CHALMERS
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Real Time Modeling of Engine Coolant Temperature

In Engine with Double Cooling Circuits at Two Temperature Levels

Master's thesis in Innovative and Sustainable Chemical Engineering

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Department of Applied Mechanics
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2017

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Abstract

An engine with double cooling circuits operating at two temperature levels has been developed at Volvo Cars. In this thesis the cooling circuit at the lower temperature level is studied and a model estimating the coolant temperature in this circuit has been developed in Simulink/TargetLink. The model is to be implemented in the engine control unit and used for function based diagnosis of the cooling system.

The cooling system consists of the following components: a water cooled air cooler which is used to cool the charged air entering the engine, turbochargers in two-stages with cooled bearings systems and a cooled compressor house in one of the turbochargers, an inlet throttle (ETM) cooled for component protection, an SCR injector also cooled for component protection, and a radiator used to cool the coolant. After an investigation of the average heat transfer rate from each component, the ETM was excluded from the model. The SCR injector was also excluded since this component had not yet been installed in the studied engine.

The model was formulated with a physical foundation, using energy balances of the system as well as experimentally obtained heat transfer relationships. An overall energy balance was used to calculate the coolant temperature in each discrete time step, based on heat transfer from the modeled components in the system (the water cooled air cooler, turbochargers and radiator). Model evaluation was performed using vehicle data obtained from real time measurements in a four cylinder diesel engine with extra measurement sensors installed in the air system and cooling system.

The developed model estimates the temperature with a total mean error of $-0.2\,°\mathrm{C}$. The 95 th and 5 th percentile for all simulated data is $1.4\,°\mathrm{C}$ respectively $-2.8\,°\mathrm{C}$. The model was also shown to be robust against input errors in a sensitivity analysis done for a representative test case.

Keywords: engine, cooling system, double cooling circuit, temperature, modeling, water cooled air cooler, turbocharger, radiator, Simulink, diagnosis.

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Caroline Palm, Gothenburg, January 2017

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1
Introduction

As a way of reducing environmental issues related to the transport sector the emission legislation is becoming more stringent. This pushes technical innovation in the vehicle industry to meet these requirements without compromising good engine performance (such as torque and acceleration). In a step towards meeting future demands Volvo Cars has developed an engine platform where one of the changes include the introduction of a second cooling circuit.

1.1 Background

In several parts of the world emissions from internal combustion engines are regulated by legislation. Regulated emissions include hydrocarbons (HC), carbon monoxide (CO), nitrogen oxides (NO_x) and particulates, the emission limits and conditions are regularly updated [1]. In the European Union emissions are regulated by so called Euro emission standards and the current standard for light duty vehicles is *Euro 6* [2]. As mentioned previously the emission requirements are continuously updated, and one upcoming change (phase-in starting September 2017) is that all emission tests are to be done under real driving conditions with a so called Real Driving Emissions test procedure (RDE) as apposed to previous laboratory testing [1, 2]. EU legislation has also introduced mandatory emission reduction targets for carbon dioxide (CO_2) of 130 g CO_2/km from 2015 and 95 g CO_2/km by 2021, which is directly related to fuel consumption [3, 4, 5]. These kinds of changes in legislation acts as a driving force that pushes and encourages technical solutions regarding internal combustion engines, with lower emissions at unchanged or increased engine performance.

Is has been shown that the engine cooling system has large influence on the emissions and that an optimized cooling system can, at a relatively low cost, yield a decrease in both primary pollutants and CO_2 [4, 5, 6]. One strategy to improve the cooling system performance is to divide the cooling system into two separate circuits at different temperature levels, with benefits such as reduced fuel-consumption, improved engine warm-up and improved air boosting [4, 5, 7]. This type of system can be illustrated as in figure 1.1 where a high temperature radiator provides cooling of the engine and other high temperature needs and the circuit operating at lower temperature ensures cooling of components with lower temperature needs. This strategy has been incorporated into Volvo Cars engine platform.

The above mentioned emission requirements should not only be fulfilled when the car is new. There is also legislation regarding detecting faults occurring in the car that can increase emissions, throughout the lifetime of the car – On-Board Diagnostics (OBD)

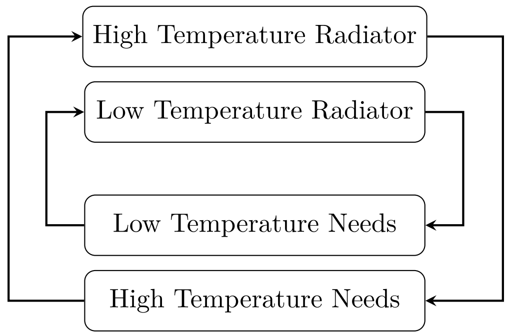


Figure 1.1: Concept of a double cooling circuit. The high temperature circuit ensures cooling of the engine, oil and other high temperature needs. The low temperature circuit provides cooling of the charged air, the turbocharger and other low temperature needs. Figure redrawn from [5].

regulations [1, 8]. Basically this legislation says that the driver must be notified if a fault or malfunction occurs that causes emissions to increase over some threshold values [1]. Diagnostic functions are therefore implemented into the engine control unit (ECU) and decide whether there is a fault in the system or not, and identify the fault. One diagnostic method is model based diagnosis, where faults are detected by comparing a measured sensor value to a predicted (modeled) value [8].

Regarding the coolant system, faults that can affect emissions are thermostat faults (leaking etc.) and sensor faults. These kinds of faults can be detected by using a model. And for Volvo Cars double circuit coolant system a model for the high temperature circuit is already available in the ECU, but a model for the low temperature circuit needs to be implemented.

1.2 Project Description and Aim

The aim of the thesis work is to develop a model in Simulink/TargetLink that estimates the coolant temperature in the low temperature cooling circuit. The temperature estimation should have a physical foundation and use information available in the ECU. The model should be composed in such a way that it can easily be modified to handle different setups of hardware that could be connected to the circuit. Ease of calibration for different car or engine variants should also be considered.

Model validation is to be done in Simulink using real world vehicle data obtained from testing in a vehicle equipped with extra measurement sensors. The finished model should then be able to be implemented into the ECU.

1.3 Limitations

The model is developed for a four cylinder diesel engine. Although the model should have the possibility to be modified for other engine variants and different hardware setups, it is outside the scope of the thesis to do so and only the diesel engine will be considered.

The method for model initialisation is not considered in this thesis. Instead the temperature sensor in the coolant circuit is used to initialize the model.

Due to the early stage of the engine development project, the engine control systems are not yet completely calibrated. For instance the control of the coolant flow in the circuit has not yet been calibrated and therefore the coolant flow is at its maximum level throughout the project. All validation measurements are made at maximum fan speed and completely open grille shutter, although these parameters are varied to find the radiator air flow rate. Once the control system is complete the model might need slight adjustments.

1.4 Thesis Outline

The thesis report is divided into six chapters. This introduction is followed by chapter 2, containing general theory of heat transfer, a description of the studied system and an introduction to engine control and diagnosis.

In chapter 3 the method used for developing the model is described. This includes the model development in Simulink/TargetLink, measurements in car and model validation.

The developed model is described in chapter 4. This chapter starts with an overview of the model, before going into detail about each subsystem in the model.

For the final model, the results is presented and discussed in chapter 5. The results contain a comparison of the estimated temperature and the actual sensor temperature in the circuit. This is then concluded in chapter 6.

2

Theory

The physical basis for describing the temperature change of the coolant is heat transfer from the components in the system. In this chapter basic heat transfer theory will be given, followed by a description of the studied system. At the end of the chapter, engine control and diagnosis will be introduced to give an overview of the intended model application – function based diagnosis.

2.1 Heat Transfer Theory

One of the fundamental laws of physics is the first law of thermodynamics. This law is based on conservation of energy, that energy can neither be created nor destroyed, and states that

> *If a system is carried through a cycle, the total heat added to the system from its surroundings is proportional to the work done by the system on its surroundings.* [9]

For a system with a certain control volume the first law of thermodynamics can, according to [9], be expressed as

$$
\begin{aligned}
&\left\{\begin{array}{l}\text{rate of addition} \\ \text{of heat to con-} \\ \text{trol volume from} \\ \text{its surroundings}\end{array}\right\} - \left\{\begin{array}{l}\text{rate of work done} \\ \text{by control volume} \\ \text{on its surround-} \\ \text{ings}\end{array}\right\} \\
&= \left\{\begin{array}{l}\text{rate of energy out} \\ \text{of control volume} \\ \text{due to fluid flow}\end{array}\right\} - \left\{\begin{array}{l}\text{rate of energy} \\ \text{into control vol-} \\ \text{ume due to fluid} \\ \text{flow}\end{array}\right\} + \left\{\begin{array}{l}\text{rate of accumu-} \\ \text{lation of energy} \\ \text{within control} \\ \text{volume}\end{array}\right\}.
\end{aligned}
\tag{2.1}
$$

This is the general energy balance for the system.

A device with primary purpose of transferring energy from a hot fluid to a cold fluid is known as a heat exchanger [9]. The basis for any heat exchanger is conservation of energy and the energy balance can be written for the individual streams in the heat exchanger, or the system boundaries can be defined around the entire heat exchanger. Defining the system boundaries around the entire heat exchanger, assuming no heat loss (adiabatic), the energy in the inlets must be equal to the energy in the outlets. This means that all energy lost in the hot fluid will be gained by the cold [10]. Defining Q as the rate of heat transfer from the hot fluid, this can be expressed as

$$Q = (\dot{m}C_P)_h(T_{h,in} - T_{h,out}) = (\dot{m}C_P)_c(T_{c,out} - T_{c,in}), \tag{2.2}$$

where $\dot{m}$ is the mass flow rate of the fluid, C_P is the heat capacity of the fluid and T_{in} and T_{out} is the temperature in respectively out of the heat exchanger. The index h and c denotes the hot respectively cold stream.

Heat transfer is usually divided into three different heat transfer mechanisms: convection, conduction and radiation. These types of heat transfer is explained further in the following sections.

2.1.1 Convection

The heat transfer between a surface and an adjacent fluid is categorized as convective heat transfer [9]. The rate for convection is expressed by the Newton rate equation:

$$\frac{Q}{A} = h\Delta T, \tag{2.3}$$

where Q is the rate of convective heat transfer, A is the surface area normal to the direction of the heat flow, ΔT is the difference in temperature between the surface and the fluid and h is the convective heat transfer coefficient [9]. Important parameters determining the heat transfer rate for convective heat transfer are the magnitude of the temperature difference (ΔT), the geometry of the system and properties of the fluid and the fluid flow [9].

2.1.2 Conduction

Heat transfer by conduction is primarily a molecular phenomenon, where the motion of a molecule at a higher energy level imparts energy to adjacent molecules at lower energy levels [9]. Fourier's first law of heat conduction for heat transfer in x-direction describes the heat transfer rate as

$$\frac{Q_x}{A} = -k\frac{dT}{dx}, \tag{2.4}$$

where Q_x is the conductive heat transfer rate, A is the area normal to the heat flow direction, $\frac{dT}{dx}$ is the temperature gradient in the x-direction and k is the conductivity which is a material specific property [9].

2.1.3 Radiation

Heat transfer by radiation is different from convection and conduction in that no medium is needed to transfer the heat [9].

For a perfect radiator (a black body that does not reflect nor transmit any thermal radiation) the thermal rate equation is used to describe the heat transfer rate:

$$\frac{Q}{A} = \sigma T^4 = E_b, \tag{2.5}$$

this equation is also known as the Stefan-Boltzmann equation [9]. $E_b = \frac{Q}{A}$ is the heat transfer rate per area of the emitting surface also known as the total emissive power of a black body, σ is the Stefan-Boltzmann constant and T is the surface temperature.

For actual surfaces the total emissive power will be less than that for a black body, since some thermal radiation will be reflected and transmitted. This is described with the emissivity, ϵ, defined as the ratio between the total emissive power of the surface and the total emissive power of a black body at the same temperature ($\epsilon = \frac{E}{E_b}$) [9].

The net rate of radiant energy transfer between two objects (1 and 2) at different temperature can be expressed as

$$Q_{1,2} = \sigma A_1 F_{1,2}(T_1^4 - T_2^4), \tag{2.6}$$

where $F_{1,2}$ is a function of the shape factors and emissivity of both objects [11]. For a configuration where one object (1) is completely surrounded by the other object (2) equation (2.6) can be written as

$$Q_{1,2} = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1}{\epsilon_1 A_1} + \frac{1-\epsilon_2}{\epsilon_2 A_2}}. \tag{2.7}$$

And when the enclosure is much larger than the enclosed object ($A_2 \gg A_1$), as for an object in a large, isothermal environment at temperature $T_2 = T_\infty$, the expression can be simplified to

$$Q_{1,2} = \epsilon\sigma A_1(T_1^4 - T_\infty^4), \tag{2.8}$$

which is useful for describing the heat transfer when a hot object is radiating to its environment [11].

2.2 System Description

The system studied is, as mentioned in section 1.3, a four cylinder diesel engine equipped with double cooling circuits. The cooling circuits differs slightly between different engine variants depending on if it is a diesel or a petrol engine, the engines can also have slightly different hardware configurations (e.g. one-stage or two-stage turbocharging). However, in this section only the diesel engine studied is described. First the air system for the engine is introduced, then the low temperature cooling circuit and its included components is described.

2.2.1 Diesel Engine Air System

The air system for the studied engine is shown in figure 2.1. Ambient air enters through the front grille and spoilers of the vehicle and passes heat exchangers placed at the front. The intake air then pass through an air filter that removes impurities. The air is compressed for improved efficiency in turbochargers in two stages. Compressing the air raises its temperature, and therefore the charged air passes a heat exchanger (charge air cooler) before reaching the engine inlet manifold. The air then enters the engine cylinders, fuel is injected and the combustion process takes place. The exhaust from the combustion passes through the turbine side of the two turbochargers before it reaches the aftertreatment system consisting of a combination of a lean NO_x trap (LNT) and selective catalytic reduction with a diesel particulate filter (SCRF). The exhaust gases are then emitted to the ambient via the exhaust pipe.

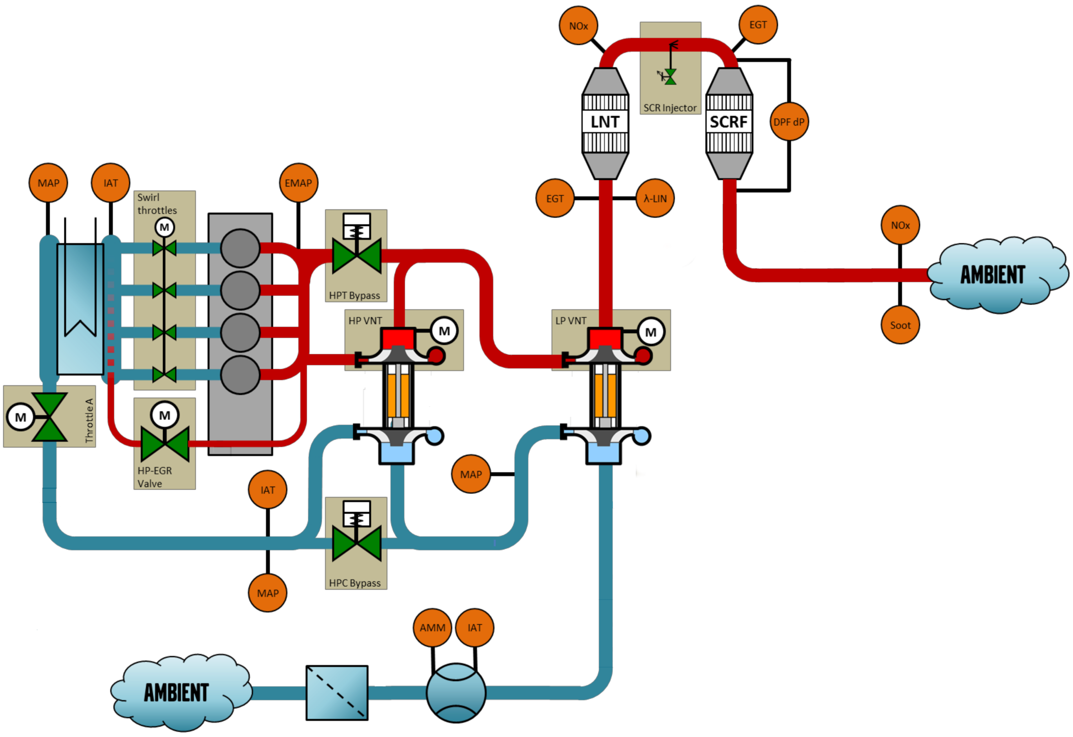


Figure 2.1: Schematic overview of the air system of the studied diesel engine. The blue lines represent air and the red represent exhaust gas. The orange circles in the figure are sensors with abbreviations: IAT = intake air temperature sensor, AMM = air mass meter, MAP = manifold pressure sensor, EMAP = exhaust manifold pressure sensor, EGT = exhaust gas temperature sensor, λ-LIN = linear lambda sensor, NO_x = nitrogen oxides sensor, DPF dP = differential pressure sensor for diesel particulate filter, EGR dP = differential pressure sensor for exhaust gas recirculation, Soot = soot sensor.

2.2.2 Low Temperature Cooling Circuit

As described in the introduction, the cooling of the engine system is divided into two circuits. One operating at a high temperature level, and one at a lower temperature level for cooling at lower temperature needs. The low temperature circuit (LT-circuit), which is the focus of this thesis, is responsible for cooling the charged air in the heat exchanger located before the inlet manifold, the turbochargers, the electronic throttle module at the inlet manifold (Throttle A in figure 2.1), and the SCR injector. A schematic drawing of the low temperature cooling system is shown in figure 2.2. The sensor in the figure, located before the thermostat (square), is the temperature sensor which the model is to mimic. The coolant flow is divided after the pump, where the main flow (thicker line) goes through the WCAC (water cooled air cooler) and the rest is divided between the ETM, SCR injector and turbocharger bearings. The LP turbocharger is cooled by two coolant streams: one cooling the turbocharger bearings and one cooling the compressor house in the turbocharger (further explained in 2.2.2.2). The thermostat regulates if the flow is passing through the radiator, or is bypassed (see section 2.2.2.1).

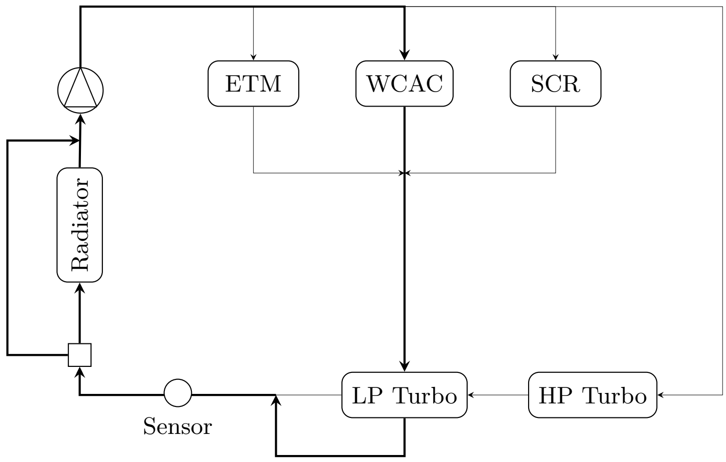


Figure 2.2: Schematic drawing of the LT coolant circuit. The sensor in the figure measure the temperature which the model is to mimic. ETM is the electronic throttle module, WCAC is the water cooled air cooler, SCR is the SCR injector, LP and HP turbo are the low-pressure respectively high-pressure turbochargers. The square is the thermostat. The thick line represents the main coolant flow, the thinner line is a smaller fraction of the flow.

Usually coolant is an aqueous solution of ethylene glycol (50 vol-%) [12]. In contrast to the HT-circuit which uses a pump driven by the engine crank, an electrical pump is used to pump the coolant in the LT-circuit. This makes it possible to circulate coolant in the system even after the engine had been shut off (known as after run), which can be used for component protection.

The components in the LT-circuit is explained in further detail below with focus on the occurring heat transfer.

2.2.2.1 Radiator

As the coolant flow through the components in the LT-circuit it will accumulate heat before it reaches the radiator where the coolant is cooled itself. The radiator is a type of heat exchanger located in the front of the vehicle (behind the grille) utilizing ambient air to cool the coolant flowing through its finned heat exchanger tubes [12].

The heat transfer between the coolant and air occurs by conduction through the tube walls and fins as well as by convection [12]. As with all heat transfer, the temperature difference between the coolant and the air is an important aspect determining the heat transfer rate in the radiator. Two other important aspects of the radiator cooling power is coolant flow and air flow, which in contrast to the temperature difference can be controlled.

In order to avoid overcooling of the coolant, keeping the coolant temperature optimal, the coolant flow is sometimes directed past the radiator [13]. This is controlled by a self-regulating wax thermostat which opening area is determined by the temperature of the coolant.

Generally the air flow through the radiator is determined by the vehicle speed. However, the air flow can be controlled by using radiator shutters (grille shutter and spoiler shutter) and a radiator fan. The radiator shutters are vanes that blocks air flow through the grille and spoilers, this is done both in order to control the temperature of the coolant but also to reduce drag (improve fuel economy) [13]. The radiator fan is placed behind the heat exchangers to regulate the air flow, this is particularly useful when idling since the "natural" air flow caused by driving then is close to zero.

2.2.2.2 Turbocharger

The air supply to the combustion chamber is an important aspect in engine control, since the air supply impacts engine performance as well as the emissions and fuel consumption [14]. One way to optimize the air supply is by using a turbocharger. A turbocharger utilize the hot exhaust gas flow in a turbine to generate power used to compress the intake air in a compressor directly coupled to the turbine by a shaft [15].

The process of compressing the intake air in order to supply the engine with more air is referred to as boosting, supercharging or pressure charging. The compression increase the density of the air, and hence the mass of air in the engine cylinder, which means that a proportional extra amount of fuel can be added yielding an increased power output or thus allows engine downsizing [14]. The turbocharger can be considered a form of waste heat recovery, since it utilizes the otherwise unused thermal energy in the exhaust gas and improve the efficiency of the engine [14].

The engine studied in the thesis has controlled two-stage turbocharging with two turbochargers in series with a controlled bypass on the compressor side (HPC bypass in figure 2.1) [16]. The first turbocharger (seen from the inlet air point of view in figure 2.1), the so called low-pressure (LP) turbocharger, is larger than the second, so called high-pressure (HP) turbocharger. At low engine speeds the HPC bypass valve is closed and the high-pressure turbocharger is used, resulting in rapid development of turbocharger pressure. Whereas at high engine speeds the bypass valve opens so that only the larger low-pressure turbocharger is operating.

The turbocharger is also controlled on the turbine side to prevent the turbocharger from being damaged or overloading the engine at high exhaust gas flows. This is achieved in two ways: with a wastegate valve which diverts the flow past the high-pressure turbine at high engine speeds, and with the turbines having a adjustable geometry (VNT, variable nozzle turbine) that controls the flow rate through the turbines [16, 17]. The VNT is also used to control the air compression. At low engine speeds, when the high-pressure turbocharger is used, the VNT of the LP turbocharger is closed, hence the LP compressor is not operating.

Exhaust gases from a diesel engine can reach temperatures up to $700-900\,°C$ and hence the turbocharger turbine is made of materials that can sustain such high temperatures [18, 19]. On the compressor side of the turbocharger temperatures are lower, ranging from $30-150\,°C$ [19]. This large temperature gradient across the turbocharger will yield a significant conduction heat transfer through the less heat tolerant bearing system supporting the shaft connecting the turbine to the compressor. This heat transfer through the bearings system may be problematic and is called heat soak-back [20]. While the engine is running oil lubricating the turbocharger shaft will absorb most of this energy thus preventing damage to the turbocharger bearings system. But once the engine is shut off the oil flow stops even though the bearings still needs cooling. Therefore water cooling of the turbocharger bearings has been introduced, since water can be pumped also after engine shut off by the electrical pump (after run) [20].

Both turbochargers in the system has cooled bearings systems, seen as the thinner lines through the turbochargers in figure 2.2. In the figure it can be seen that the low-pressure turbocharger also is cooled, by the major part of the coolant flow (thick line in the figure). This is to cool the LP turbocharger compressor house.

2.2.2.3 Water Cooled Air Cooler and Electronic Throttle Module

As mentioned above, turbocharging is a way of increasing the density of the intake air. However, the compression also increase the temperature of the air. And since density is coupled to the temperature (hot air is less dense than cool) a reduction in temperature can further increase the density of the charged air, hence more air can be supplied to the engine without increasing the pressure further [16, 21]. Too high temperatures can also cause too high combustion temperatures, which can affect power, torque and emissions [21]. Therefore, cooling of the charged air is desired.

A common way of achieving the charge air cooling is by using a heat exchanger usually referred to as a charge air cooler (CAC) [21]. Charge air coolers can either utilize air as cooling media (ACAC) or water coolant (WCAC), the latter being introduced in the engine platform studied in this thesis. Compared to the previously used ACAC, a WCAC has several advantages including smaller pressure drop in the charge air system and improved transient response since the WCAC can be placed in direct vicinity to the intake manifold whereas a ACAC needs to be located by the radiator and hence giving a longer air path [21, 22]. Since water has a higher heat capacity than air a smaller heat exchanger can be used for a WCAC than a ACAC, thus resulting in weight benefits [21, 22].

In direct vicinity to the WCAC is the electronic inlet throttle (electronic throttle module, ETM) a component that control the flow of air into the engine. To avoid overheating and damage to this component, it is cooled by coolant in the LT-circuit.

2.2.2.4 SCR injector

In diesel engines the most troublesome emissions are nitrogen oxides (NO_x) and particulates [23]. And as a measure towards reducing these emissions due to the regulations discussed in section 1.1 advanced aftertreatment methods are being implemented. In the studied engine the exhaust aftertreatment system includes a lean NO_x trap (LNT) and selective catalytic reduction (SCR) with an integrated diesel particulate filter (SCRF) (see figure 2.1).

In selective catalytic reduction of NO_x urea is injected to the exhaust mixture and decompose to ammonia which then reduce the NO_x to nitrogen gas N_2 [24]. Due to the high exhaust gas temperature (especially after LNT regeneration) the urea injector can become overheated. The injected urea cools the injector to some extent but not sufficiently, therefore the injector is cooled by water coolant in the LT-circuit.

2.3 Engine Control and Diagnostics

The finished model will be implemented in the software in the engine control unit (ECU). It will be used by diagnostic functions to detect faults in the low temperature cooling circuit. A brief introduction to the subject of diagnostics and the basic principle of the ECU will be given here.

2.3.1 The Engine Control Unit – ECU

Engine control is the core for over seeing and managing various engine functions [25]. The engine control system consists of the engine control unit (ECU), sensors, actuators and a communication system. And what was previously controlled mechanically has, as technology has evolved, transitioned to computerized control by the engine control unit [26].

The ECU consists of electronic components on a microcontroller chip (CPU), a memory to store information and software [26]. The ECU basically works in the same way as a conventional PC, where data is entered and used to calculate output signals [16].

In the ECU the entered data comes from sensors that measure physical quantities and converts it to an electric signal proportional to the measured value. The electric signal is processed in real-time in the microprocessor of the ECU and instructions to actuators is calculated. The calculations can be based on mathematical functions or using a set of pre-calculated results covering different engine operating conditions (maps or look-up tables). The calculated instructions are then sent to the actuators controlling different parts of the engine [26]. The basic principle of the engine control with the ECU is illustrated in figure 2.3.

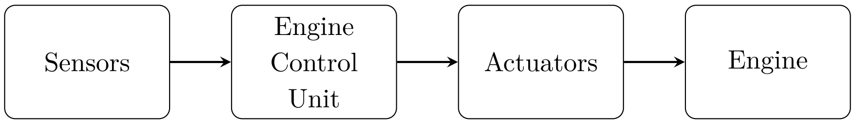


Figure 2.3: Basic principle of engine control.

2.3.2 Fault Diagnosis

As mentioned in section 1.1 there is legislation regarding supervision of components and functions that cause increased emissions if malfunctioning – the OBD regulations. This supervision is known as diagnosis and the diagnostic functions are part of the ECU software and detects and identify faults and generate fault codes stored in the ECU memory [8]. If a fault is detected an indication to the driver can be sent from the ECU and the generated fault code is stored and can later be read with specific diagnostic tools, facilitating repair of the car.

One useful method for diagnosis, the method related to this thesis, is model based diagnosis.

In model based diagnosis (schematically shown in figure 2.4) a model is formulated based on known inputs and the output from the model is then compared to the actual system output. Based on the residual between the system and model output a fault decision is made [8]. Model based diagnosis has the advantages of being a cost-effective method with high diagnosis performance (small faults can be detected and the detection time is relatively short) [8].

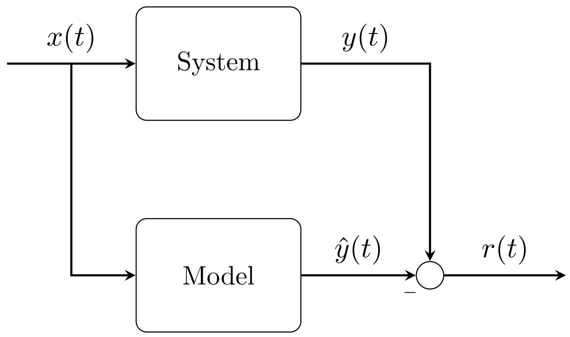


Figure 2.4: Model based diagnosis. Input to the system $x(t)$ gives output $y(t)$ and is used to calculate the model output $\hat{y}(t)$. The residual $r(t)$ is the basis for diagnosis. Redrawn from [8].

The model developed in this thesis will be used to detect faults on the temperature sensor measuring the temperature of the coolant in the circuit. One fault that can occur on the temperature sensor is a so called "stuck fault" meaning that the sensor returns the same value regardless of the actual state of the system. By comparing the modeled temperature to the measured sensor value and evaluating the error this type of fault can be found, according to the above described approach.

This approach can also be used if the thermostat is leaking. If the thermostat is stuck open the coolant temperature sensor will show a lower value than it should at those conditions, since more coolant is passed through the radiator. This is also detected by evaluating the accumulated error between the model and the sensor temperature.

Once a faulty coolant temperature sensor has been detected the model can further be used as a replacement value for the sensor, basing the requests to actuators on the modeled temperature instead of the sensor temperature. This way control functions that depends on the coolant temperature, e.g. coolant pump request, can still function properly.

3
Methods

The project procedure consists of three main parts: model formulation, data gathering and model evaluation. The model is constructed in Simulink/TargetLink and throughout the model development measurements in car are performed and used as input to the model and for model evaluation. To investigate the influence of uncertainties in the model inputs a sensitivity analysis is performed on the final model.

3.1 Model Formulation in Simulink/TargetLink

The model is formulated in Simulink 2011b with the dSpace TargetLink application. Simulink is a MathWorks block diagram simulation environment where the blocks perform mathematical operations on the system input [27]. TargetLink is a software integrated to Simulink that is used to be able to produce C-code directly from the Simulink model. TargetLink has a specific, extended blockset, with all necessary information needed for C-code generation [28]. The TargetLink blockset is used in the model.

The model is formulated in discrete time, with sampled input data from sensors (temperature, pressure etc.), actuator requests and modeled values in the ECU software. The sample time used in the model is 0.08 s.

3.2 Measurements with INCA

All input values to the model are obtained using measured data in real driving conditions. The model can then easily be evaluated by comparing the measured sensor temperature during driving with the estimated temperature at those conditions.

To record data from the ECU, engine and additional hardware (installed extra temperature and flow sensors) while driving the software INCA from ETAS Group is used. INCA is a measurement, calibration and diagnostics tool for automotive electronic systems [29]. INCA connects to an ETK (emulator test probe) which in turn is connected in parallel to the ECU, that way all data in the control unit memory is directly accessed [29]. INCA is also connected via CAN bus to access the data from the additional hardware installed (temperature and flow sensors).

In the INCA software, an experiment is created which defines all the relevant signals to be measured and recorded during driving. These recordings are then used as input to the Simulink model.

3.2.1 Data Collection

All measurements are performed in a Volvo XC90 with a four cylinder diesel engine equipped with the double circuit cooling system. In addition to the standard measurement sensors in the car, the test car is equipped with additional measurement sensors to simplify the model evaluation. The location of relevant extra measurement sensors installed are listed in appendix A.

The data is gathered for various driving conditions, and is used both to evaluate the model and also to gain a better understanding of how different driving conditions affect the cooling system and the model. Examples of different driving conditions are: steady state driving at constant speed, city driving, highway driving and mixed driving at varying vehicle speeds. The ambient temperature and other outer conditions vary between the tests and no standardized test cycles are tested.

3.3 Model Evaluation and Sensitivity Analysis

Using the obtained data from measuring in car, the model is continuously validated to improve and calibrate the model. The extra measurement sensors is used to decompose the validation further, evaluating each subsystem in the model and not just the final temperature. The extra sensors are also used to calibrate the model and find parameters and heat transfer properties.

For the final evaluation of the model the difference between the measured temperature and the estimated temperature is calculated for a number of different driving cases. For each case the maximum difference is also calculated. Using a data evaluation tool in MATLAB developed at Volvo Cars a statistical evaluation is performed on a large selection of data gathered through out the project (in total data from 7 hours of driving). The evaluation consists of a density plot showing the model error as a function of time for all data samples. It also consists of a histogram showing the model error normalized for all data points. The mean error, the median of the error and the 95 th and 5 th percentile of the error are also calculated and displayed in the histogram.

A sensitivity analysis of the model is performed for a representative test case, using a one-at-a-time approach [30]. In this approach one parameter value in the model is moved, keeping the others at their baseline values for that measurement. While doing this one parameter at a time the changes in the output is monitored and the resulting increase in the model error is calculated. This is then summarized in a bar diagram to give an indication of faults in the model or in the system that may yield large secondary errors.

4

Model Description

The model is constructed with one main model, calculating the temperature of the coolant each discrete time step. The calculations in the main model is based on heat transfer from the components in the circuit, modeled in subsystems.

Each part of the model has a physical foundation and the principles and equations used to develop the model is described is this chapter. First the main model will be described followed by each subsystem in the model.

The resulting Simulink models are shown in each section. For information about the blocks used in the model and the mathematical operations they perform, the reader is referred to Simulink documentation [31].

4.1 Main Model

The general energy balance in (2.1) can, with the coolant as control volume, be simplified according to

$$\left\{\begin{array}{l}\text{rate of addition}\\ \text{of heat to con-}\\ \text{trol volume from}\\ \text{its surroundings}\end{array}\right\} = \left\{\begin{array}{l}\text{rate of accumu-}\\ \text{lation of energy}\\ \text{within control}\\ \text{volume}\end{array}\right\}, \tag{4.1}$$

since the coolant system is a closed system with no heat flows in or out of the system, and the coolant performs no work. This means that all heat added to the coolant control volume must be accumulated. Equation 4.1 can be written, and discretized, as

$$\sum Q_i = \frac{d(\rho V C_P T)}{dt} = \rho V C_P \frac{\Delta T}{t_s}, \tag{4.2}$$

where $\sum Q_i$ is the total rate of heat addition to the coolant from each of the components in the circuit. ρV is the mass of coolant that the heat is transferred to, if the thermostat is closed the coolant mass in the radiator is not included. C_P is the heat capacity of the coolant, obtained from tabulated values in [32]. The temperature difference of the coolant in one time step (t_s) is denoted ΔT.

In the main model all heat transfer contributions from the components are summarized for each sample and by feeding back the coolant temperature from the previous time step the current temperature can be calculated using (4.2). The main model is shown in figure 4.1 below. To the left in the figure is all input signals to the model, available in the ECU software. The model inputs enters the subsystems where the heat transfer rates in the components are calculated. The mass of coolant in the system depends on whether the thermostat is open or not, if the thermostat is closed the coolant mass in the radiator is not accounted for. In the first time step the model needs to be initialized, this is currently done with a switch using the sensor value (seen in the top right of the figure). The initialization will be further discussed in section 5.4.

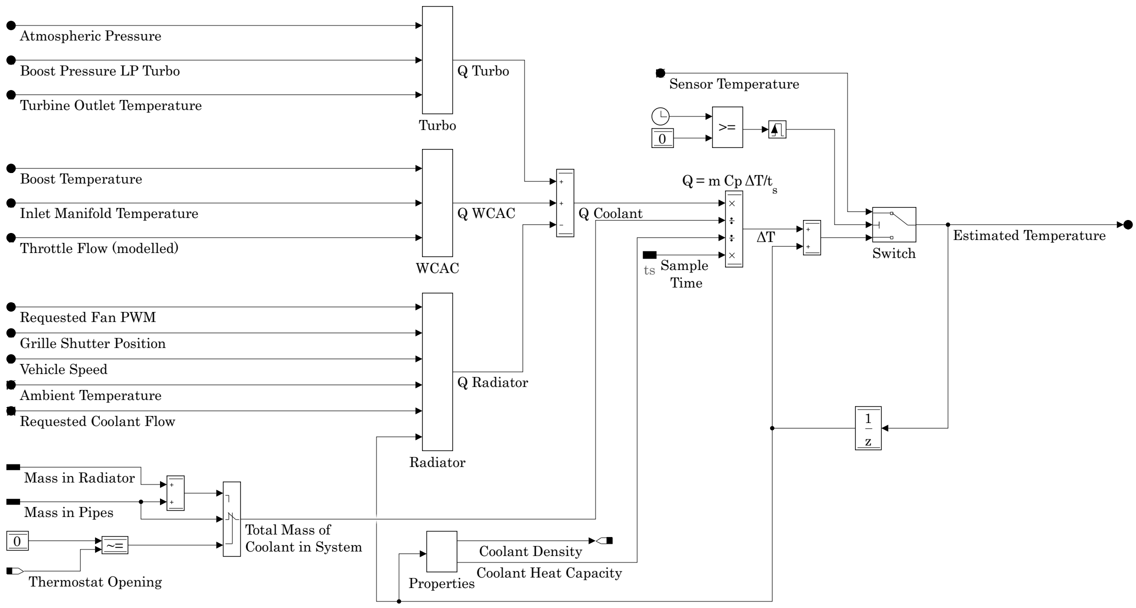


Figure 4.1: The main model calculating the coolant temperature from the summarized heat transfer from cooling of the turbocharger, the water cooled air cooler and the radiator.

The heat transfer contributions from each component is calculated in subsystems in the global Simulink model, in the following sections each of the subsystems in figure 4.1 will be described.

4.2 Radiator Subsystem

The supplier of the radiator provides data for the heat transfer performance of the radiator at different air and coolant flow rates measured at a constant temperature difference (45 °C) between the coolant and the air. This data is presented in figure 4.2.

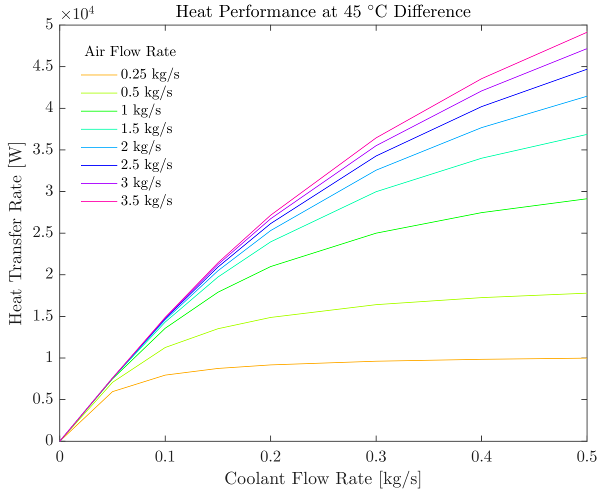


Figure 4.2: Supplier data of radiator heat transfer rate at coolant flow rates ranging from $0.1-0.5\,\mathrm{kg/s}$ and air flows ranging from $0.25-3.5\,\mathrm{kg/s}$ at a temperature difference between coolant and air of $45\,°\mathrm{C}$.

A linear relationship between the heat rate and the temperature difference is assumed and all data points in figure 4.2 are divided by the temperature difference to obtain a heat transfer rate expressed per unit Kelvin difference in temperature between the coolant and the ambient air, shown in figure 4.3.

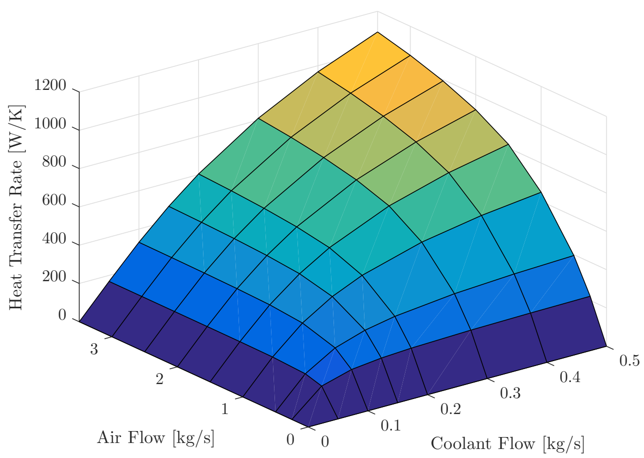


Figure 4.3: Heat transfer rate per Kelvin as function of air flow and coolant flow, obtained from supplier data in figure 4.2.

By knowing the coolant temperature calculated in the model (the previous time step) and the ambient temperature of the air flowing through the radiator the map in figure 4.3 can be used to model the heat transfer in the radiator. Before the heat transfer rate can be obtained, the flow rate of coolant and air through the radiator is needed.

As mentioned in section 2.2.2.1 the air flow through the radiator is dependent on the vehicle speed, the speed of the fan and the grille shutter position. Based on steady state measurements at velocities ranging from $0-200\,\mathrm{km/h}$ at varying fan speed and grille

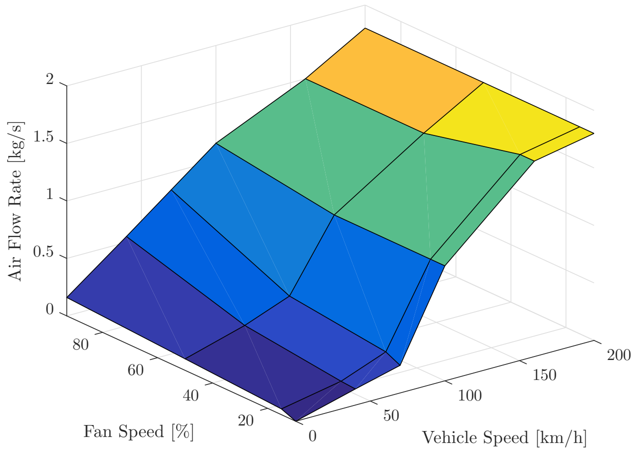


Figure 4.4: Map of the air flow through the radiator at varying vehicle speed and fan speed.

shutter positions the air flow through the radiator is estimated with help of the extra sensors installed. To avoid the complexity of having a map with three inputs (a map in 4 dimensions) one map is constructed for the vehicle speed and fan speed (seen in figure 4.4), and the grille shutter position is used to obtain a compensation factor, tabulated in table 4.1. Linear interpolation between the tabulated values are performed in the model.

Table 4.1: Compensation factor multiplied with the radiator air flow rate at different grille shutter positions. 0 % is when the grille is completely shut and 100 % is when the grille is completely open.

Grille Compensation	
Grille Shutter	Factor
0 %	0.4
50 %	0.9
100 %	1

In the control unit there is a signal for the total coolant flow request sent to the coolant pump. The requested coolant flow rate is converted from l/min to kg/s using the coolant density (obtained from tabulated values in [32]), and transformed into actual coolant flow rate using a look-up table with corresponding measured flow rates. However, as mentioned in section 2.2.2.1 coolant is sometimes bypassed the radiator and hence the total coolant flow can not always be used as a whole, the thermostat opening must also be taken into account. This is done by multiplying the total coolant flow rate by the thermostat opening factor, explained further in the following section (4.2.1).

The final radiator model is presented in figure 4.5.

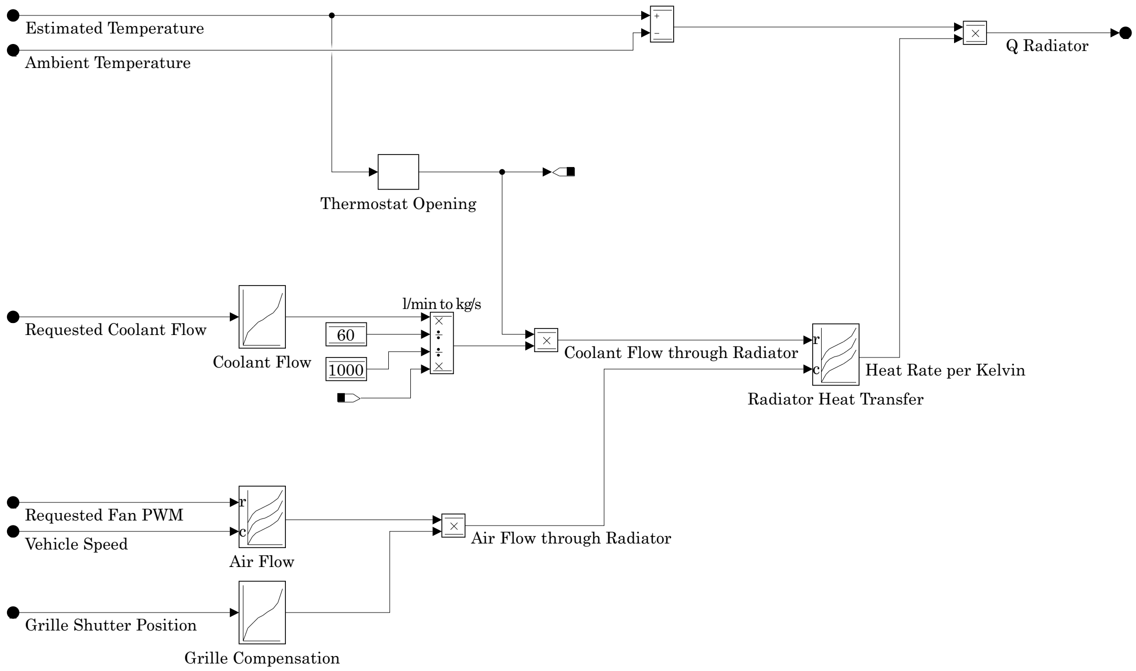


Figure 4.5: The radiator subsystem. The resulting heat transfer rate from the radiator is calculated using supplier characteristics of the radiator in a map with coolant flow rate (depending on thermostat opening) and air flow rate (as a function of vehicle speed, fan speed and grille shutter position) as input. The heat rate per Kelvin is multiplied with the temperature difference between the ambient air and the coolant temperature at the previous time step to yield the radiator heat transfer rate.

4.2.1 Thermostat

The thermostat is the component regulating the coolant flow through the radiator. It contains a wax that starts to expand when the coolant temperature warms up to a certain level, thus opening and starts letting coolant through the radiator. As the coolant temperature continues to increase the thermostat will continue to open, until a certain temperature has been reached and the thermostat is completely open. The thermostat has a hysteretic behaviour, the closing of the thermostat when temperature falls does not follow the opening of thermostat, it has a certain delay.

The thermostat opening and closing is in the model simplified to a linear relationship of the temperature according to

$$\begin{cases} x = \frac{T - T_{start\,open}}{T_{open} - T_{start\,open}}, & \text{if } T > T_0 \\ \\ x = \frac{T - T_{closed}}{T_{start\,close} - T_{closed}}, & \text{if } T < T_0 \end{cases} \tag{4.3}$$

where x is the opening factor. The hysteresis of the thermostat is such that if the temperature is increasing ($T > T_0$) the thermostat is following the opening curve and if the temperature instead is decreasing ($T < T_0$) the closing curve will be followed instead. In the transition between the opening curve and the closing curve the opening factor is kept constant.

To determine the thermostat behaviour, specifically the opening and closing temperatures in (4.3), the extra temperature sensors in the coolant is used to calculate the opening of the thermostat. Coolant temperature sensors are located around the radiator according to figure 4.6.

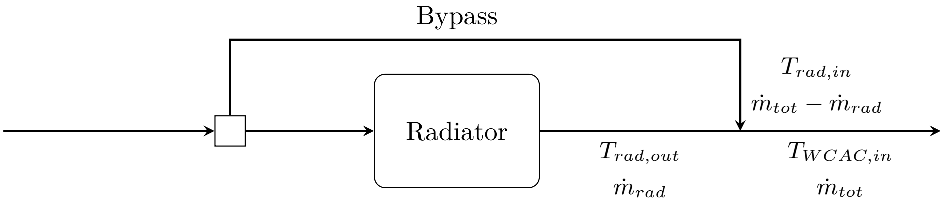


Figure 4.6: Schematic view of the radiator with extra temperature sensors $T_{rad,in}$, $T_{rad,out}$ and $T_{WCAC,in}$. The square symbolizes the thermostat determining the amount of coolant passing through the radiator. The total coolant flow is denoted $\dot{m}_{tot}$, $\dot{m}_{rad}$ is the mass flow of coolant passing through the radiator and $\dot{m}_{tot} - \dot{m}_{rad}$ will bypass the radiator.

To find the opening of the thermostat as a function of the measured temperatures an energy balance around the mixing point in figure 4.6 is used:

$$\dot{m}_{tot} C_P T_{WCAC,in} = (\dot{m}_{tot} - \dot{m}_{rad}) C_P T_{rad,in} + \dot{m}_{rad} C_P T_{rad,out}. \tag{4.4}$$

Assuming constant heat capacity in the temperature range and defining the thermostat opening as the ratio between the mass flow through the radiator and the total mass flow:

$$x = \frac{\dot{m}_{rad}}{\dot{m}_{tot}}, \tag{4.5}$$

equation (4.4) can be simplified and rearranged to obtain an expression for the thermostat opening calculated from the temperature sensors:

$$x = \frac{T_{WCAC,in} - T_{rad,in}}{T_{rad,out} - T_{rad,in}}. \tag{4.6}$$

This equation is used to find the opening and closing characteristics for the thermostat in an experiment where the coolant temperature is constantly increased by driving with constant acceleration. The calculated thermostat opening from (4.6) is plotted against the coolant sensor temperature (see figure 4.7) and the result is used to find a linear approximation of the thermostat characteristics, with $T_{start\,open} = 22\,°\mathrm{C}$, $T_{open} = 55\,°\mathrm{C}$, $T_{start\,close} = 48\,°\mathrm{C}$, $T_{closed} = 21\,°\mathrm{C}$.

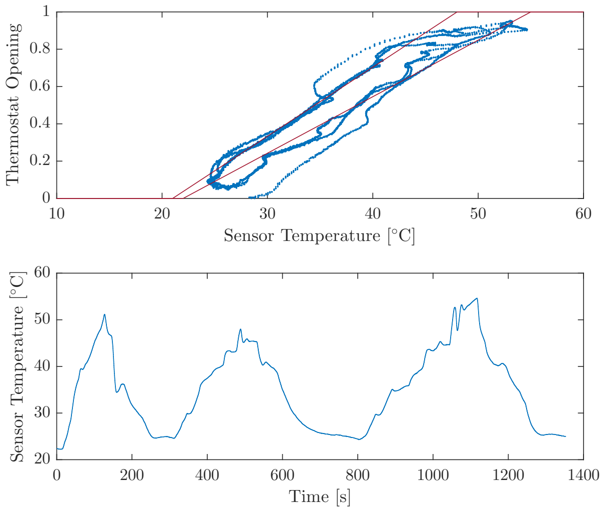


Figure 4.7: Calculated thermostat opening against sensor temperature obtained by varying the sensor temperature during constant acceleration according to the bottom figure. The red lines are the linear approximation.

The Simulink model for the thermostat hysteresis is shown in appendix B.

4.3 Water Cooled Air Cooler Subsystem

The model for the heat transfer from the water cooled air cooler is based on the equation for conservation of energy for heat exchangers (2.2), which when applied to the WCAC heat exchanger can be written as

$$Q_{WCAC} = (\dot{m}C_P)_{air}(T_{air,in} - T_{air,out}) = (\dot{m}C_P)_{coolant}(T_{coolant,out} - T_{coolant,in}), \tag{4.7}$$

where subscript *air* refers to the charged air being cooled in the heat exchanger, and *coolant* refers to the coolant.

To calculate the heat transfer rate in the WCAC (Q_{WCAC}) information of the air side of the heat exchanger is used. There are temperature sensors measuring the temperature of the air out of the compressor (referred to as boost temperature) and out of the WCAC (inlet manifold temperature), seen in figure 2.1. In the ECU software there is a modelled mass flow of air into the throttle, which is used in the WCAC model. The heat capacity of air is estimated at the average temperature of the air based on data from [9], assuming the heat capacity to be pressure independent. The resulting Simulink model is presented in figure 4.8.

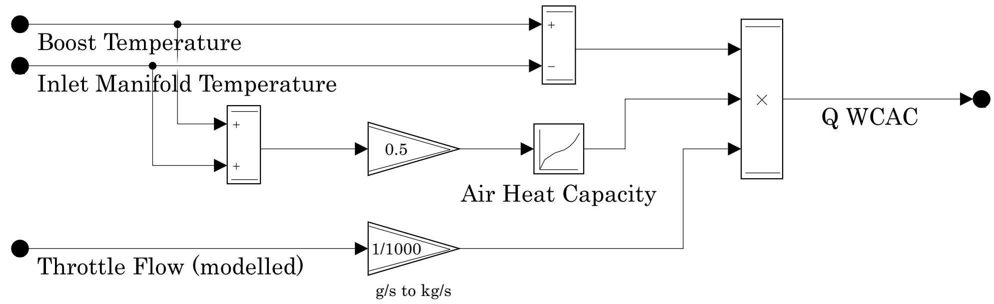


Figure 4.8: The WCAC subsystem. The resulting heat transfer rate from the WCAC is calculated based on air side information in the heat exchanger. Boost temperature is the temperature of the air entering the WCAC, inlet manifold temperature is the temperature of the air exiting the WCAC (entering the engine cylinders). Throttle flow is modelled within the car software.

4.4 Turbocharger Subsystem

To model the heat transfer rate to the coolant in the turbocharger (cooled compressor house and bearings system) two different approaches are tested. In the first approach an energy balance over the turbocharger in steady state is constructed. Then a method using lookup tables to find the separate heat transfer rates in the bearings system and the compressor house is used.

4.4.1 Energy Balance Approach

In an article by Aghaali et al. the main heat fluxes within a turbocharger is presented [19]. Experimentally and theoretically they derive heat transfer equations for each part of the turbocharger and quantify the heat transfer in each part.

With the purpose of finding the heat transferred to the coolant, the total energy balance for the turbocharger is used. This balance is derived assuming a control volume around the entire turbocharger. Fresh air enters the turbocharger compressor and is heated due to the compression. Exhaust gases passes through the turbine to produce power which is used by the compressor. The turbocharger is cooled by oil lubricating the bearings system as well as by coolant. Some heat will also be lost to the surroundings. Summarizing these energy flows over the control volume the resulting energy balance can be written as

$$\begin{aligned}(\dot{m}C_P)_T(T_{T,in} - T_{T,out}) + (\dot{m}C_P)_C(T_{C,in} - T_{C,out}) - \dot{Q}_{coolant} - \dot{Q}_{oil} - \dot{Q}_{ext,T} \\ - \dot{Q}_{ext,C} - \dot{Q}_{ext,B} = 0,\end{aligned} \tag{4.8}$$

where the first term represents the change of energy for the exhaust gases in the turbine, the second term represents the change of energy for the air in the compressor, $Q_{coolant}$ is the heat transferred to the coolant, Q_{oil} is the heat transferred to the lubricating oil and Q_{ext} is the external heat transfer rates in the turbine, bearings and compressor. To find the heat transfer rate to the coolant, the remaining terms in (4.8) needs to be calculated.

The change of energy for the air and exhaust can be obtained from temperature sensors and modelled flow rates in the ECU. Based on the results from Aghaali et al. the external heat transfer from the compressor house and the bearing housing were assumed to be negligible. The external heat transfer from the turbine is assumed to occur by radiation and convection and (2.8) and (2.3) are used to calculate these heat transfers. In the ECU, there is little information regarding the oil through the turbocharger bearings so a constant value estimated based on the results of Aghaali et al. is used initially.

Equation (4.8) is derived and evaluated for steady state conditions at constant engine speeds [19]. However, regular driving is not only steady state driving, but sometimes consists of transients where the turbine power is not equal to the compressor power (explained further in [33]). This makes the energy balance hard to use during non-steady driving. Furthermore difficulties finding properties needed for external heat transfer (such as the heat transfer coefficient in (2.3)) and the heat transfer rate to the lubricating oil, makes the model complicated to calibrate. Therefore, this turbocharger model was considered unsuitable for the intended purpose the will not be presented further in this thesis.

4.4.2 Lookup Table Approach

In the energy balance approach the total heat transfer rate to the coolant from the turbochargers is modeled. In the lookup table approach the heat transfer rates are instead separated, modeling the heat transfer from the turbocharger bearings system and the heat transfer in the compressor house separately.

As explained in section 2.2.2.2, the heat transfer in the bearings system is a result of the large temperature gradients between the turbine and the compressor. As a measure of how much heat will be transferred across the bearings system the exhaust gas temperature is used in the model. To find a relationship between the exhaust gas temperature and the energy transferred to the coolant in the bearings a density plot (figure 4.9) is constructed, using the extra sensors measuring the coolant temperature before and after the turbo bearings to calculate the heat transfer to the coolant. The density plot is constructed using all measured data for a range of exhaust gas temperature. The coloring in the plot represent the relative occurrence of data, where blue corresponds to few data points and red is the more frequent occurrence of data.

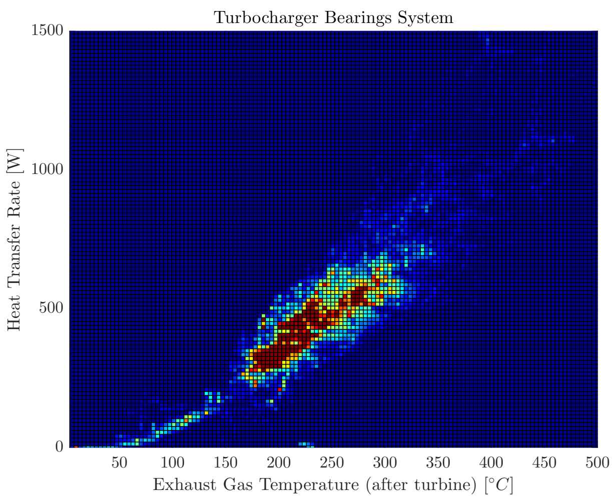


Figure 4.9: Density plot of calculated heat transfer rate in turbo bearings against the exhaust gas temperature measured after the turbines.

From the density plot a lookup table is constructed in Simulink using the exhaust gas temperature as input and extracting corresponding data points from figure 4.9 as output.

When the LP compressor is used the compressor house will heat up, as a result of the temperature increase arising when the air is compressed. The compressor house is cooled in the LT-circuit and a similar approach as for the bearings system is used to find the heat transfer rate to the coolant in the compressor house. There is a pressure sensor measuring the pressure of the compressed air (seen in figure 2.1). And the pressure difference between the atmospheric pressure of the air entering the compressor and the boosted air pressure is used as a measure of how much the LP compressor is used, assumed related to the amount of heat transferred to the coolant.

A density plot of the compressor house cooling power against the pressure difference for all measured data is shown in figure 4.10. To avoid the largest transients in the pressure difference the pressure is filtered using a low pass filter in Simulink.

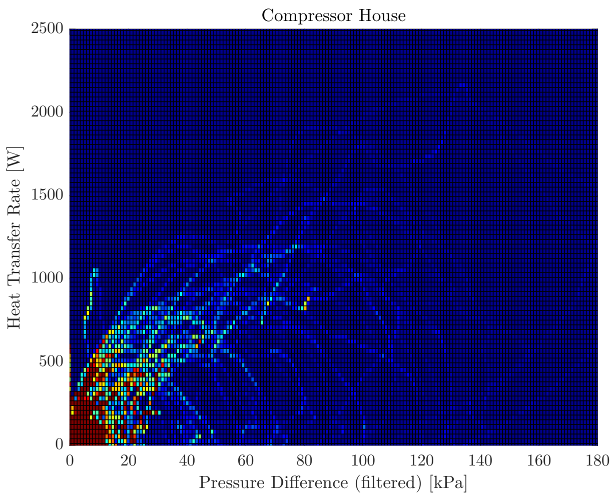


Figure 4.10: Density plot of calculated heat transfer rate in LP compressor house against the air pressure difference across the compressor.

The trend in figure 4.10 is not as strong as the trend describing the heat transfer in the turbocharger bearings system. Still, data from the plot is extracted and used in a lookup table for the compressor house. The final turbocharger model is shown in figure 4.11.

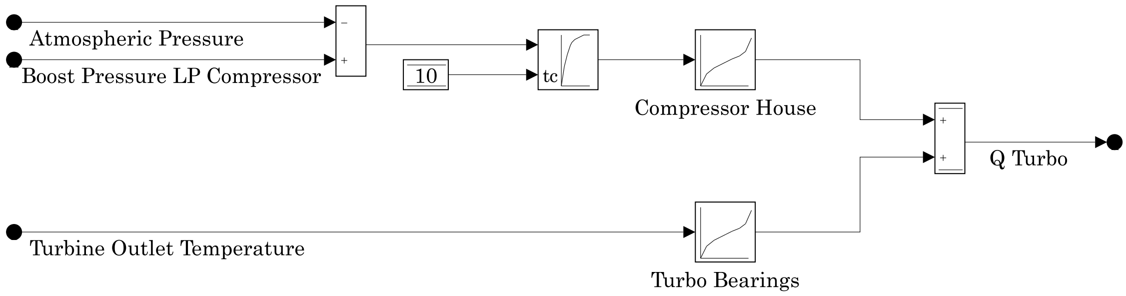


Figure 4.11: The turbocharger subsystem. A lookup table using the filtered difference in pressure across the LP compressor as input to obtain the compressor house heat transfer rate. The heat transfer rate from the turbo bearings is obtained in a lookup table with the exhaust gas temperature after the turbine as input.

4.5 Electronic Throttle Module

Due to the very small heat transfer contribution from the ETM on the total heat transfer (see figure 5.1 of the total amount of heat transferred in each component) the ETM was not included in the model. If the throttle where to be included, one simple approach might be to map the cooling power in the ETM based on the air temperature in the throttle, the boost temperature.

4.6 SCR injector

The hardware for urea injection is not yet implemented in the test vehicle, so this part is not included in the model. One method might be to model the cooling power based on the temperature sensor of the exhaust after the lean NO_x trap and the temperature of the urea. Another alternative might be to calculate the heat transfer rate based on conduction through the coolant and injector walls.

4.7 Calibration Procedure

The model is not entirely based on mathematical equations, and therefore some test has to be performed to find the lookup tables and other parameters used in the model. The calibration procedure for the model is summarized below.

1. Retrieve coolant properties (heat capacity and density). If 50 vol-% ethylene glycol is used, no changes should be needed.
2. Adjust the coolant mass in the system, based on coolant system specifications.
3. Use flow measurements to find the mapping of the requested coolant flow to the actual coolant flow rate.
4. WCAC:
    - Retrieve air heat capacity (assume pressure independent). Should remain unchanged.

5. Turbocharger: extra coolant temperature sensors around the turbochargers are required, as well as estimations of the flow fraction through the compressor house and the bearings system.
    - For a range of exhaust gas temperatures calculate the heat transfer to the coolant using the temperature sensors. Plot the heat transfer rate against the turbine exhaust temperature and extract values for a lookup table (as in figure 4.9).
    - Gather data when the LP compressor is used (particularly at high engine speeds) and calculate the heat transfer to the coolant using the temperature sensors. Plot the heat transfer rate against the LP pressure difference and extract data for a lookup table (as in figure 4.10).
6. Radiator:
    - Use radiator specifications from the supplier and create a mapping for the heat transfer rate per Kelvin as a function of coolant flow rate and air flow rate (as in figure 4.3).
    - Thermostat: requires extra coolant temperature sensors around the mixing point in figure 4.6. Perform a measurement with constant acceleration and plot the calculated thermostat opening against the coolant temperature. Make a linear approximation and to find the thermostat characteristics.
    - Perform steady state measurements at varying vehicle speeds at different radiator fan speeds and grille shutter settings and find the air flow rate required for the model to fit. Then use the information to map the air flow rate as a function of speed, fan speed and grille shutter position.

For model maintenance and adaption of the model for different engine variants and future engine changes, each component can be calibrated separately in the model subsystems. This is an advantage of the model structure, compared to if all components would be interconnected in the model.

5

Results and Discussion

In this chapter the results from the model simulations are presented and discussed. First an investigation of the heat transfer contributions from the different components in the circuit is presented in section 5.1. In section 5.2 the model is evaluated and discussed for three representative driving cases and a statistical analysis of the model error is given. The result from the performed sensitivity analysis is presented in section 5.3. The initialization of the model is discussed in section 5.4. Lastly a more general discussion regarding the calibration of the model and possible improvements is given in section 5.5 and 5.6.

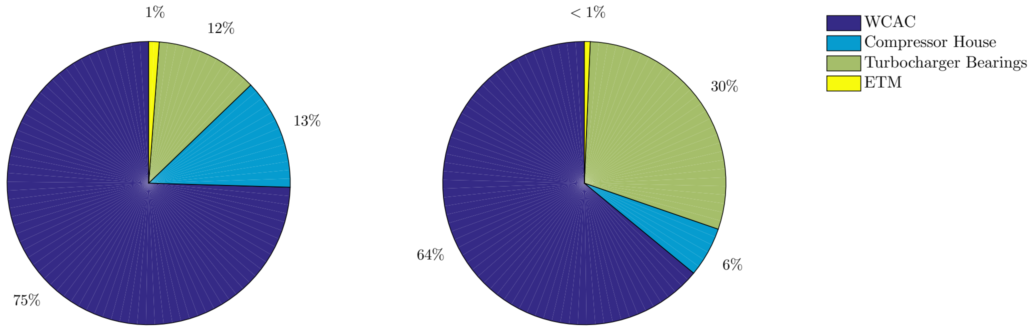
5.1 Cooling Circuit Heat Transfer Estimation

To get a sense of how the different components influence the temperature change, the extra temperature sensors installed was used to estimate the heat transfer contribution from each component in the circuit. The estimation is done using equation (2.2) for the coolant. The mass flow rate through each component is approximated based on the total flow rate in the circuit and measured data of the flow rate fraction in each component obtained from a Volvo Cars test report of the cooling system.

Two test cases are used for the heat transfer estimation and for each case the average heat transfer rate in each component is calculated and summarized in a pie chart in figure 5.1. The first calculation is done for a scenario where it is assumed that the heat developed in the components will be at its maximum – during maximum acceleration, seen in figure 5.1a. The second estimation is for a more mixed driving case (the same case used to illustrate mixed driving in the model validation in section 5.2.3), this result is shown in figure 5.1b.

It can be seen in the figure that the largest heat transfer contribution is from the water cooled air cooler. Based on these tests it could be concluded that the heat transfer from the electronic throttle module is insignificant (approximately 1 %) and it was therefore decided to exclude this component from the model. When the driving is more varied the LP compressor is sometimes not used therefore the heat transfer rate in this case is smaller compared to the heat transfer rate from driving with maximum acceleration. The heat transfer from the turbocharger bearings system becomes more important in less aggressive driving, mainly due to decreased heat transfer rate from other components while the heat transfer rate from the turbocharger has a more constant order of magnitude.

It should be noted that the radiator is not included in these figures. This is simply due to the fact that the radiator will have an counteracting heat transfer rate, approximately equal to the other heat transfer rates when the temperature is stable. Hence the radiator



(a) Heat transfer rate in each component for driving case during maximum acceleration.

(b) Heat transfer rate in each component for a mixed driving case (seen in section 5.2.3).

Figure 5.1: Calculated mean heat transfer rate in each component in the circuit calculated from the installed extra temperature sensors and estimated mass flow rate through the components.

heat transfer will always be important to include, otherwise the temperature would only increase.

5.2 Model Evaluation

Throughout the project measurement data has been gathered continuously to evaluate and develop the model. Here three different driving cases are used for the final evaluation of the model: highway driving, city driving and mixed driving. At the end of this section all measured data collected throughout the project is used to perform a statistical analysis of the model error.

The model error is defined as the difference between the actual sensor temperature and the estimated temperature. A positive error hence means that the model underestimates the temperature and a negative error means that the temperature is overestimated. For each test case the mean error is calculated. Since the error oscillate between negative and positive values the mean might be somewhat misleading, therefore the mean of the absolute value of the error is also calculated. The largest momentary error is also presented for each case.

The characteristics of each component becomes apparent in these evaluations. It can be seen that the heat transfer from the turbocharger is quite constant, compared to the WCAC which is characterized by transient peaks in the heat transfer as a result of vehicle accelerations. The radiator heat transfer follows the vehicle speed curve once the thermostat has been opened. The resulting temperature characteristics can also be distinguished. The coolant temperature also peaks during to vehicle accelerations, due to the heat transfer peaks from the WCAC. In more calm driving, such as city driving, the temperature is generally lower.

5.2.1 Highway Driving

Model evaluation for highway driving is done at vehicle speeds from 50 km/h to 70 km/h with a few stops or decelerations due to turns or red lights. The measurement is started with a cold engine and the warm-up period can be seen in the first 500 s of the measurement shown in the figure below. Thereafter the coolant temperature is kept quite constant, ranging between 20 − 25 °C. Figure 5.2 shows the temperature development of both the measured sensor temperature and the estimated temperature, the vehicle speed, the total heat transfer rate to the coolant and the heat transfer contribution from each of the components.

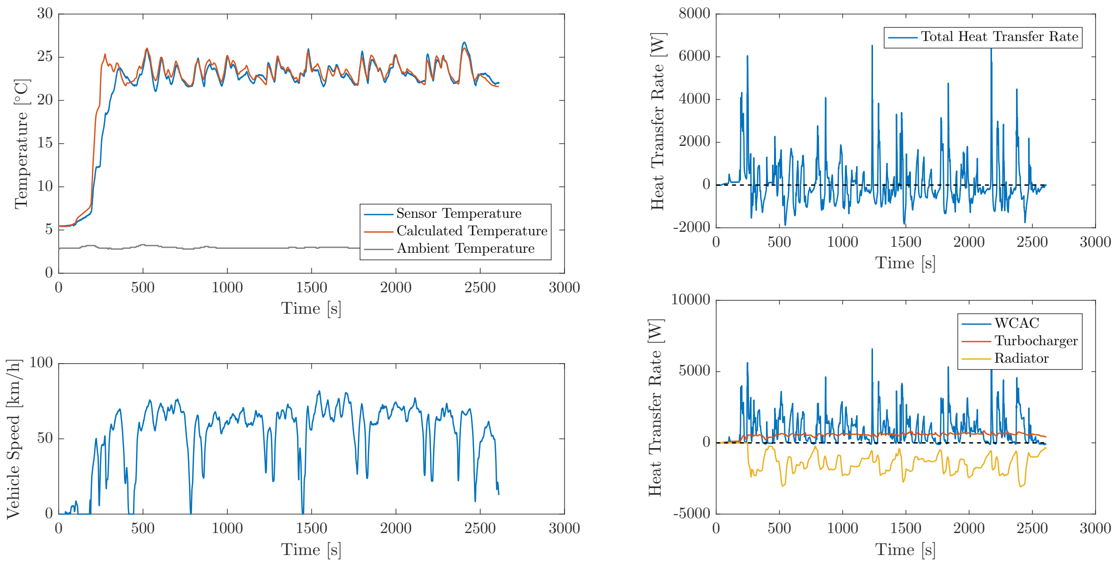


Figure 5.2: Results from simulation of highway driving. Top left: temperature comparison between actual sensor temperature and the estimated model temperature. Bottom left: vehicle speed. Top right: total heat transfer rate to the coolant. Bottom right: heat transfer contribution from each of the components in the model.

During the warm-up period it can be seen that the modeled temperature is rising too fast, giving rise to significant errors in this period (as large as −9.1 °C). Once the temperature is stabilized around 20 − 25 °C the error is decreased. The model error is presented in table 5.1 and it can be seen that the overall model error is relatively small (both the absolute value and the regular average) despite the large error in the beginning of the simulation.

Table 5.1: Resulting model error for highway driving, calculated as the difference between the measured sensor temperature and the estimated temperature.

Model Error	
Mean error	−0.5 °C
Mean absolute error	0.7 °C
Largest error	−9.1 °C

5.2.2 City Driving

When driving in the city, the vehicle speed is significantly slower compared to highway driving, ranging from 0 km/h to 50 km/h with several stops for pedestrians, red lights etc. The resulting calculated heat transfer rate for these driving conditions and a temperature comparison between the model and the sensor is shown in figure 5.3. In this case the engine was already warm when the measurement started, so no warm-up period of the coolant is seen here.

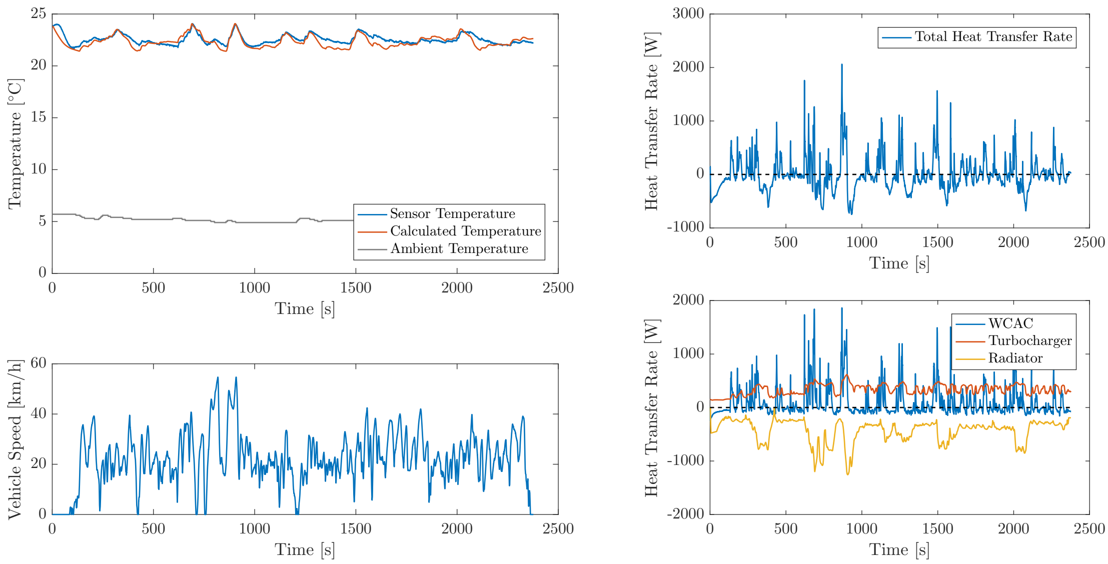


Figure 5.3: Results from simulation of city driving. Top left: temperature comparison between actual sensor temperature and the estimated model temperature. Bottom left: vehicle speed. Top right: total heat transfer rate to the coolant. Bottom right: heat transfer contribution from each of the components in the model.

It can be seen that the resulting heat transfer rate for these driving conditions is less, that in turn gives a slightly lower coolant temperature that oscillates less compared to the temperature in figure 5.2. Here the error is also somewhat smaller, with a maximum error of 1.2 °C. This could however be explained by the fact that no warm-up period (which is where the large errors originated from in the above case) is measured. In table 5.2 the model error for this test case is presented.

Table 5.2: Resulting model error for city driving, calculated as the difference between the measured sensor temperature and the estimated temperature.

Model Error	
Mean error	0.2 °C
Mean absolute error	0.3 °C
Largest error	1.2 °C

5.2.3 Mixed Driving

A larger span of vehicle speeds ($0 - 130$ km/h) with some hard accelerations are used to evaluate the model for a mixed driving case. This gives rise to a larger span of coolant temperatures compared to the more steady temperature profile at highway and city driving. The maximum heat transfer rate to the coolant is larger with peaks originating from the hard accelerations and giving rise to higher temperatures. This can be seen in figure 5.4.

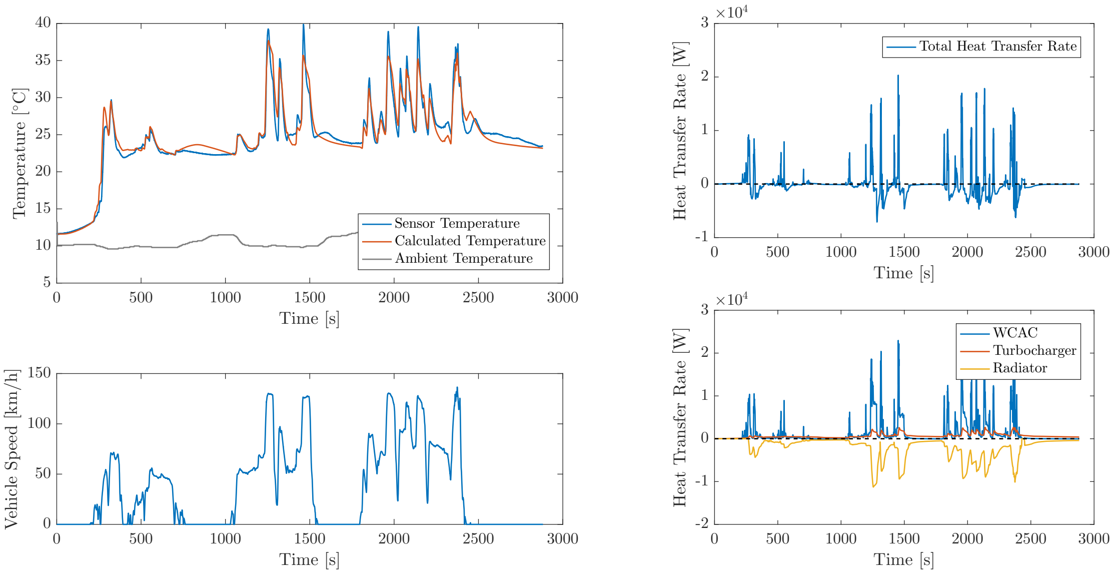


Figure 5.4: Results from simulation of mixed driving. Top left: temperature comparison between actual sensor temperature and the estimated model temperature. Bottom left: vehicle speed. Top right: total heat transfer rate to the coolant. Bottom right: heat transfer contribution from each of the components in the model.

Here the observed problem in the warm-up period is not seen. Instead the largest errors occur in the transient areas. The model error is tabulated in table 5.3 and overall it can be concluded that the model works well also for a more aggressive driving case. The mean error is for this case very small, but the absolute value of the error is somewhat larger.

Table 5.3: Resulting model error for mixed driving, calculated as the difference between the measured sensor temperature and the estimated temperature.

Model Error	
Mean error	$-0.1\,°C$
Mean absolute error	$0.8\,°C$
Largest error	$-5.4\,°C$

5.2.4 Statistical Analysis of the Model Error

Using measured data for a total of 7 hours of driving the model error is calculated and plotted against time in figure 5.5. To show the relative occurrence of an error value, the data is illustrated in a density plot. The color scaling ranges from blue to green to red, where red represents the most occurring error at that time and blue is least occurring.

It can be seen that the largest error occur during the warm-up period of the coolant in the beginning (as for the measurement of highway driving). The error then stabilizes within $\pm 5\,°C$ error, with some transient deviations. The mean error (indicated with the black line in the plot) is close to zero.

The model error is also compiled in a histogram normalized based on the number of

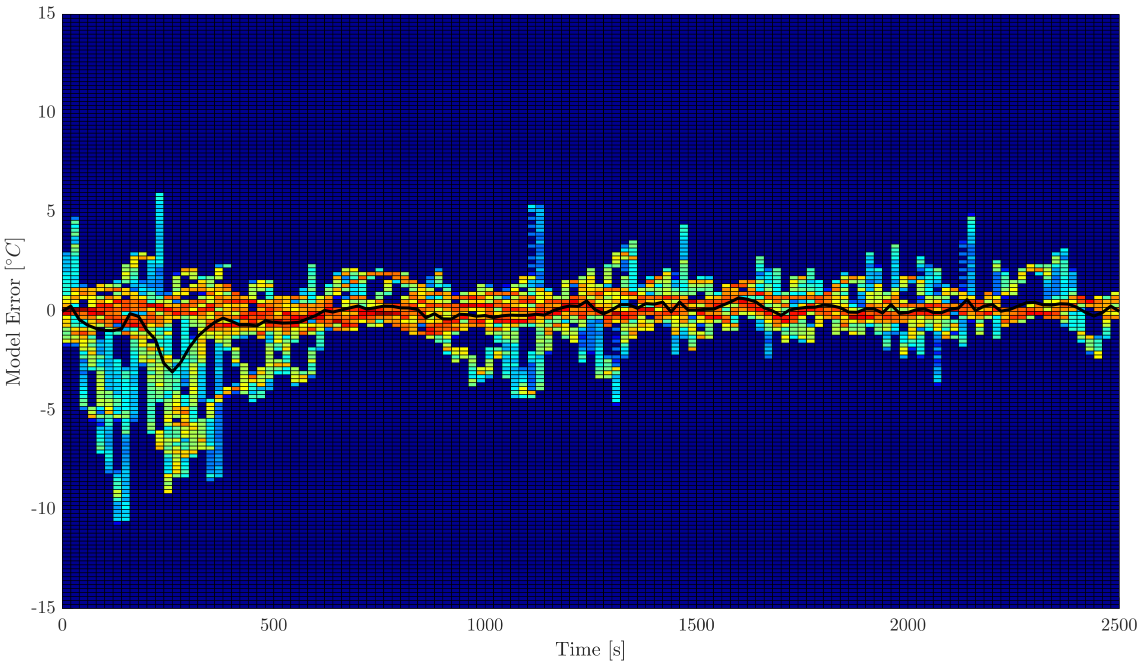


Figure 5.5: Density plot of the model error against time. The black line is the mean error. The color represents the number of samples (from blue to red) in each bin.

samples, in figure 5.6. In the histogram the mean and median of the model error are shown, as well as the 95 th and 5 th percentile of the error. 5 % of the data falls below $-2.8\,°\mathrm{C}$ error and 95 % of the data falls below $1.4\,°\mathrm{C}$ error. This means that 90 % of all data is within -2.8 and 1.4 error.

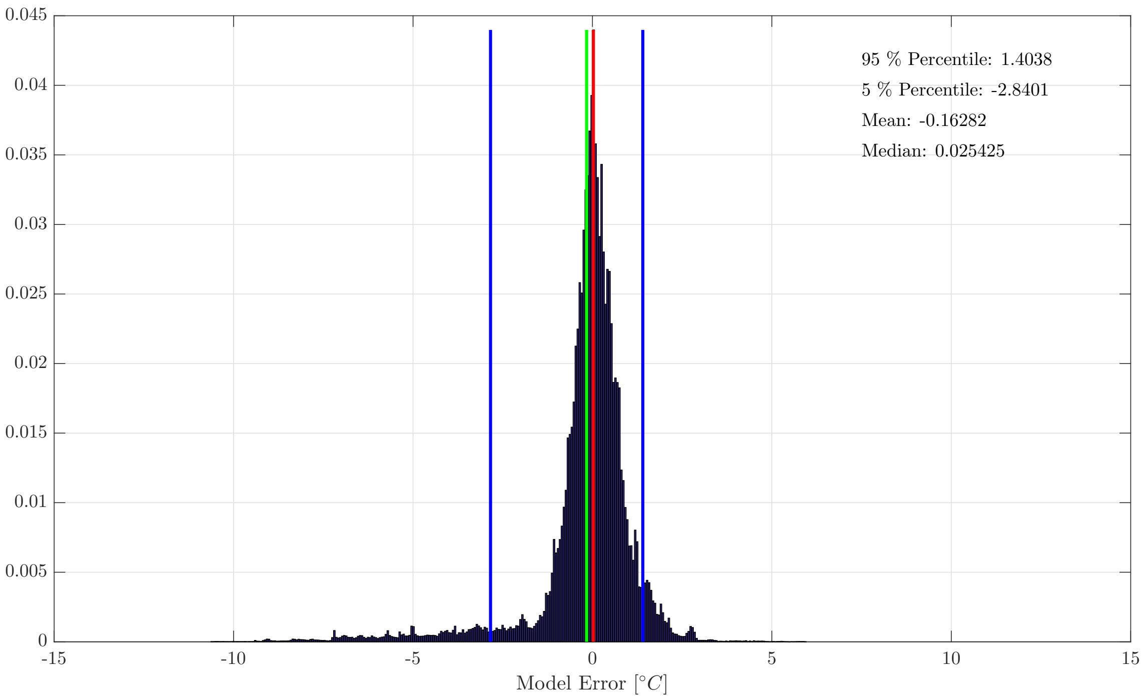


Figure 5.6: Normalized histogram of the model error for all measured data. The green line marks the mean error. The red line is the median. The blue lines are the 95 th and 5 th percentile of the error.

The transient temperature and heat transfer peaks in figure 5.4 arise during accelerations (which can be seen in the plot displaying the vehicle speed). The acceleration is coupled to the boost pressure, the pressure increase during the accelerations in order to supply the engine with more air. This in turn means that more and warmer air enters the WCAC, resulting in increased heat transfer to the coolant and hence the peaks in the temperature arise. In figure 5.4 larger errors in the transient areas was noted. To investigate this, the model error for the simulation data is displayed in a density plot against the air boost pressure in figure 5.7.

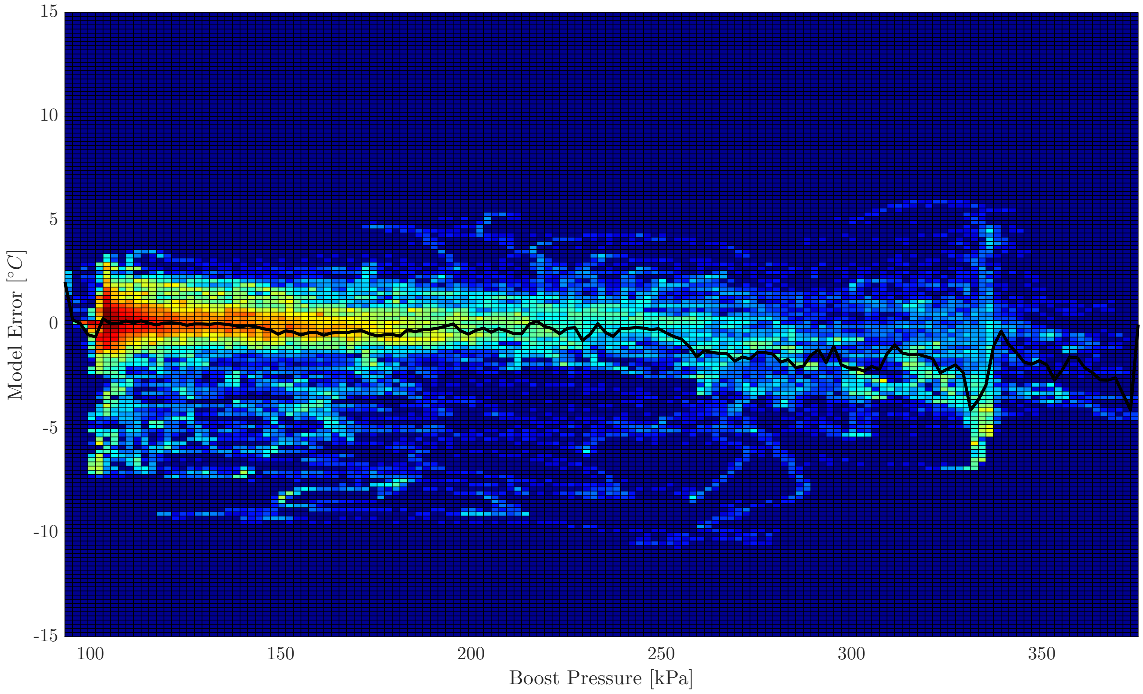


Figure 5.7: Density plot of the model error against boost pressure. The color represents the number of samples (from blue to red) in each bin.

It can be seen that most of the data is at low boost pressure, and as the boost pressure increase the error increase slightly. However, to be able to draw any conclusions more data at high boost pressure needs to be collected.

5.3 Sensitivity Analysis

In the sensitivity analysis each of the studied parameters are multiplied with an offset factor (ranging from 0.8 to 1.5) one parameter at the time and the resulting increase of the mean error is calculated. The test case for mixed driving (in section 5.2.3) is used for the analysis. The mean error for this test case without any offset is $-0.1\,°\mathrm{C}$, this is subtracted from each calculated error in the analysis to obtain a relative error increase.

The studied parameters are the model inputs. For the radiator air flow the input parameters (grille shutter position, vehicle speed and fan speed) are not varied separately, instead the total radiator air flow is multiplied with the offset factor. The radiator coolant flow is also varied as a whole and the offset could in reality originate from error in the total coolant flow rate, or from errors in the thermostat model.

The results from the sensitivity analysis are presented in figure 5.8. All in all it can be concluded that the model is robust against uncertainties in the inputs with a maximum error increase of $-2.5\,°C$ for a 1.5 offset of the boost pressure. This means that if a sensor or a model used as input has an offset, it will probably not give rise to large secondary errors. In addition there are diagnostic functions monitoring and indicating if an input sensor is malfunctioning or other parts of the system fail.

The model is sensitive to errors in the boost temperature, which is expected since this is an input to the WCAC model which has the largest heat transfer contribution (seen in figure 5.1 and figure 5.4). The inlet manifold temperature is less affected by the offset, most likely since this temperature generally is more stable compared to the boost temperature. The turbocharger pressure difference and turbocharger exhaust temperature also has impact on the error. Whereas the coolant mass seems to have very little influence on the error.

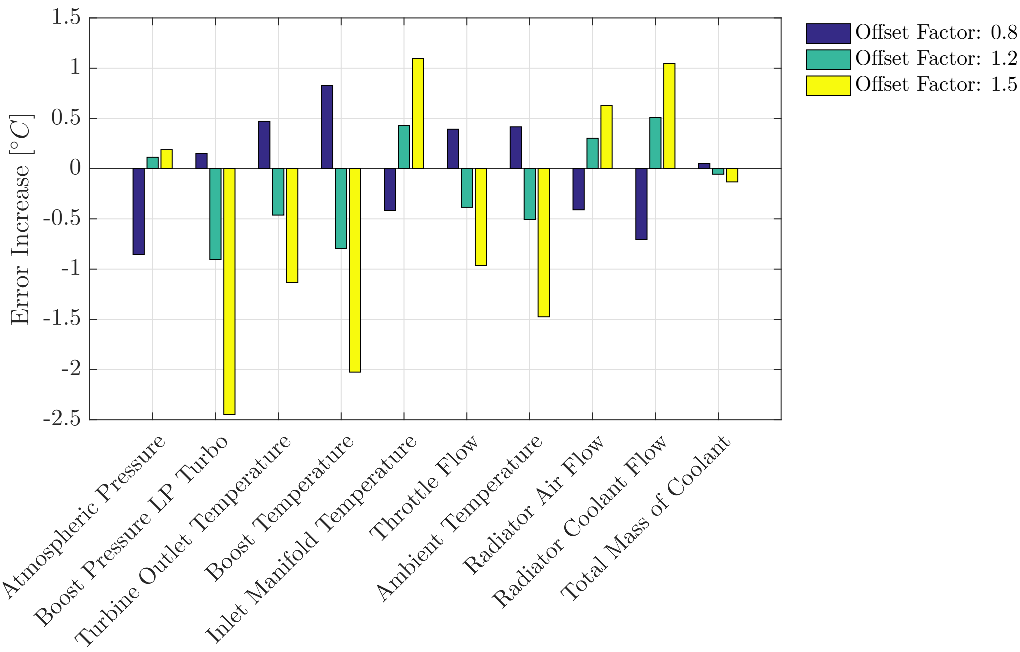


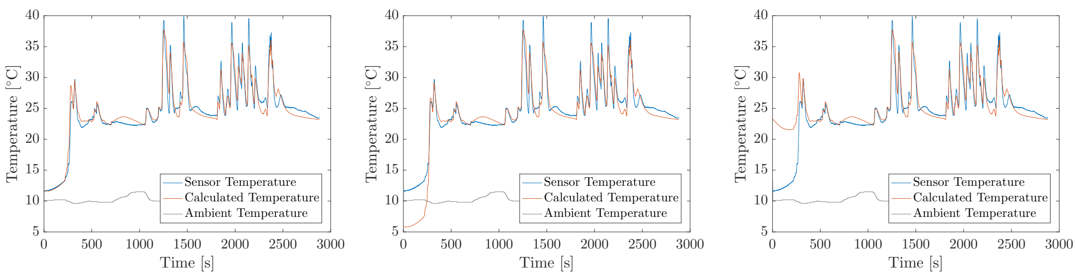
Figure 5.8: Resulting mean error increase from multiplying each parameter with a factor of 0.8, 1.2 and 1.5. Based on the mixed driving measurement in section 5.2.3.

This type of sensitivity analysis does not give insight to how a combination errors influence the model or the interactions between different parameters. It does however serve its purpose of giving an indication of possible secondary errors affecting the model. Another way of performing the sensitivity analysis would use an factorial design method [30]. In this method the simulation would be performed for all combinations of the variations, this way combinations of errors could also be seen. This method would however be very time consuming.

5.4 Model Initialization

To initialize the model for $t = 0$ the measured sensor temperature is currently used. This type of initialization is sufficient if the model is to be used for diagnosis of a ”stuck fault” of the sensor, or for detecting a leaking thermostat. These kinds of diagnostic functions assume a functioning sensor at the engine start up. If the sensor is already broken at start up this will be detected by another diagnostic monitor comparing all temperature sensor values and detecting if any sensor deviate significantly from the average temperature (if the engine has been shut off for a long time all temperatures should be approximately equal). If however the model is to be able to be used as a replacement value, the model must be accurate even if the sensor is broken from the start, and hence the model can not be initialized using the sensor value.

It is outside of the scope of this thesis to model an initialization independent of the sensor value. But the influence of inaccurate initialization is investigated for the test case of mixed driving, which can be seen in figure 5.9 below. In this test the model is initialized at half the actual sensor temperature, and at two times the actual sensor temperature. It can be seen that this will yield large errors at the beginning of the simulation, but the temperature will in time converge, and once it does the bad initialization will not affect the rest of the simulation. This implies that the initialization model might not need to be very accurate, since the temperature will converge in time, although the best results are obtained from good initialization.



(a) Model initialization at temperature sensor value.

(b) Model initialization at half the temperature sensor value.

(c) Model initialization at twice the temperature sensor value.

Figure 5.9: The effect of initialization shown for the mixed driving case in section 5.2.3.

It should be noted that the initialization in figure 5.9b is unphysical, since the coolant temperature can not go below the ambient temperature. However, the picture is just an illustration. In a proper initialization the initial temperature can be limited so that it is bound by the ambient temperature.

One possible method to model the initialization is to measure the cool down curve for the coolant (as a function of time) after the engine has been shut off and construct a mapping of the temperature against time. The engine off-time can then be used as input to obtain an initial temperature estimation. However, as mentioned above this coolant circuit can be active to cool down the components (particularly the turbocharger) after engine shut off – after run. The after run will cause the temperature to increase (there is no cooling power from the radiator) depending on how warm the different components are, before the coolant can begin cool down. The after run request should therefore also be included in the initialization model.

5.5 Calibration of the Model

Part of the project aim is to make sure the model is easily calibrated for different hardware setups and car models. By building the model based on the different components in the circuit the model becomes easy to configure for different hardware setups. This approach also makes it is easy to distinguish between the different heat transfer contributions from the components and find the characteristics of each component. This way if for instance only the radiator size differs between two car models, only the radiator subsystem will need to be recalibrated.

One drawback of the model calibration procedure is that extra measurement sensors needs to be installed to get an accurate calibration. For the calibration of the thermostat model it is possible that supplier information regarding the hysteresis might be sufficient to use. But for the turbocharger model coolant temperature sensors needs to be installed in order to estimate the heat transfer rate in these parts and create the lookup tables used for the heat transfer.

5.6 Possible Model Improvements

As can be seen in figure 5.5 there are problems using the model during the warm-up period of the coolant. The cause of these errors is not known and needs to be further investigated. This is of particular importance to be able to use the model to diagnose a leaking thermostat. A thermostat leakage is characterized by a slower warm-up of the coolant, thus it is important that the model is accurate in this period to avoid miss-detection.

The turbocharger model is a possible area of improvement for better model accuracy. It was seen in figure 4.10 that the use of the pressure difference across the LP compressor to describe the heat transfer from the compressor house gave quite a large spread of the data points and not a very clear trend. More data from when the LP compressor is used needs to be collected to further evaluate if this relationship adequate. Other parameters to describe the heat transfer should also be investigated. If possible, it would also be good to investigate options regarding the calibration of the turbocharger model, it would be optimal if the model could be calibrated without the use of extra temperature sensors.

The recirculation of exhaust gases (EGR), seen in figure 2.1, was not activated during the model development. It is possible that the activation of the EGR will affect the temperature measurement of the air after the WCAC, and hence affect the calculated heat transfer rate in the WCAC. If so is the case, the amount of exhaust gas recirculation (and the exhaust gas temperature) will need to be taken into account in the WCAC model.

The introduction of so called long-route exhaust gas recirculation, where exhausts are recirculated from a point after the aftertreatment system to the compressor intake, is a possibility for future engine development [34]. It is possible that this activation of long-route exhaust gas recirculation will affect the heat transfer in the system. Perhaps this effect will need to be accounted for in the turbocharger model, since the temperature of the air in the compressor inlet will be increased. The pressure after the valve regulating the EGR might not be equal to the atmospheric pressure which is currently used as model input, but judging from the sensitivity analysis is figure 5.8 error in the input pressure does not seem to have a very large impact on the total model error.

The development of the model was limited by the development of the full functionality of the coolant control system. Once the control of the coolant system is in place, the model will need further calibration. For instance the mapping of the coolant flow rate needs to be updated. The model should also be evaluated further for the radiator air flow rate with properly controlled fan speeds and grille shutter positions, since the evaluation was performed at maximum fan speed and grille shutter positions except for the steady state measurements where the settings were forced. Looking at the values for the grille compensation factor in table 4.1, the obtained values seem quite high with 90 % of the air passing through the radiator if the grille shutter is open halfway. These values needs to be further investigated.

In the model, the air entering the radiator is assumed to be at ambient temperature. This might contribute to possible model errors when the high temperature radiator, cooling down the coolant in the high temperature circuit is used. The high temperature radiator is installed in front of the low temperature radiator. This means that the air will be heated before entering the low temperature radiator and the actual heat transfer rate in the radiator will be less than the heat transfer rate would be if the air entered at ambient temperature. Looking at the sensitivity analysis in figure 5.8 errors in the inlet temperature to the radiator might give rise to errors in the model, depending on how much the air temperature is increased in the high temperature radiator. If this is found to be a problem, one way might be to couple the model for the high temperature circuit with this model, and use the heat transfer in the high temperature radiator to calculate the actual air inlet temperature to the low temperature radiator.

Once the cooled SCR-injector has been implemented into the system an investigation of its influence on the heat transfer needs to be done. If it turns out that the heat transfer contribution from this component is large, it needs to be included in the model.

The model evaluation was based on a relatively small amount of data (7 h of driving). To increase the reliability of the statistical analysis more data needs to be collected, and for more variations of driving conditions (high boost pressure etc.). The evaluations should also be done for a wider range of ambient conditions, including both hot climate and cold.

6 Conclusion

The goal of the thesis work was to develop a model that estimates the coolant temperature in the low temperature cooling circuit in a Volvo Cars diesel engine. The model is based on heat transfer from three components in the circuit: the water cooled air cooler, the turbochargers and the low temperature radiator. This model structure enables the model to be configured for different hardware setups. The model calibration is relatively easy, with few tests needed. The calibration of the model does however require extra installed temperature sensors in the coolant, which is a downside of the model not being completely based on mathematical equations.

The developed model is shown to be a good prediction of the coolant temperature. Measured data from a range of different vehicle speeds and conditions is used to evaluate the model. For all measurements performed the total mean difference between the measured temperature and the estimated temperature is calculated to $-0.2\,^{\circ}\mathrm{C}$. $90\,\%$ of all the data lies within $-2.8\,^{\circ}\mathrm{C}$ and $1.4\,^{\circ}\mathrm{C}$ error.

In the performed one-at-a-time sensitivity analysis the model is shown to have low sensitivity to input variable errors. The largest error increase observed in the sensitivity analysis is when the boost pressure is multiplied with an offset factor of 1.5, resulting in $-2.5\,^{\circ}\mathrm{C}$ relative mean error increase. The initialization of the model is studied, and it is found that if the model is initialized badly the estimated temperature will in time converge without it affecting the rest of the simulation.

While evaluating the model, a problem in the warm-up period of the coolant was discovered. In this period the model errors are significantly larger. The cause of these errors needs to be further investigated before the model is reliable to use in diagnosis applications. And before the model can be implemented into the ECU, additional model evaluations are needed for a fully functioning coolant control system.

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A

Installed Extra Sensors

Table A.1: Relevant extra measurement sensors installed in the test vehicle.

Air/Exhaust System	
Type	Placement
Temperature	High Pressure Compressor, Before
Temperature	High Pressure Compressor, After
Temperature	Low Pressure Compressor, Before
Temperature	Low Pressure Compressor, After
Temperature	Water Cooled Air Cooler , Before
Temperature	Inlet Manifold, Cylinder no. 1-4
Temperature	Engine Exhaust, Cylinder no. 1-4
Temperature	Exhaust Manifold
Temperature	High Pressure Turbine, Before
Temperature	High Pressure Turbine, After
Temperature	Low Pressure Turbine, Before
Temperature	Low Pressure Turbine, After
Coolant System	
Type	Placement
Temperature	Low Pressure Compressor House, In
Temperature	Low Pressure Compressor House, Out
Temperature	High Pressure Turbo Bearings, In
Temperature	Low Pressure Turbo Bearings, Out
Temperature	Inlet Manifold Throttle (ETM) , In
Temperature	Inlet Manifold Throttle (ETM), Out
Temperature	Water Cooled Air Cooler, In
Temperature	Water Cooled Air Cooler, Out
Temperature	Low Temperature Radiator, In
Temperature	Low Temperature Radiator, Out
Mass Flow	Water Cooled Air Cooler, In

B

Thermostat Model

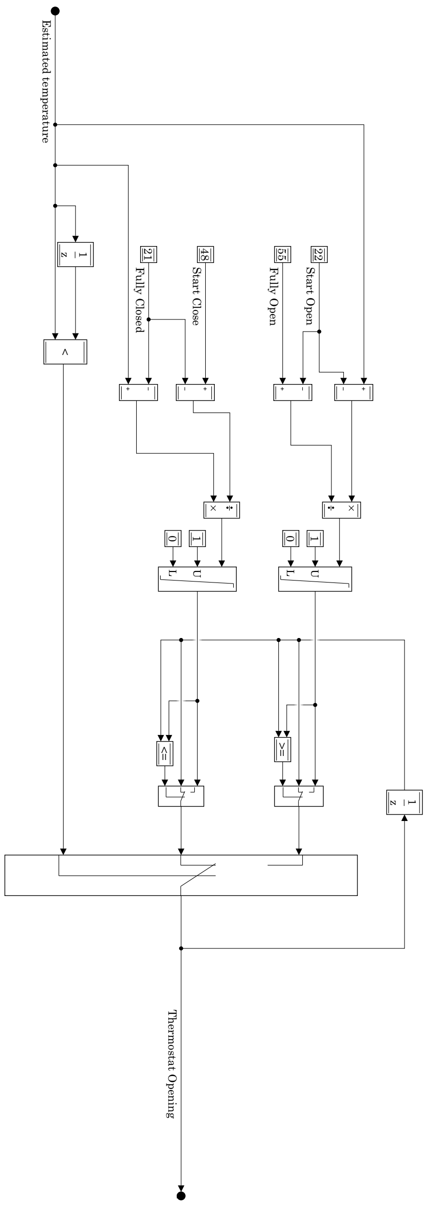

Figure B.1: Thermostat model.