 4.4 shows the flow rate at the radiator. The flow rate provided by measurements is quite smooth but at the beginning of thermostat opening the increase in flow rate is much higher than in the simulations. A reason might be in the modeling and setup of the thermostat used in the simulations. Although both simulation approaches catch the right moment of the beginning of the opening of the thermostat, both show potential for improvement. At this point it has to be noted that the air path and therefore the heat rejection performance of the radiator as well

## 4 Results and Comparisons

as the thermostat opening behavior is still under heavy development in the state-of-the-art approach.

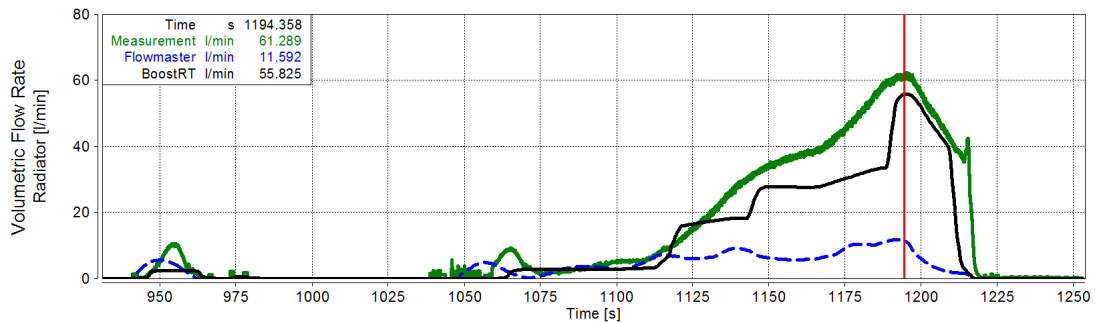


Figure 4.4: Volumetric flow rate at the radiator in the last 300 seconds of a cold NEDC.

Furthermore the black line BoostRT result of the flow rate at the radiator shows clearly the hysteresis, and therefore the opening behavior of the setup thermostat model. The behavior might get improved by increasing the thermal inertia of the thermostat wax pellet and as a consequence resulting in the slower reaction times by lowering the reference temperature of the opening characteristics in order to smooth out the opening. Another indicator for a too fast reacting thermostat in the BoostRT model is the fact that it closes the radiator path while the engine is not active. The thermostat in the real vehicle seems to stay partly opened in this case.

### 4.3.2 Temperatures at the Radiator

This section should start in the same manner as the section on volumetric flow rates before. It displays a mixing temperature, the one at the thermostat, in figure 4.5. Again both simulation approaches match the overall temperature development of the circuit quite well.

The temperatures at the radiator outlet (figure 4.6) show the same behavior as the volumetric flow rates in the section before. It is clearly visible that the state-of-the-art approach has a far too low radiator outlet temperature compared to the measurements. But due to the low volumetric flow rate this misbehavior has almost no consequence on the overall system temperature, at least in this case. As a consequence the misbehavior is not visible at the engine maps and the fuel consumption (section 4.3.4). The result of the BoostRT

## 4 Results and Comparisons

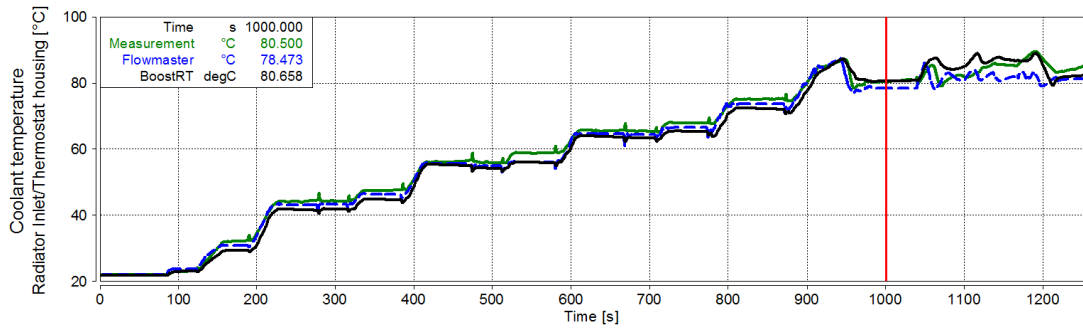


Figure 4.5: Coolant temperature at the thermostat, cold NEDC.

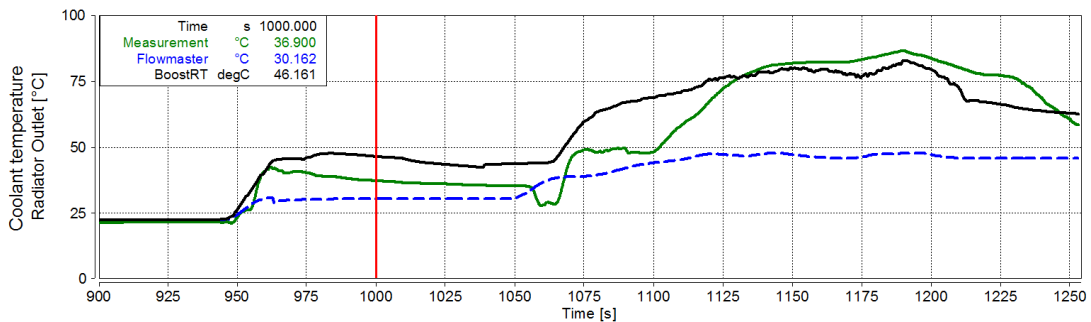


Figure 4.6: Coolant temperature at the radiator outlet in the last 300 seconds of a NEDC.

model shows the influences of the wrong volumetric flow rates from the section before also in the temperature result. It renders a similar figure as the measurement, but similar to the volumetric flow rates, it is not capable to follow the measurement completely.

### 4.3.3 Heat Transfer at the Radiator

The following section deals on the heat transfer at the radiator and displays the room for improvement as well as the problems to evaluate measurements.

Figure 4.7 displays the comparison of the heat flow rate at the radiator. In contrast to the parameters before (temperature and volumetric flow rate) the heat flow rate is not directly measured at the vehicle but calculated based on these measurements.

There are two approaches used to calculate the heat flow rate, the first without considering the deviation of the inner energy of the coolant in the section and

## 4 Results and Comparisons

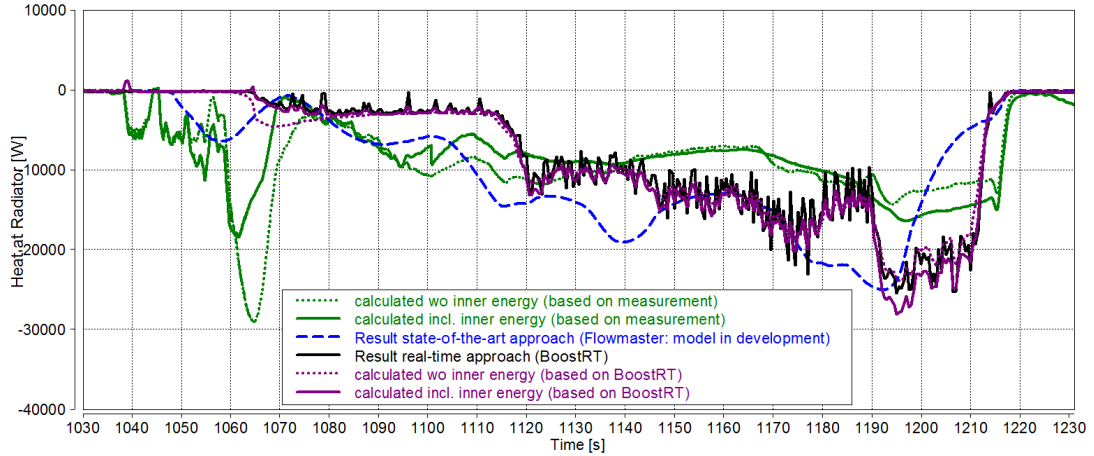


Figure 4.7: Heat flow rates at the radiator at the end of a cold NEDC.

the second including a model to estimate the change of inner energy in this section. The dotted lines display the result of the first simplified approach the solid line is either directly taken from the software packages or in case of the violet lines calculated in the same manner as the measurement based results.

The equations used for the manual calculation with the simplified approach (dotted line in figure 4.7):

$$\dot{Q} = \rho|_1^2 \cdot \dot{V}_{rad} \cdot c_p|_1^2 \cdot (T_2 - T_1) \quad (4.1a)$$

$$T_1 = T_{thermostat} \quad (4.1b)$$

$$T_2 = T_{radiator\ outlet} \quad (4.1c)$$

This approach balances the inlet and outlet, but does not consider the fluid mass between the two locations of measurement. As a consequence there is a mismatch at the beginning of the opening of the thermostat. This mismatch lasts until the cold fluid mass in the radiator is exchanged with fresh hot coolant from the engine. Furthermore this graph displays the need of further calibration especially in conjunction with the figures 4.6 and 4.5, displaying the temperature at radiator inlet and outlet, and figure 4.4 containing the volumetric flow rate at the radiator. It is clearly visible although the inlet temperatures are not far off the measurement, the heat transfer as well as the mass flow and the outlet temperature is quite off the measurement. Additionally



## 4 Results and Comparisons

the hysteresis of the thermostat is at least in the BoostRT model visible. In this particular case, especially in the BoostRT model, the air path and the thermostat still need calibration.

Based on the BoostRT temperature and the measurements a second manual calculation including an estimation of the inner energy at the radiator was performed (solid lines with “incl. inner energy” in figure 4.7).

$$\dot{Q} = \rho|_1^2 \cdot \dot{V}_{rad} \cdot c_p|_1^2 \cdot (T_2 - T_{1_m}) + \rho_m \cdot V_{rad} \cdot c_{p_m} \cdot \frac{dT_m}{dt} \quad (4.2a)$$

$$f = \frac{\int_t \dot{V}_{rad} dt}{V_{rad}} \quad (4.2b)$$

$$\dots f \geq 1 \rightarrow k = 1 \wedge k < 1 \rightarrow k = f \quad (4.2c)$$

$$T_1 = T_{thermostat} \quad (4.2d)$$

$$T_2 = T_{radiator\ outlet} \quad (4.2e)$$

$$T_{1_m} = T_1 \cdot k + T_2 \cdot (1 - k) \quad (4.2f)$$

$$T_m = \frac{T_{1_m} + T_2}{2} \quad (4.2g)$$

This approach estimates the average temperature inside the radiator by calculating the average between the outlet temperature of the radiator and the volume weighted inlet temperature of the radiator. Volume weight means the inlet temperature is scaled between the outlet temperature and the actual inlet temperature by the fill level of the radiator with fresh hot coolant. This approach still does not consider the duration while the combustion engine is switched off while the vehicle is driven by the electrical motor. In case of the manual calculation the temperature in the radiator stays constant but in reality it would decrease.

In order to validate this procedure, it is applied on the measurement as well as on the BoostRT based results. It shows a quite good matching on the BoostRT results between directly simulated heat fluxes and the manually calculated ones. In case of the measurement it shows a quite significant decrease of the peaks at thermostat opening but still these results have to be assessed carefully. On the one hand side the volumetric flow rate at least at the beginning of the opening of the thermostat is quite low and therefore at the lower limit of the measurement equipment and on the other side the temperature

## 4 Results and Comparisons

measurements have an accuracy of 0.1 degree Celsius, leading to uncertainties in the calculation of the transferred heat. Those two circumstances make the evaluation of the results much harder, because without smoothing out of the measured parameters the result based on the measurement would not be readable in the figure above. For this reason, although the heat flux is required for the calibration, it is best to validate the simulation results at all measured parameters of a component.

### 4.3.4 Fuel Consumption

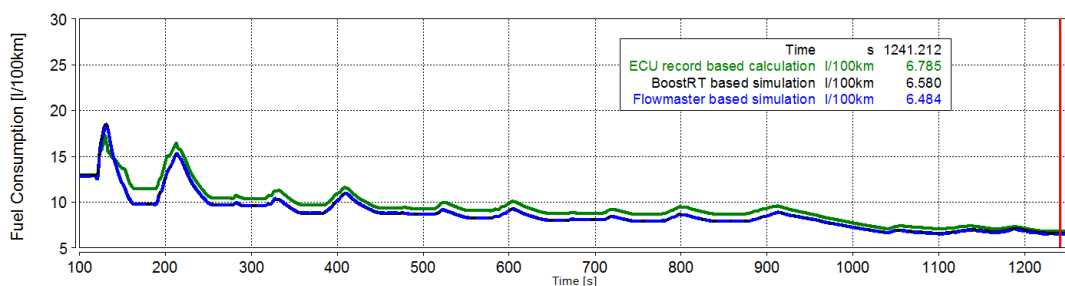


Figure 4.8: Fuel consumption at cold NEDC compared to ECU.

As displayed in figure 4.8, both approaches calculate a similar fuel consumption. There is a slight deviation to the ECU measured fuel consumption, especially at low circuit temperatures. Since both approaches show this behavior either the reference temperature used in the engine maps is wrong and needs further investigation or the fuel consumption maps have to be re-calibrated to compensate this deviation.

The next two figures display the error in detail. These figures also include the measurements of the KMA, short for “Kraftstoffmessanlage” (german expression for fuel measurement system). The KMA is a system to measure the fuel mass flow transparently, in other words the KMA tries to be invisible for the fuel intake system of the vehicle and the ECU. It was used to validate the signal containing the recorded fuel mass flow provided by the ECU, but shows a quite intense deviation at the beginning of the measurement for around 250 seconds.

Figure 4.10 displays the error in a more detailed view. It is clearly visible that the biggest error occurs, in both measurement and simulation at engine

## 4 Results and Comparisons

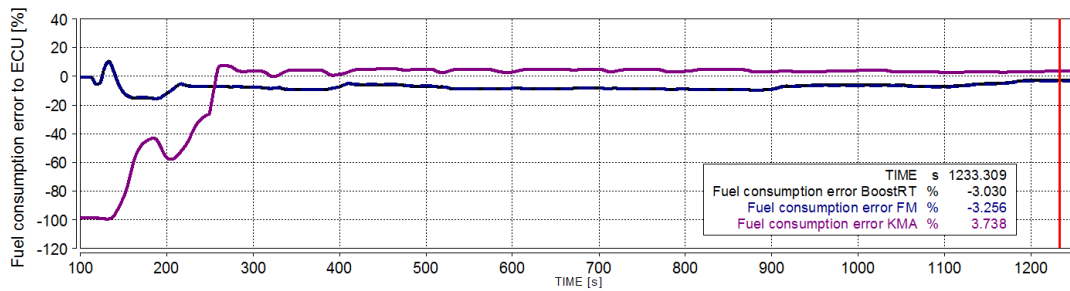


Figure 4.9: Relative error of calculated cumulative fuel consumption.

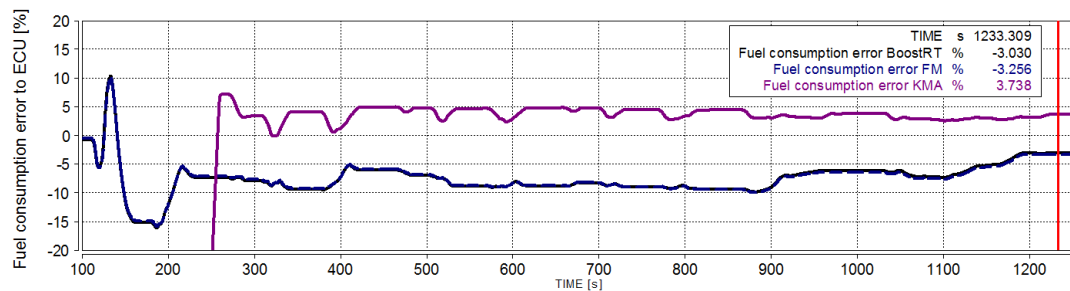


Figure 4.10: Relative error of calculated cumulative fuel consumption.

off conditions. The reason seems to be the transients at start and stop of the engine. This behavior needs further investigations in order to reduce the, at least most of the time, heavy deviations, in case of the the KMA up to a constant 5 % and in case of the simulations of up to 10 % cumulating to a deviation of around 3 % over the whole NEDC. A reason for this deviation, with exception of the first 250 seconds, might be either errors in KMA and or artifacts in the ECU signal due to engaged start/stop mechanism or in case of the simulation the reference temperature delivered to the engine maps is not matching the reference temperature used to setup the engine model. The displayed numbers are calculated under the assumption the ECU signal provides valid information. If this assumption is true is still open at the time of writing this work, but demonstrates that any measurement as well as any calculation is affected by its own errors and any information has to be confirmed at first.

### 4.3.5 Calculation Speed

Finally this subsection takes a closer look on calculation speed, the prime target of the real-time setup.

## 4 Results and Comparisons

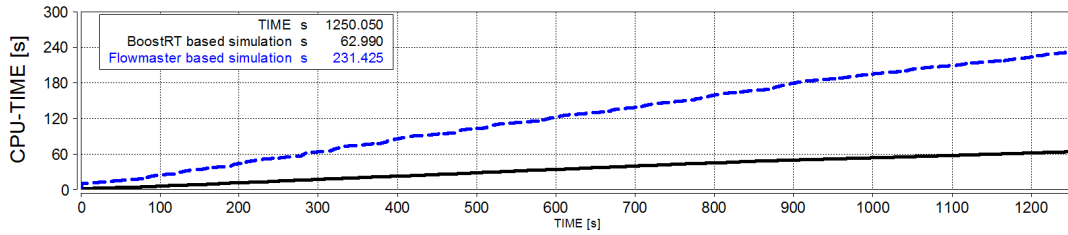


Figure 4.11: CPU Time consumed vs real time.

In figure 4.11 the difference in calculation speed is clearly visible. The real-time-factor (figure 4.12) shows the difference even more significantly than the CPU time comparison. It is visible that any transient effect in the calculation slows down the state-of-the-art approach, in some condition even more than real-time.

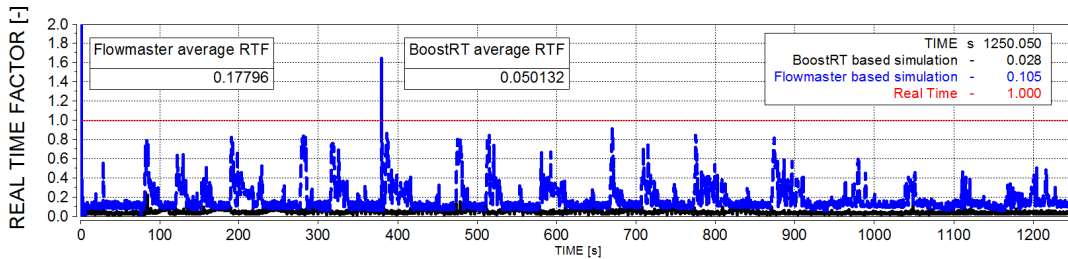


Figure 4.12: Real Time Factor of both approaches.

The calculation speed can be increased even further, by increasing the size of the time steps used in the calculation. But at least this model shows significant decrease in accuracy, if the time step of the Cruise side, the vehicle dynamics, is increased by the factor 2.

## 4.4 View on Heat Distribution - cold NEDC

This section demonstrates excerpts of energy balances and evaluated heat fluxes. Although the results of the heat fluxes (section 4.3.3) demonstrate that the heat fluxes, at least to ambient, are questionable, in both measurements and simulation, they can be used to illustrate the thermal behavior much better than any other parameter.

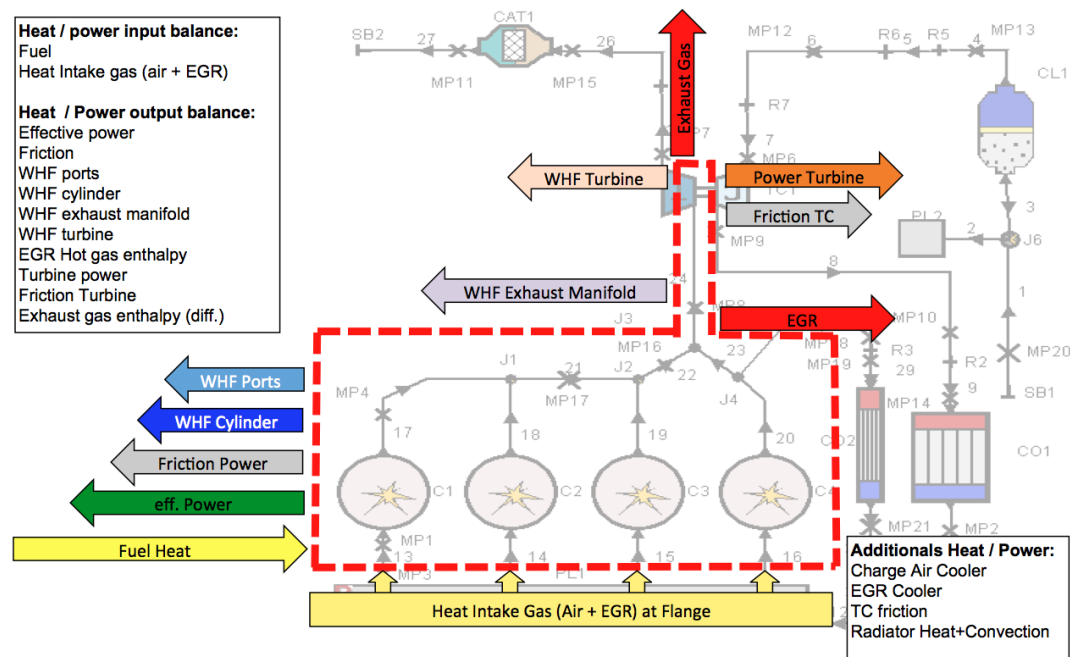


Figure 4.13: System borders of engine energy balance (border of engine maps) with additional heat/power not part of the energy balance but considered separately in the engine maps.

Figure 4.13 displays the energy fluxes at the borders of the combustion process. These borders are the ones used in the engine maps. The heat fluxes in the lower right corner, for example the heat flux at the charge air cooler or at the EGR cooler are evaluated separately since those components are not inside the system borders but are also modelled in Cruise by the same approach as the engine maps.

The first figures should demonstrate the energy conversion of the combustion process to effective work and losses while driving an NEDC. The energy of the EGR gas, the sum of the cumulated wall heat fluxes and the friction directly affect the thermal behavior of the system. The energy of the exhaust gas is lost

## 4 Results and Comparisons

to ambient, the same applies to the charge air cooler. At the air cooled charge air cooler parts the turbine power used to rise the pressure at the intake is rejected directly to ambient. A portion of the EGR gas enthalpy is rejected to structure and to coolant at the EGR cooler and the rest becomes part of the enthalpy of the intake gas at the inlet manifold. Only a small portion converts to effective work to drive the powertrain.

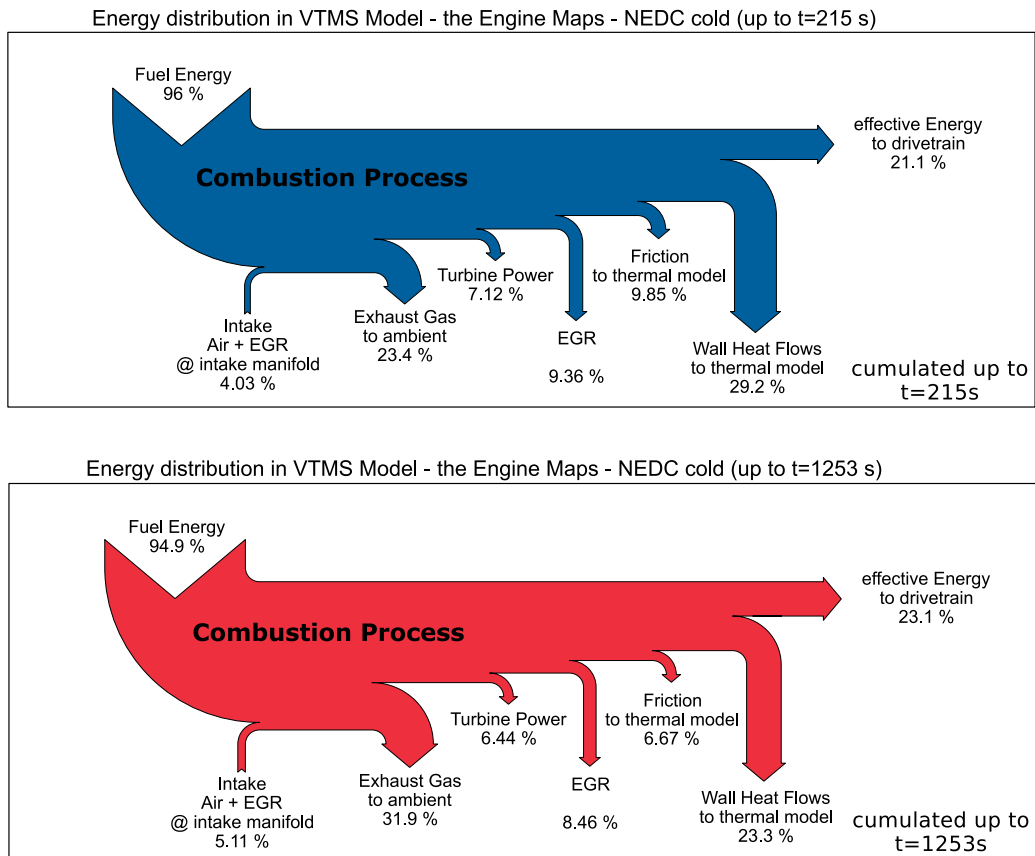


Figure 4.14: Cumulated energy distribution of combustion process in percent using Sankey diagram (cold NEDC) above: cumulated up to 215 seconds, below: cumulated up to 1253 seconds.

Figure 4.14 shows the difference between the energy conversion cumulated up to the first 215 seconds and up to the end ( $t = 1253$  s) of a cold conditioned NEDC. The reduction of friction and heat input to the thermal model as well as the increase in effective work over time and increasing temperature is visible.

## 4 Results and Comparisons

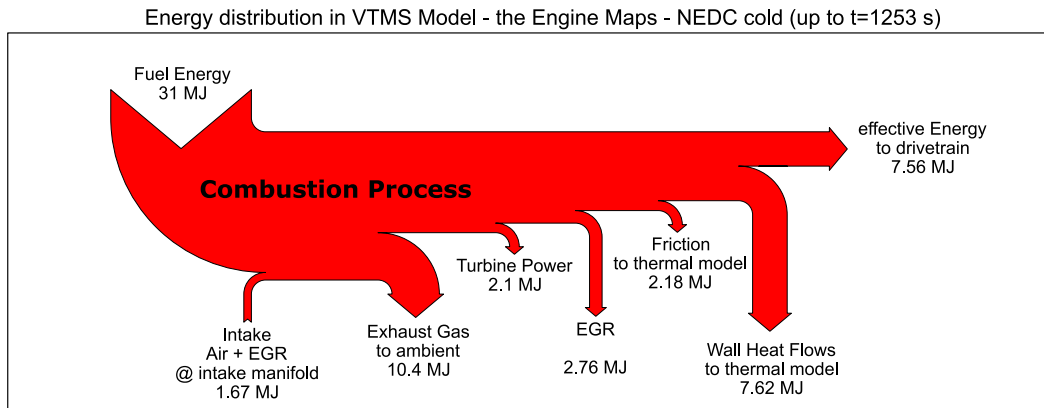


Figure 4.15: Cumulated energy distribution of combustion process using sankey diagram (cold NEDC).

Figure 4.15 displays the cumulated energy conversion in absolute numbers at an NEDC, started with a cold conditioned vehicle.

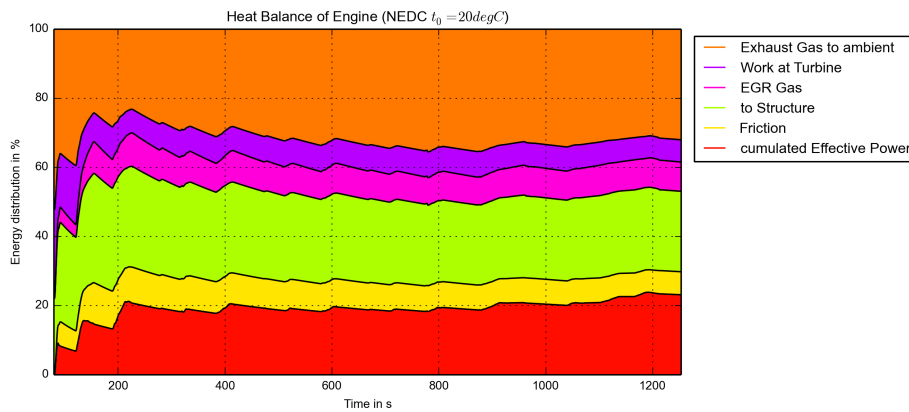


Figure 4.16: Energy conversion of combustion process in percent over time (cold NEDC).

Figure 4.16 illustrates energy conversion over the whole NEDC. It demonstrates the change in the distribution of energy even better than the sankey diagrams before.

## 4 Results and Comparisons

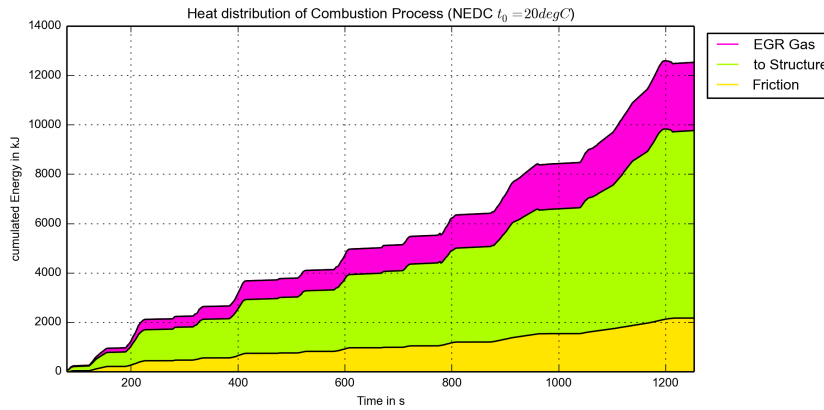


Figure 4.17: Cumulated heat input to thermal model (cold NEDC).

Figure 4.17 displays the cumulated energy input to the thermal model being 12.5 MJ. As displayed in the next figures only a fraction (24%) is reaching the coolant or is even rejected to ambience during the NEDC. Only 12% of this heat input is rejected to ambient during the NEDC. 88% of the heat input to the thermal network is stored in the structure, oil and coolant.

The next figures display the calculated cumulated heat input to the HT coolant circuit and the heat taken from the coolant, in absolute numbers as well as in percent.

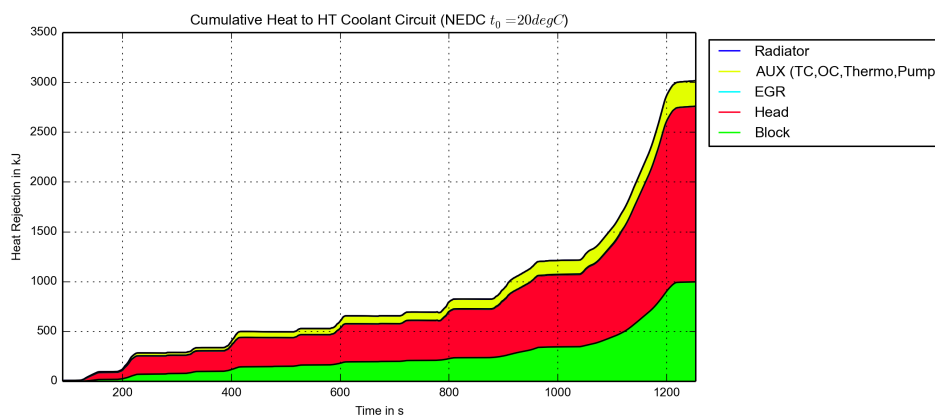


Figure 4.18: Cumulated heat input to HT coolant circuit (cold NEDC).



## 4 Results and Comparisons

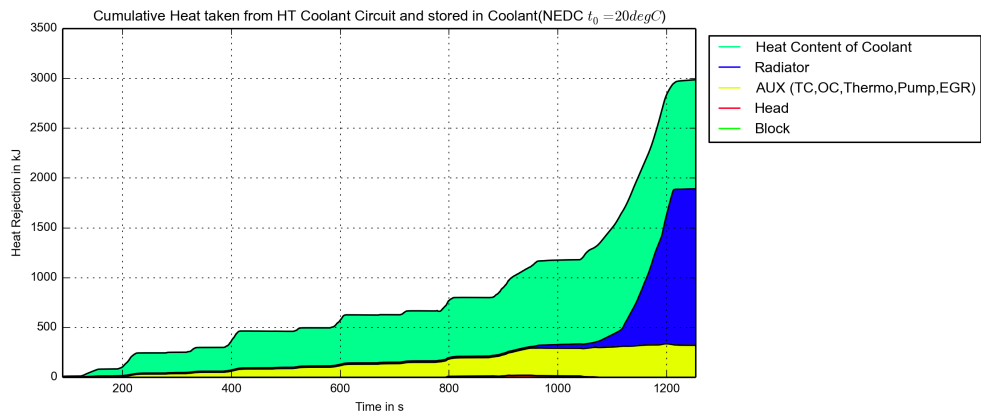


Figure 4.19: Cumulated heat taken from HT coolant circuit (cold NEDC).

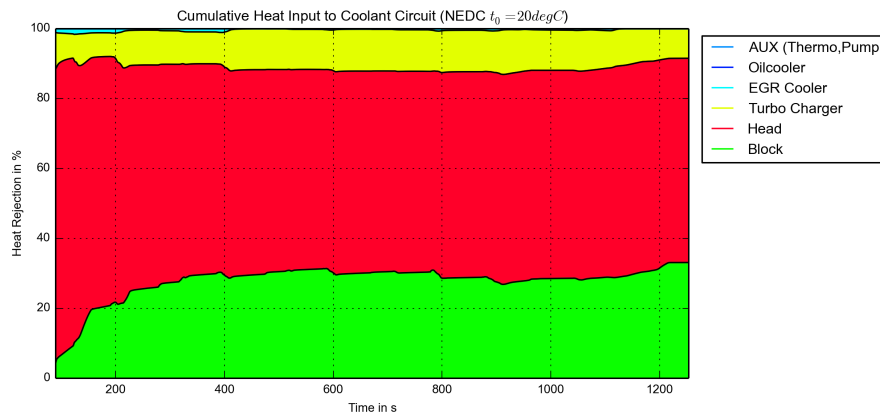


Figure 4.20: Cumulated heat input to HT coolant circuit in percent (cold NEDC).

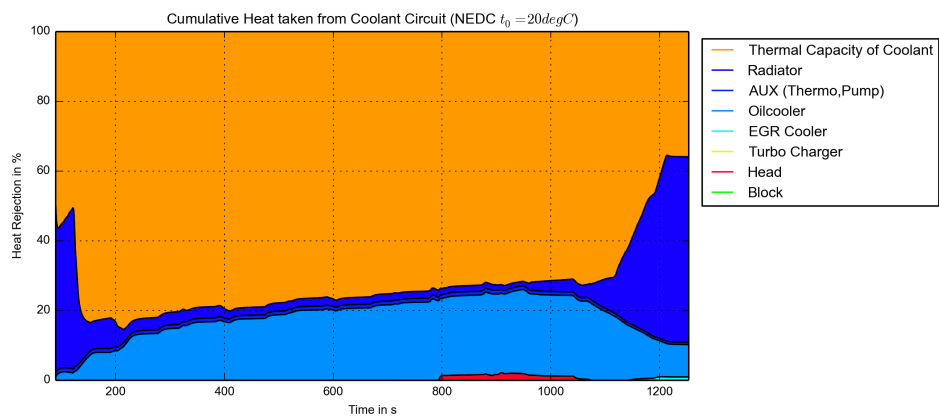


Figure 4.21: Cumulated heat taken from HT coolant circuit in percent (cold NEDC).

# 5 Conclusion and Prospects of Development

In this work it has been shown that it is possible to develop a real-time capable VTMS model using the software tools Cruise and BoostRT. Although many features like easy to use heat exchangers as well as thermostats have to be built from scratch, the basic feature-set of BoostRT, regarding cooling and lubrication, is sufficient to model and simulate a basic thermal network in order to complement a vehicle and engine model to a fully featured VTMS model. It might even be possible to get such a model on to a prototype ECU like explained in the section on the topic “Prospects” (section 5.2) in this chapter.

## 5.1 Summary of performed Work

After a general research and analysis of the state-of-the-art approach of arranging the thermal model of a VTMS simulation, the discretization of the thermal model was defined with respect to the mathematical models available in BoostRT and with the measurements in the background. The discretization and therefore the mechanisms used in the modelling process are described in chapter “Modeling of the VTMS - Thermal Circuit” (chapter 2) and more detailed in chapter “Details on Modeling” (chapter 2.4). These mechanisms and the corresponding elements available in BoostRT were used to setup a baseline thermal model.

The next step was the measurement validation and post-processing in order to calibrate the thermal model. The workflow of this process is documented in detail in chapter “Calibration of the Thermal Model of the Real-Time Capable

VTMS Model” (chapter 3). In order to increase the speed of postprocessing, some tools, written in Python, were developed to parse all result files of Cruise and BoostRT. This step not only increases the postprocessing speed but provides access to all result data at once. The calibration of the thermal model still needs some fine-tuning, but it was stopped in order not to exceed the time frame of this work as the model started revealing reliable and satisfying results. An excerpt of the result is displayed in the section “Results and Comparison” (chapter 4). The simulation results are compared to measurements and to those of the state-of-the-art approach containing temperatures, volumetric flow rates, heat fluxes, fuel consumption and calculation speed.

The final result is a thermal model featuring real-time capability. The lubrication circuit, the coolant circuits as well as the thermal masses behave as expected. The low temperature circuit is already prepared for connection to the hybrid system in the Cruise model. The hybrid system is still in calibration on the Cruise side of the simulation and not in the scope of this work. But due to the separation of the two circuits that circumstance has no influence on the other parts of the models, at least as far as the thermal model is concerned.

### 5.2 Prospects: Workflow to get the Model into an ECU and open Issues

There is almost no chance to get the fully featured vtms model of the thermal network on to a regular production type ECU. The memory, precision as well as the computational power of a conventional embedded controller used as ECU seems to be much too low. In general, programs on a controller, at least those with the claim to run in real-time, have just a fixed limited number of CPU-cycles during a timestep and have to stick to a fixed breakpoint to ensure a result at a given moment. This number of cycles can be increased by using higher powered computational resources, normally not found in vehicles.

In contrast to series production units, for prototyping purposes there are also Intel Atom or PowerPC powered controllers used. BoostRT runs already native on the first and at least can be compiled to run on PowerPC powered devices. But this approach has to prove its feasibility and reliability in the future but the low real-time factor of the simulation on the workstation give

good hope. Additionally concepts of proving the result of the simulation have to be developed in order to ensure the integrity of the results as well as their ability to reproduce the real process. The development of criteria to decide between the result of the simulation and a possible fallback strategy is also an open point.

There is another issue to get the simulation model to run on an ECU. Even if the simulation is reliable and fast enough there is still a problem of initialization. The simulation model needs the temperature of each element in order to calculate the right heat flux among each other. Therefore knowing the initial conditions is crucial to get a fast plausible result.

At least from the authors point of view, those aspects, the proof of reliability, the development of methods to ensure the right result, the initialization and the development of criteria to evaluate the results automatically are the dominating topics in the future to get a more sophisticated fully featured VTMS model into the ECU.

### 5.3 Benefits of this Real-Time Approach in VTMS

Even if it is not directly used in a vehicle, at least at the moment of writing this work, this real-time approach using AVL Cruise and BoostRT reduces the overhead of communication between applications and it provides much faster calculation times than the state-of-the-art approach. Additionally it opens a wide field of development of applications, some of this applications may even not require a fully featured model but only a sub-model of the VTMS simulation model:

**Control system implementation into ECU:** Instead of map based control strategies, a physical model may get implemented into the ECU in order to provide access to more sophisticated control strategies.

**Hardware in the loop applications:** The real-time capable VTMS model is able to provide a virtual environment, which is close to reality. This virtual boundary conditions can get used to develop hardware without the need of a fully equipped prototype vehicle.

**Virtual sensors for thermal management:** It also opens the possibility to reduce the sensor count in the vehicle and use estimated virtual ones to control cooling and lubrication system actuators. Several possible fields of application are, for example, at the combustion engine:

- Valve bridge virtual temperature estimator
- Piston head and cylinder wall temperature estimator
- Stagnant coolant temperature estimator

**Predictive Energy Management:** In combination with GPS data and an “electronic horizon” the VTMS may gain the capability to control its strategy dependent on the actual environment. For example it may alter its maximum temperature levels for the coolant and lubrication system in order to reduce fuel consumption dependent on the local vehicle velocity limits and expected loads (urban vs rural/highway area).

One more step towards this future developments is already done with the, just arrived first release of AVL CruiseM, M for multiphysics. AVL CruiseM is a software tool providing means for global vehicle simulations in one single package, by combining among others Cruise and BoostRT. Furthermore the modelling engine is planned to evolve and provide more specialized components and reduce the amount of time required to setup models. The post-processing will become united and therefore the effort of result evaluation and assessment will get reduced.

# Appendix

# Glossary

**BMEP** Brake Mean Effective Pressure

**ECU** Engine Control Unit, main controller in a vehicle, collects sensor data, calculates load condition and controls the engine and auxiliaries

**NEDC** New European Driving Cycle

**real time factor** This factor is the fraction of time needed to calculate a simulation in contrast to the time the same process needs to execute in reality

**SOT** start-to-open-temperature of the thermostat

**VTMS** Vehicle Thermal Management System, a simulation of all thermal processes in a vehicle, with the target to optimize energy consumption and emissions.

# Used Software

## Pre-processing of Input Data

- AVL Concerto (version 4.5B253)
- Python (version 2.7.3 and 2.7.6)
- Pylab/matplotlib (version 1.3.1)

## Simulation

- AVL Cruise (version 2011.2.0-310 build May 23 2012 14:08:44)
- AVL BoostRT (version 2011.2.0-241)
- Flowmaster (version 7.8.3)

## Post-processing of Results

- AVL Concerto (version 4.5)
- Python (version 2.7.3 and 2.7.6)
- Pylab/matplotlib (version 1.3.1)
- L<sup>A</sup>T<sub>E</sub>X- texlive (version 2013)

Additionally AVLs internal benchmark data, workflow databases and measurement data was used in the process of setup and calibration.



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- [2] AVL M. Muehloegger. *Boost RT Product Description*. 2010. URL: [https://www.avl.com/c/document\\_library/get\\_file?uuid=71957383-1be2-4a53-89e4-40b37ad489ff&groupId=297166](https://www.avl.com/c/document_library/get_file?uuid=71957383-1be2-4a53-89e4-40b37ad489ff&groupId=297166) (cit. on p. 9).
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- [5] Verein Deutscher Ingenieure VDI. *VDI Wärmeatlas CD-ROM*. 8. Auflage. Springer Verlag, 1997 (cit. on pp. 15, 25–28, 30, 31).

## Additionally used Literature

- Manual of AVL BoostRT 2011.2
- Manual of AVL Cruise 2011.2
- Manual of Mentor Graphics Flowmaster 7.8.3