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Modelisation of the engine coolant warming-up behavior

Master's Thesis in Automotive Engineering

ANDRÉ HAURY
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Department of Applied Mechanics
CHALMERS UNIVERSITY OF TECHNOLOGY
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Abstract

This master thesis report describes the development of a physically based model which can predict the coolant temperature of an internal combustion engine during its warm-up phase. The model is used to detect if the thermostat in the cooling system is functioning correctly which is a requirement for present day On-Board Diagnostics II (OBD II). The model uses information available within the Engine Control Unit (ECU) and needs little calibration work. Thereby, the model can easily be extended with extra features to remain applicable with future power train applications. The temperature prediction capability has been tested for steady state conditions as well as for "real-world" driving and regulated driving cycles, thereby the model has been tested on a vehicle with a start-stop system. The model includes a diagnostic function which is performed at the most accurate prediction of the model. The sensitivity of the temperature prediction to sensor input errors and varying parameters (like fuel quality) has been investigated in a sensitivity study. Finally, a threshold value is proposed for the diagnostic function by determining when an invalid faulty thermostat is detected or a faulty thermostat is not detected.

Keywords: Engine, cooling system, thermostat, physically basedmodel, simulink, OBD, coolant temperature, warm-up, diagnostic, temperature prediction

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Chapter 1

Introduction

1.1 Background

In a water cooled engine, the coolant temperature is regulated by a thermostat. The thermostat is usually mounted between the engine and the radiator input (figure 1.1). When the engine is cold the thermostat is closed and the coolant is directed directly from the engine coolant output to the input, bypassing the radiator. This allows the engine to heat up as quickly as possible. When the engine reaches its desired operating temperature, the thermostat opens and the coolant temperature is kept constant by allowing more or less coolant to flow through the radiator.

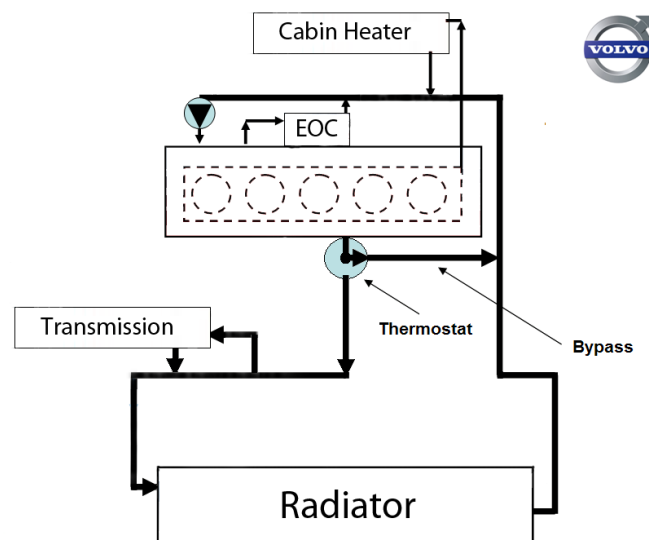


Figure 1.1: The Cooling system (Source Volvo PV)

In case a thermostat is leaking, there will be an undesired coolant flow through the radiator during the warm up period. This causes a considerably longer warm up period and it can even cause the engine to never reach its operating temperature at all. A leaking thermostat can have a negative impact on the vehicle exhaust gas emissions and it may prevent other monitoring systems in the vehicle. This is why the US Environmental Protection Agency (EPA) requires that a failing thermostat is detected by the vehicles on board diagnostics system (OBDII).

Currently, the diagnostic function compares the coolant temperature during warm up (with closed thermostat) with the temperature predicted by a model. If there is too much deviation between the predicted coolant temperature and the real (measured) temperature, the thermostat is diagnosed as "failing". The downside of the current model is that it is hard and time consuming to calibrate. It also has limited accuracy since it doesn't use physical parameters. To reduce the calibration time and improve the accuracy, a physically based model is needed.

1.2 Project definition

The goal of this project is to find the best suitable structure and develop a physically based model for predicting the coolant temperature which can be easily calibrated and implemented in the Engine Control Unit (ECU) of a car. This is done by first performing a literature study in order to improve the knowledge of a cooling system energy balance, to investigate if (and how) this problem has been solved in the past and to find possible structures for the model.

In a second phase, possible structures and sub models are evaluated and the most suitable for this application is chosen. Thereafter, a computer model is made (in this case with use of MatLab/Simulink). Each of the sub models is parameterized and validated with data available from the product development department and from measurements done on vehicles or engines. In addition, the sensitivity of the model on each sensor input error and parameter errors is investigated.

Finally, a diagnostic function to detect a failing thermostat is proposed and implemented in the model.

1.3 Legal requirements

The official requirements of the US EPA and California Air Resource Board (CARB) are stated in Section 1968.2 of the California Code Regulations, Malfunction and Diagnostic System Requirements for 2004 and Subsequent Model-Year Passenger Cars, Light-Duty Trucks, and Medium-Duty Vehicles and Engines (OBD II). This document says:

"The OBD II system shall detect a thermostat malfunction if, within an executive officer approved time interval after starting the engine, either of the following two conditions occur: 1- The coolant temperature does not reach the highest temperature required by the OBD II system to enable other diagnostics; 2- The coolant temperature does not reach a warmed-up temperature within 20 degrees Fahrenheit of the manufacturer's nominal thermostat regulating temperature. Subject to executive officer approval, a manufacturer may utilize lower temperatures for this criterion upon the executive officer determining that the manufacturer has demonstrated that the fuel, spark timing, and/or other coolant temperature-based modifications to the engine control strategies would not cause an emission increase of 50 or more percent of any of the applicable standards (e.g., 50 degree Fahrenheit emission test, etc.)"

Chapter 2

Methodology

2.1 Literature study

In order to improve the knowledge of cooling system energy balance, to investigate if (or how) this problem has been solved in the past and to find possible structures for the model a literature study is carried out. The literature used for this project consists mainly of SAE papers, patents, master theses and PhD theses. The main topics of interest are the heat flows within the engine and engine bay and how to model these heat flows.

2.2 Physical background

This project is based on energy flows particularly. Therefore, to model the coolant temperature, the general laws for heat transfer, heat capacity and chemical energy are applied. The following physical phenomena are modeled.

2.2.1 Energy balance

An engine is a thermodynamic system. Therefore the first law of thermodynamic (eq. 2.1) is applicable on this system.

$$dU = \Delta Q - \Delta W \quad (2.1)$$

With U the internal energy, Q the heat added to the system and W the work produced by the system. This relation is the base of the developed models.

2.2.2 Convection

By definition, convective heat transfer is the transfer of heat from one place to another by the movement of fluids. There are two types of convective heat transfer:

- Free or natural convection: when the fluid flow is not forced to move by an external source but moves on its own
- Forced convection: when the fluid is forced to flow over the surface by an external source such as a fan, stirrer, and pump, creating an artificially induced convection current.

The heat flow by convection is given by the equation 2.2.

$$Q = h \cdot A \cdot (T_{env} - T) \quad (2.2)$$

The Nusselt number Nu is the ratio of convection to conduction heat transfer between two systems (eq 2.3).

$$Nu = \frac{h \cdot l}{k} \quad (2.3)$$

Therefore the convection coefficient is given by:

$$h = \frac{k}{d} \cdot Nu \quad (2.4)$$

It is important to differentiate the forced and natural convection to calculate the Nusselt coefficient.

For forced convection:

A common and relatively simple way to estimate the Nusselt coefficient Nu for forced convection is to use an equation of the form :

$$Nu = A \cdot Re^m \cdot Pr^n \quad (2.5)$$

The Dittus-Boelter heat transfer correlation for fluids in turbulent flow is one of them.

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^n \quad (2.6)$$

The Reynold's number is used to characterise the turbulent behaviour of the flow. By definition :

$$Re = \frac{\rho V L}{\mu} \quad (2.7)$$

The Prandtl number Pr is a characteristic of the fluid. By definition:

$$Pr = \frac{c_p \mu}{k} \quad (2.8)$$

For natural convection:

In natural convection the Reynolds number doesn't really make sense since the velocity of the fluid is very low. Therefore the Rayleigh number is used instead. By definition:

$$Ra_x = \frac{g \cdot \beta}{\nu \cdot \alpha} (T_s - T_\infty) x^3 \quad (2.9)$$

With β the thermal expansion coefficient, ν the kinematic viscosity, α the thermal diffusivity, T_s the surface temperature, T_∞ the temperature far from the wall, g the gravity and x the characteristic length.

In an engine, the convection is an important phenomenon. Since most of the fluid flows are driven by a pump, the forced convection is the most common.

2.2.3 Conduction

Conduction is the transfer of thermal energy between regions of matter due to a temperature gradient. Heat spontaneously flows from a region of higher temperature to a region of lower temperature, and reduces temperature differences over time, approaching thermal equilibrium. To calculate the heat transfer by conduction the Fourier's law is used.

$$q = -k \cdot \nabla T \quad (2.10)$$

The conduction heat transfer coefficient is a property of the material and can be found in tables. Considering one dimensional and homogeneous material, the Fourier's law can be expressed as:

$$\frac{\Delta Q}{\Delta T} = -k \cdot A \cdot \frac{\Delta T}{\Delta x} \quad (2.11)$$

In an engine the conduction mainly occur through the block and liners but also between the different engine parts.

2.2.4 Radiation

Thermal radiation is electromagnetic radiation emitted from all matter due to its possessing thermal energy which is measured by the temperature of the matter.

$$P = \epsilon \cdot \sigma \cdot A \cdot T^4 \quad (2.12)$$

Epsilon is the emissivity of the material and sigma the Stefan-Boltzmann constant. In the engine, radiation occurs in two places:

- Between the burning gas and the cylinder wall
- Between the block and the air
- Between the block and the other surfaces in the engine bay

2.2.5 Heat capacity

Heat capacity is the measurable physical quantity that characterizes the amount of heat required to change a body's temperature by a given amount. In the International System of Units, heat capacity is expressed in units of joules per Kelvin.

$$C = \frac{\Delta Q}{\Delta T} \quad (2.13)$$

In this application the heat capacity is used in the following formula in order to estimate the temperature in function of the heat flow:

$$Q = c \cdot m \cdot dT \quad (2.14)$$

2.2.6 Heat of combustion

The heat of combustion is the heat released by the fuel-air mixture during the combustion process. The main parameter describing the heat energy content of a fuel is its heating value. Different heating values exist. The higher heating value (also called gross) is the amount of heat produced by a complete combustion of a certain unit of fuel. It is determined when the products of combustion are at the same temperature as before combustion and when the produced water is condensed. The lower heating value is the gross heating value minus the latent heat of vaporization of water. The energy required to vaporize the water is not realized as heat. The lower heating value (LHV) is considered to predict the heat from combustion in the engine because the water is not condensed after combustion in the engine.

2.3 Engine and under hood heat flow

Based on the information that is found in the literature, an overview of the possible heat flows in the engine and in the engine bay is made, see figure 2.1. The arrows in figure 2.1 represent the energy flows by convection, conduction and radiation. The "Mechanical energy" is the energy that results from the combustion process that is available on the crankshaft. The "Heat from gas" represents the available energy in the inlet air flow and exhaust gas flow. One can see that a part of the exhaust gas energy is used by the turbocharger. The

”Heat due to compression” arrow represents energy resulting from a physical process like in this case, compression of inlet air by the compressor. The figure shows what heat flows can have an affect on the coolant temperature. As can be seen, the expected heat flows to the coolant are from combustion in the cylinders (via liner), from the engine block, from oil cooler, turbo cooler and transmission cooler. Heat is rejected from the coolant by the cabin heater.

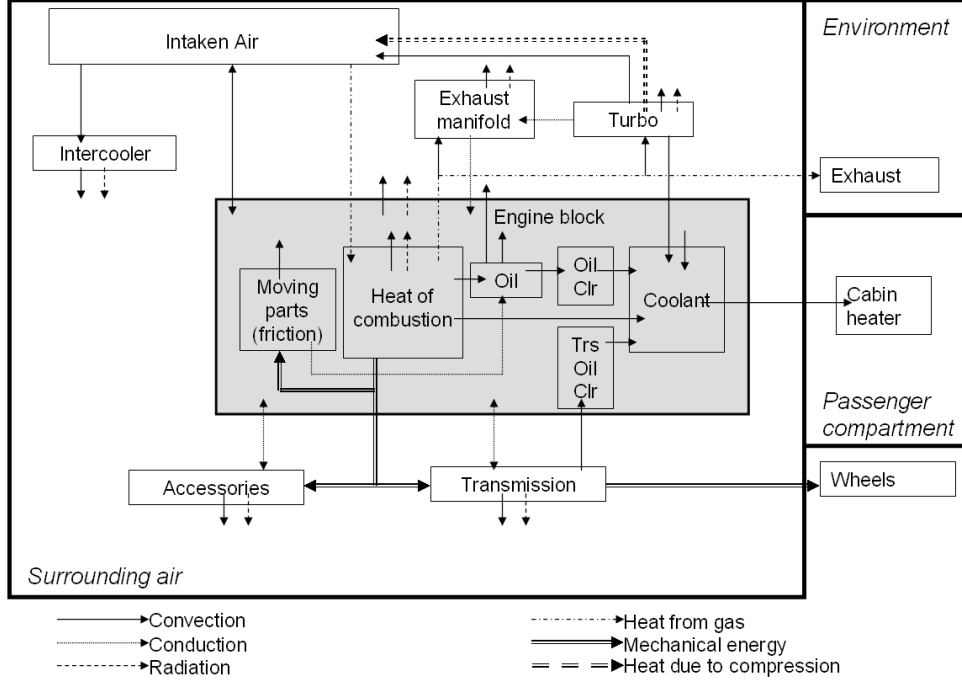


Figure 2.1: Global heat flow diagram of an engine

2.4 Model structure

Modeling every single heat flow shown in figure 2.1 to predict the coolant temperature is complicated and requires a lot of knowledge, time and calibration effort. Thereby, running such a complicated model requires high CPU capacity from the ECU. For these reasons, a simplified model structure is needed.

It was found in the literature, that for modeling the heat from combustion going to the cylinder walls, basically two methods are used:

1. Determine the heat flow from combustion into the cylinder wall/liner.
2. Determine the heat input from fuel, heat out of exhaust and mechanical power output and use the following relation to determine the heat flow to the cylinder walls;

$$Q_{wall} = Q_{fuel} - Q_{exh} - P_{out} \quad (2.15)$$

Method 1 is a direct method to calculate the main heat flow to the walls. The method is mainly used for modeling complete engines and cooling systems in a development phase where real engines aren't available yet. This method requires in cylinder combustion data and works well for engines at operating temperature. However, during warm up, parameter settings that affect the combustion process such as mixture quality and ignition timing are different compared to when the engine is at operating temperature. These effects are hard to model and sensitive to calibration changes which makes the method less suitable for this application. Additionally, the heat exchange between the hot gas and the wall are very complex due to the high turbulence and the combustion process complexity.

Method 2 is an indirect method and needs input data that is mostly already available from the ECU. This makes the model more flexible for use on different engines and less sensitive to calibration changes. For these reasons method 2 is used in this application.

When the heat that goes to the walls is derived by one of the two previous methods, different methods can be used to find out the good amount of heat going to the coolant:

1. All the heat will go directly into the coolant.
2. Derive it by allowing a heat transfer equation between and through the different materials. For instance conduction through the cylinder wall or convection between the block and the coolant.
3. Consider the coolant as a part of a bigger system and apply the first law of thermodynamic to it.

The first method is very approximative since the coolant is also losing a part of its energy via other systems such as the heater.

The second method is the most complex since every connection between the different parts has to be modeled. That involves knowing the exact geometry of every contact surface but also the exact flow characteristic to find the heat transfer coefficient between different fluids and the surfaces. Considering the calibrateability, this approach is pretty complex since every heat flow has to be calibrated. It could be hard to obtain measurements to be able to do it in a limited amount of time. Another problem is the future adaptation of this model to hybrid cars. That could involve the adjunction of new elements and more flow models.

The third method is more global and doesn't consider every "small" heat transfer between parts. Every global heat loss is calculated and subtracted or added to the "Heat from combustion". In that case, the measurements are limited and the data fully available. The calibration time and the understanding of the

model are fast and intuitive. For the hybridization, few more blocks like battery cooling can be added in a very easy way.

Therefore, the last method is chosen for further development.

To develop the simplest model structure needed to fulfill the requirements, models are made starting from the most basic structure possible and increased in complexity until the wanted accuracy is reached. The relative importance of the different sub-models is determined by the literature study. The sub-models with the greatest influence are added first, followed by sub-models with less influence. In this way a structure is developed which is the least advanced as needed. The structure is increased in complexity according to the following planning.

Model 1:

The first model developed includes the most important energy flows: the heat from the combustion, the power output and the heat to the coolant. The coolant, the oil and the engine block are considered as one mass with uniform temperature (see table 2.1 and figure 2.2).

Table 2.1: The characteristics of the model 1

Energy flow	Mass	Heat to mass
Heat from fuel Heat to exhaust Mech. power output	Single mass for coolant, block and oil	$Q_{coolant} = Q_{fuel} - Q_{exh} - P_{out}$

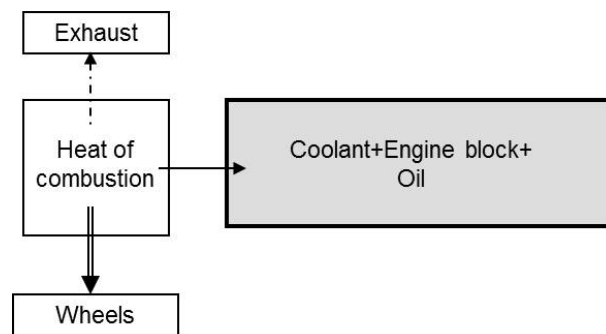


Figure 2.2: Considered system for the first model

Model 2:

The model of the cabin heater is added to the model 1 (see table 2.2 and figure 2.3).

Table 2.2: The characteristics of the model 2

Energy flow	Mass	Heat to mass
Heat from fuel Heat to exhaust Mech. power output Heat to cabin heater	Single mass for coolant, block and oil	$Q_{coolant} = Q_{fuel} - Q_{exh} - P_{out} - Q_{heater}$

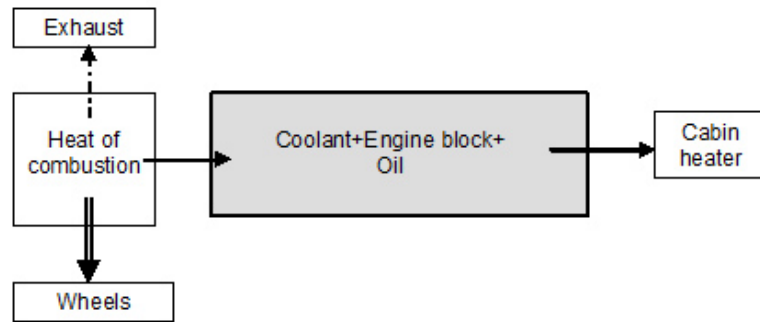


Figure 2.3: Considered system for the second model

Model 3:

The convection to surrounding air is added to the model 2.

Table 2.3: The characteristics of the model 3

Energy flow	Mass	Heat to mass
Heat from fuel Heat to exhaust Mech. power output Heat to cabin heater Heat conv. engine block to air	Single mass for coolant, block and oil	$Q_{coolant} = Q_{fuel} - Q_{exh} - P_{out} - Q_{heater} - Q_{coolToair}$

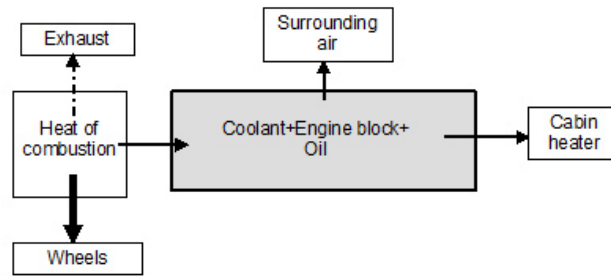


Figure 2.4: Considered system for the third model

Model 4:

Two main improvements are done to create the model 4. The heat from friction is added to the model 3 and the oil is separated from the coolant+block mass (see table 2.4 and figure 2.5).

Table 2.4: The characteristics of the model 4

Energy flow	Mass	Heat to mass
Heat from fuel Heat to exhaust Mech. power output Heat to cabin heater Heat conv. engine block to air Friction from moving parts	Single mass for coolant and block Mass for oil	$Q_{coolant} = Q_{fuel} - Q_{exh} - P_{out} - Q_{heater} - Q_{coolToair} + Q_{oilToCool} + x \cdot Q_{friction}$ $Q_{oil} = (1 - x) \cdot Q_{friction} - Q_{oilToCool}$

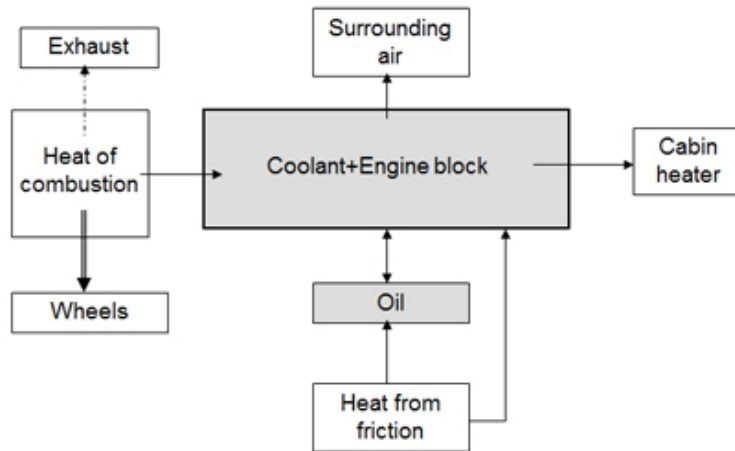


Figure 2.5: Considered system for the fourth model

Model 5:

The block mass is separated from the coolant. The system is composed by three independant masses (see table 2.5 and figure 2.6).

Table 2.5: The characteristics of the model 5

Energy flow	Mass	Heat to mass
Heat from fuel	Mass for coolant	$Q_{coolant} =$
Heat to exhaust	Mass for block	$Q_{fuel} - Q_{exh} - P_{out} - Q_{heater} -$
Mech. power output	Mass for oil	$Q_{coolantToBlock} + Q_{oilToCool}$
Heat to cabin heater		$Q_{oil} = x \cdot Q_{friction} -$
Heat conv. engine block to air		$Q_{oilToCool} - Q_{oilToBlock}$
Friction from moving parts		$Q_{ToBlock} = (1 - x) \cdot$
		$Q_{friction} + Q_{coolantToBlock} -$
		$Q_{coolToair} + Q_{oilToBlock}$

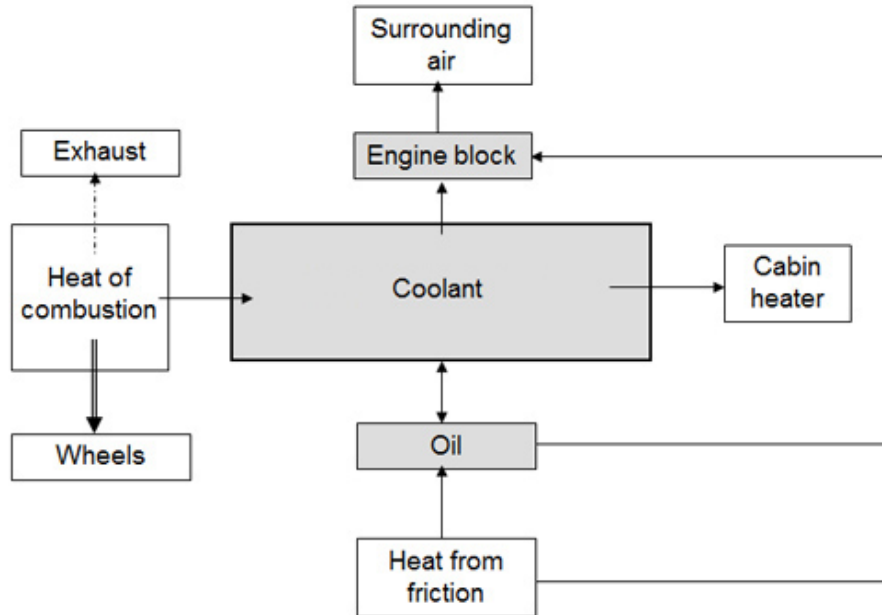


Figure 2.6: Considered system for the fifth model

Model 6:

The convection and conduction from the exhaust manifold is added to the model 5 (see table 2.6 and figure 2.7).

Table 2.6: The characteristics of the model 6

Energy flow	Mass	Heat to mass
Heat from fuel Heat to exhaust Mech. power output Heat to cabin heater Heat conv. engine block to air Friction from moving parts Conduction/convection from exhaust manifold	Mass for coolant Mass for block Mass for oil	$Q_{coolant} =$ $Q_{fuel} - Q_{exh} - P_{out} - Q_{heater} -$ $Q_{coolantToBlock} + Q_{oilToCool}$ $Q_{oil} = x \cdot Q_{friction} -$ $Q_{oilToCool} - Q_{oilToBlock}$ $Q_{ToBlock} = Q_{ToBlock} =$ $(1 - x) \cdot Q_{friction} +$ $Q_{coolantToBlock} - Q_{coolToair} +$ $Q_{oilToBlock} + Q_{manifoldToBlock}$

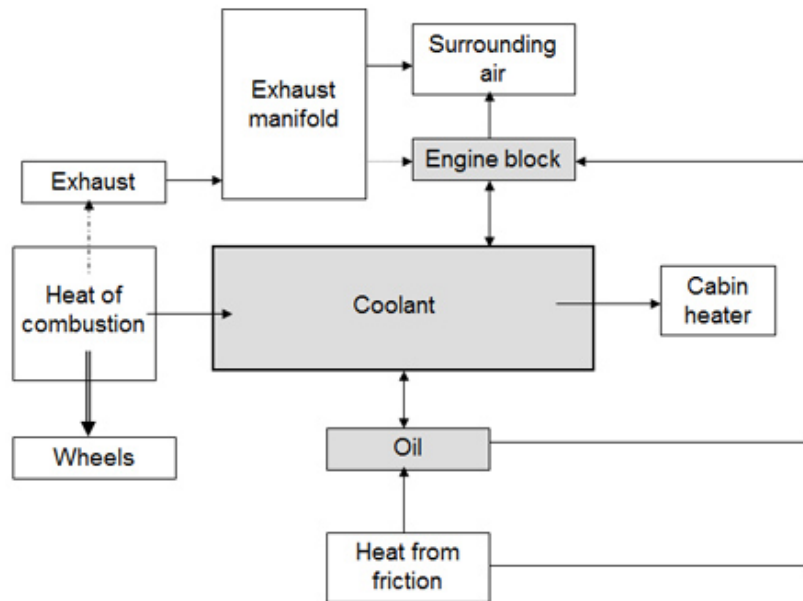


Figure 2.7: Considered system for the sixth model

Model 7:

The turbo watercooling is added to the model 6 (see table 2.7 and figure 2.8).

Table 2.7: The characteristics of the model 7

Energy flow	Mass	Heat to mass
Heat from fuel	Mass for coolant	$Q_{coolant} =$
Heat to exhaust	Mass for block	$Q_{fuel} - Q_{exh} - P_{out} -$
Mech. power output	Mass for oil	$Q_{heater} - Q_{coolantToBlock} +$
Heat to cabin heater		$Q_{oilToCool} + Q_{turbo}$
Heat conv. engine block to air		$Q_{oil} = x \cdot Q_{friction} -$
Friction from moving parts		$Q_{oilToCool} - Q_{oilToBlock}$
Conduction/convection from exhaust manifold		$Q_{ToBlock} = Q_{ToBlock} =$
Turbo water cooling		$(1 - x) \cdot Q_{friction} +$
		$Q_{coolantToBlock} - Q_{coolToair} +$
		$Q_{oilToBlock} + Q_{manifoldToBlock}$

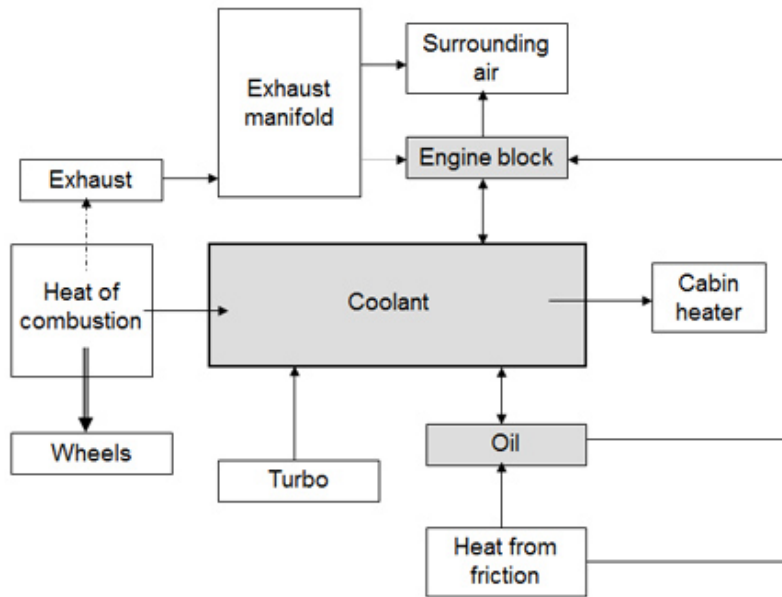


Figure 2.8: Considered system for the seventh model

The transmission oil cooling is not considered in this project since during the warm-up period, the transmission oil is not cooled.

To determine the coolant temperature the first law of thermodynamics is applied to the system including the coolant (in grey with bold boarder).

With model 3, a satisfying prediction of coolant temperature is made. Improving the model to model number 4 did not make a significant improved to the coolant temperature prediction (see table 2.8) and required parameters can be difficult to obtain. For this reason model 3 is further used in this project. As can be seen in the table 2.8 the improvement from model 1 to model 2 and to model 3 are very important. The improvements made by the model 4 are negligible regarding the application of this model.

Table 2.8: Comparison between the error (in C) of the different model for all the measurements and the driving measurements

	Model 1	Model 2	Model 3	Model 4
Global average error	11.1	6.0	3.2	3.5
Global Standard deviation	5.9	4.0	4.3	4.0
Driving average error	12.9	6.4	1.3	-0.01
Driving standard deviation	5.2	4.6	4.0	3.7

2.5 Sub-models

In the model, the different heat flows are simulated with sub models. From the literature many models to simulate the different heat flows are found, each with its own properties. To find the best suitable sub models to use in the model a rating scheme is made. Each sub model is rated on four criteria; calibrateability, accuracy, complexity and applicability.

- Calibrateability; describes if it is possible to find the needed constants and parameters within a reasonable time, cost and equipment. 0 means hard to calibrate, 5 means easy to calibrate.
- Accuracy; describes how accurate the model is compared to reality. Considering if the sub model validated with measurements and if it is a first version or if it has been improved. 0 means not accurate, 5 means accurate.
- Complexity; describes how complex the model is in terms of number of equations, maps, parameters and the degree of the equations used. 0 means complex, 5 means simple.
- Applicability; describes how well the sub model can be used for this application, considering the inputs needed, if the state of technology is up to date and if the sub model is used in a comparable application. Also if the model works well for both transient and steady state engine operating and engine warm up. 0 means hardly applicable, 5 means well applicable.

Not all criteria are equally important, therefore the criteria are weighted to their importance. Table 2.9 shows the weight factor for every criterion. The score of a sub model is the average of the rating times the weight factor.

Table 2.9: Importance of the different rating criteria

Criterion	Weight factor
Calibratability	2
Accuracy	3
Complexity	1
Applicability	3

For every relevant paper found, it is listed which sub model is inside and each of these sub models is rated. All the scores are plotted in one summary. Figure 2.9 shows the summary, scores above 3.5 are highlighted in green, scores between 3.5 and 2 are in yellow and scores below 2 are red. This method shows which sub models are the best to use in the model.

a: Calibratability b: Accuracy c: Significance submodel d: Complexity e: Applicability	Heat of combustion to oil	Heat of combustion to coolant	Heat of combustion to block	heat from block to air	Heat from coolant to block	Heat from oil to block	Heat to exhaust gas	Exhaust gas to manifold	Intake air to manifold	Heat exchanger	Heater	Accessories	Heat of combustion to wall	Global heat model	Friction model	Heat from oil to coolant	Friction to oil	Conduction into block
Paper1		2.9								1.9			3.9	2.9				
Paper2		2.9											3.9	3.2	2.6	1.9		
Paper3								4.2	4.2									
Paper4		3.6		3.9										4.0				
Paper5								4.2										
Paper6		2.2												2.1				
Paper7		2.6		3.2							3.4		3.2	2.9				
Paper8		3.6		2.9										3.9				
Paper9																		
Paper10				2.7			4.0	3.7						4.9				
Paper11																		
Paper12																		
Paper13						2.8	2.0	2.8						2.8				4.2
Paper14		3.8					3.7	3.7	3.7					3.0				
Paper15																		
Paper16		3.6			3.6								2.9					
Paper17										4.0				2.6				
Paper18														2.2				
Paper19															3.4			
Paper20													1.9	2.6				
Paper21							3.7							5.0				
Paper22																		
Paper23													2.9					
Paper24													3.6					
Paper25						3.4	3.9											
Paper26																		
Paper27																		
Paper28																		
Paper29													1.9	2.6				
Paper30																1.4		
Paper36																		

Figure 2.9: The excel sheet used to rate the papers

2.6 Model parameterization and validation

2.6.1 Measurements set-up

In order to validate the model, several measurements are done on a real car. A new Volvo S60 equipped with an I5T engine (5 cylinder turbocharged) is used. A set of measurements is done in a climatic wind tunnel. That gives fully controlled environment in terms of vehicle speed (wind), load and air temperature. During these tests, engine and environment parameters are recorded during the warm up period until the thermostat opens. The experiments are performed at steady state speed and load for various operating points. The measurements were done at temperatures of -10 and +20 degree C. Between every measurement the car is cooled down in order to perform cold start.

Other data are collected in normal driving condition to give the most realistic data to test the model. To make sure that the model is able to take into account a stop and start system, the engine is shut off when the car is stopped on some tests. All these collected data are used to calibrate the model and test it.

An equivalent set of measurements is done with and without leaking thermostat in order to validate the algorithm ability to detect defective thermostat. It is impossible to perform two times the same measurement in normal driving condition. This is why other measurements are done in a test bench following the FTP driving cycle. It gives the opportunity to compare exactly the difference between thermostat performances on a reference driving cycle. The FTP driving cycle is used for homologation in the USA.

2.6.2 Previous model

The previous model behavior is monitored for the different measurements in order to compare it with the new one. Since the previous model is used on current cars, it gives a good indication of the minimum accuracy the new model should achieve.

2.6.3 Diagnostic

The diagnostic algorithm is tested by using measurement data with leaking thermostat. A customizable leaking thermostat is mounted on the car and driving tests are done with different degree of leakage.

Chapter 3

Model Description

3.1 Global model

The main requirement on the model is to be physically based to guarantee its calibratability. An energy balance approach is chosen. Hence, to calculate the heat going to the coolant, the different heat consumption are subtracted from the heat produced.

$$Q_{cool} = Q_{combustion} - Q_{exh} - P - Q_{heater} - Q_{surrounding} \quad (3.1)$$

In such engine, the produced energy comes from the combustion of the fuel into the cylinder resulting in both mechanical energy and heat. The different power losses (not going to or leaving the coolant) are:

- The heat going to exhaust, Q_{exh}
- The mechanical output (Through the crankshaft), P
- The heat to the cabin heater, Q_{heater}
- The heat to surrounding air, $Q_{surrounding}$

Knowing the heat going to the coolant it becomes possible to calculate a coolant temperature. In the model 3 that has been chosen, the coolant, the block and the oil are considered as one system.

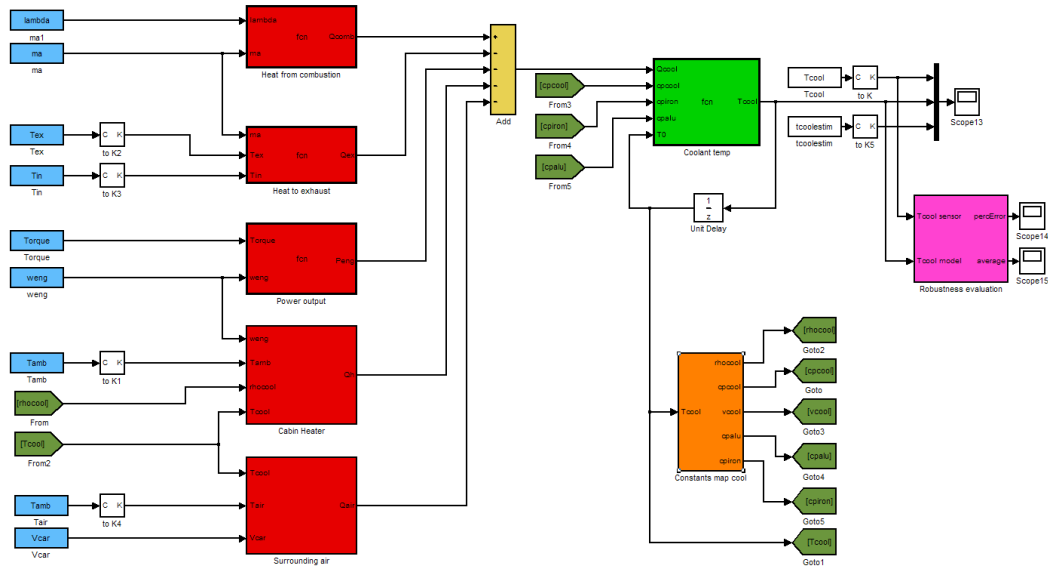


Figure 3.1: The global simulink model

The figure 3.1 shows the global simulink model. The light blue boxes are the data coming from the ECU that are used by the model. The red boxes are the different heat production or losses. The green box represents the coolant and calculates its temperature. Finally the orange box defines the density, the specific heat or the viscosity of different elements as function of the temperature. The inputs to the model are:

- λ , the relative air/fuel ratio
- The air flow through the engine
- The exhaust temperature
- The inlet temperature
- The output torque
- The engine speed
- The ambient temperature
- The speed of the car

All these values are monitored by the ECU and are easily accessible.

3.2 Heat from combustion sub model

This sub-model describes the amount of heat created during the combustion of the air fuel mixture. The figure 3.2 shows the simulink structure of this sub-model.

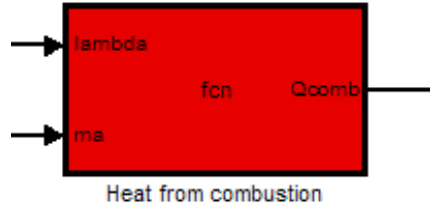


Figure 3.2: The simulink model of the heat from combustion sub-model

All the energy that warms up the coolant comes from the fuel injected into the engine. The amount of heat that comes from the fuel depends on the following factors:

- Amount of fuel injected
- Lower heating value (LHV) of the fuel
- Evaporation heat of fuel
- Lambda
- Actual AFR of the mixture
- Combustion efficiency

The lower heating value, stoichiometric AFR and the heat of evaporation are fuel characteristics and don't need to be measured by the ECU. They are calibration constants. In this model, the amount of fuel is calculated as function of the air mass flow, stoichiometric AFR and lambda. The air mass flow and lambda are measured by the ECU. With the following formula the fuel mass flow is calculated:

$$\dot{m}_f = \frac{\dot{m}_a}{stoich \cdot \lambda} \quad (3.2)$$

λ characterizes if the engine is running lean (excess of air) or rich (excess of fuel). In the case the engine is running lean, the fuel amount will determine the amount of energy by limiting the combustion process. However if the engine runs rich, the mass of air will determine the amount of energy.

Not all the fuel that is injected into the engine will turn into heat. A certain part of the chemical energy of the fuel is lost in the combustion process. The

combustion efficiency describes how much of the chemical energy is turned into heat. The combustion efficiency of a recent gasoline engine varies between approximately 0.96 and 0.97. There are several reasons why not all the injected fuel is converted into heat. First of all, the mixture in the cylinder is never perfectly homogeneous. This means that not all the fuel molecules can react with the oxygen in the air and therefore will leave the cylinder as unburned hydrocarbons. In rich conditions, this effect is worsened. In lean conditions, misfire and incomplete combustion can occur. A part of the fuel is also going into the crevices of the cylinder and remains there unburned. The fuel can also be absorbed by the oil layer on the wall. In addition, the flame quenching at the wall leaves an unburned layer of fuel nearby the wall. To calculate the total amount of heat from combustion in lean conditions, the following formula is used:

$$Q_{combustion} = \dot{m}_f \cdot \eta \cdot (LHV - H_{evap}) \quad (3.3)$$

For rich conditions:

$$Q_{combustion} = \frac{\dot{m}_a}{stoich} \cdot \eta \cdot (LHV - H_{evap}) \quad (3.4)$$

Both the lower heating value and the stoichiometric air/fuel ratio has to be calibrated using the average value from the fuel available in the region the car will drive (The fuel characteristics differ between the USA and Europe). For this project the model is calibrated according to the characteristics of the fuel used by Volvo in the test vehicles.

The main risk for this submodel is the fuel quality variation depending on the supplier, the season or the country. As it will be shown later, the impact of this variation is low enough to not create problems in the temperature prediction. The air flow sensor or model has to be accurate enough as well.

3.3 Power output sub-model

This sub-model calculates the power coming out of the engine at the crankshaft. The figure 3.3 shows the simulink structure of this sub-model.

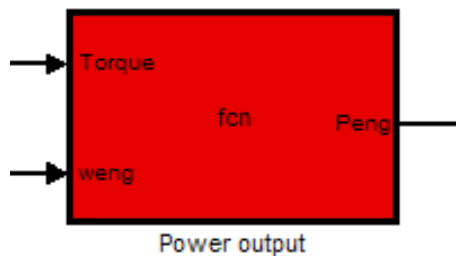


Figure 3.3: The simulink model of the power output sub-model

The power output is the mechanical power coming out of the engine. The definition of this power is :

$$P = \omega_{eng} \cdot T \quad (3.5)$$

This power is the net power minus the friction losses. In the friction losses are included: bearing, piston+piston ring friction, accessories friction and pumping losses. Both the engine speed and engine torque are variables measured or modelled by the ECU. During engine braking, the 'brake torque' caused by friction is turned into heat and warms up the coolant. In the case the engine speed is measured as less than 500rpm, the torque is set-up at 0Nm in order to take care of wrong measurement when the engine is shut off. No calibration is required for this model. The only risk comes from a bad evaluation of the torque and the engine speed.

3.4 Heat to exhaust sub model

This sub-model evaluates the amount of heat going out of the engine through the exhaust ports. The figure 3.4 shows the simulink structure of this sub-model.

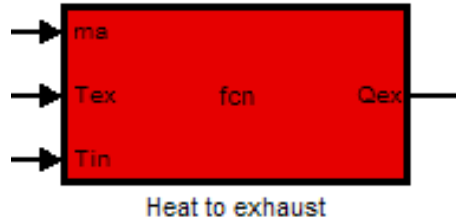


Figure 3.4: The simulink model of the heat to exhaust sub-model

The heat going to the exhaust is a main contributor of the heat losses. About 20% of the total energy produced is going to the exhaust.

To calculate the heat going to the exhaust, it is required to know the exhaust gas flow and temperature. The gas flow is the sum of the air flow and the fuel flow through the cylinder. The exhaust gas temperature and the air flow are obtained from the ECU. The following equations are used.

$$\dot{m}_f = \frac{\dot{m}_a}{stoich \cdot \lambda} \quad (3.6)$$

$$\dot{m}_{exhaust} = \dot{m}_a + \dot{m}_f \quad (3.7)$$

$$Q_{exhaust} = \dot{m}_{exhaust} \cdot c_{p,gas} \cdot (T_{exh} - T_{in}) \quad (3.8)$$

The heat capacity of the gas is calculated according to its composition. It is $1.1 \text{ J/g} \cdot \text{K}$.

The inlet air temperature is measured after compressor and intercooler and the value is available from the ECU. Exhaust gas temperature is also available from the ECU but is determined by a model. No calibration is required on this submodel.

The risk come from a wrong evaluation of the exhaust temperature (based on a model in a car) and of the air flow measurement.

3.5 Heat to surrounding air sub model

This sub-model simulates the heat exchange between the engine and the surrounding air. The figure 3.5 shows the simulink structure of this sub-model.

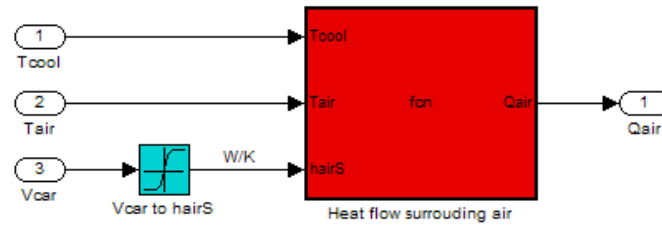


Figure 3.5: The simulink model of the heat to surrounding sub-model

The heat going from the engine block to the surrounding air has two components: the convection and the radiation. The radiation is negligible comparing with the convection. Therefore, only the convection will be considered in the model. The convection equation applied to the engine is:

$$Q_{air} = h \cdot S \cdot (T_{cool} - T_{air}) \quad (3.9)$$

The main difficulty is to estimate the heat transfer coefficient and the outer surface of the engine. The coefficient varies with the speed of the car, increasing the heat transfer by convection at high speed. The air flow around the engine is not well known (both flow and temperature) due to the complex geometry of the engine bay. The block surface temperature is not known either. Hence it becomes extremely complex to evaluate the coefficient by using the Nusselt and Prandtl numbers like shown in chapter 2. Instead of calculating the heat transfer coefficient, it is being derived from a few simple measurements. Starting from a warm engine, the engine is shut off and the coolant temperature is measured during the cooling down. By doing that at a given ambient temperature and for

different vehicles speeds (in a wind tunnel) it is possible to evaluate the product coefficient*surface with the following formula:

$$hS = \frac{m \cdot c_p \cdot (T_{c1} - T_{c2})}{(T_c - T_{air})} \quad (3.10)$$

Where, $m \cdot c_p$ is the total heat capacity of the block, coolant and oil combined. T_{c1} is the coolant temperature at time 1, T_{c2} is the coolant temperature at time 2. T_c is the actual coolant temperature and T_{air} is the ambient temperature. The easiest way to find the values of hS is to vary the coefficient used by the model until the modeled coolant temperature curve fits on the measured cooling down curve. By doing that every heat transfer that occur between the block and the air are taken into account. The fact that natural convection occur at stand still is also taken into account by this method. The determined values of hS for the different speeds are set up in a map function of the car speed (figure 3.6). Three or four measurements at different speeds are enough to give good results.

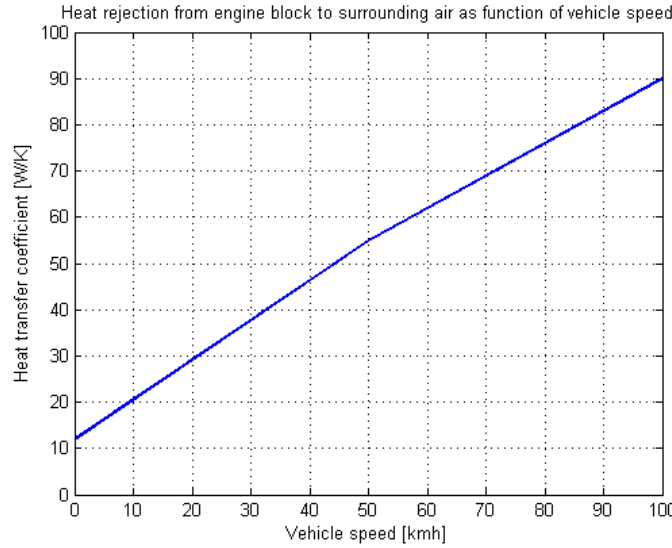


Figure 3.6: The map of heat transfer coefficient as function of the vehicle speed

In these measurements, not the difference between block and under hood air temperature is used, but the difference between the coolant and ambient air temperature. This is done because the block surface temperature and the air temperature under the hood are not known by the ECU and therefore not usable. According to measurements done at varying operating point and vehicle speed, it is concluded that the correlation with coolant and ambient temp (see figure 3.7) is close enough to use those quantities instead in the model. Therefore the heat transfer coefficient is calculated from the ambient and the coolant temperature.

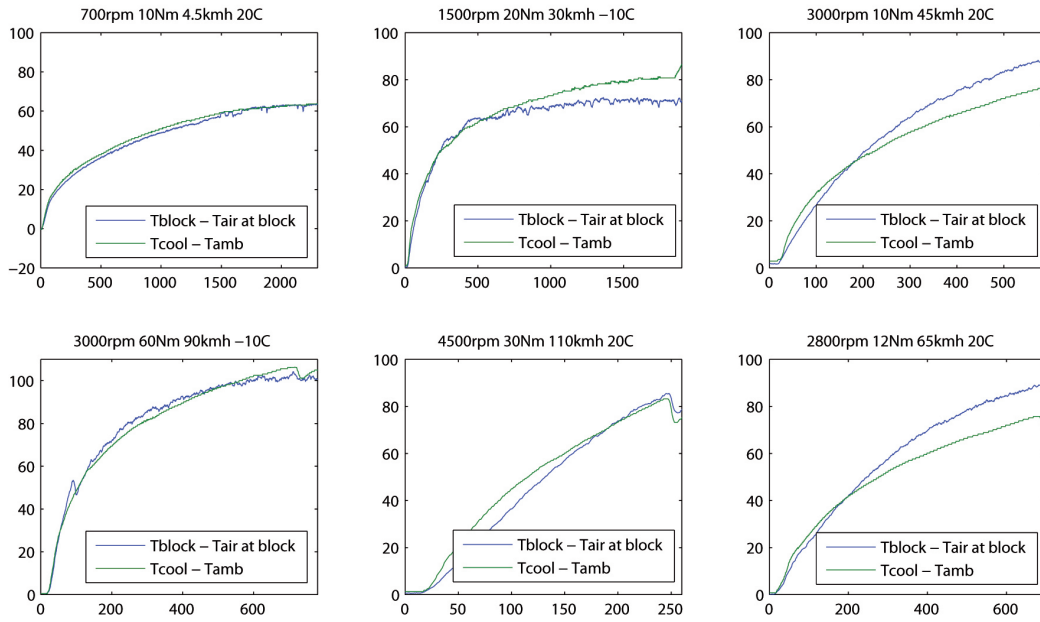


Figure 3.7: Temperature differences block under hood air vs. coolant ambient air

So in the model, the coolant temperature is taken from the previous iteration and for the under hood air temperature, the ambient temperature is taken. Using the measurements for the model means that not only convection but also radiation is taken into account which increases accuracy. The accuracy of this sub model is mainly depending on the ambient temperature and car speed sensors. Considering the surrounding air and the coolant temperature is an approximation since air close to the engine and block's wall temperature should be considered for convection. However, calculating the coefficient with coolant and ambient temperature gives good results.

3.6 Cabin heater sub model

This sub-model is used to estimate the heat withdrawn from the coolant by the cabin heater. The figure 3.8 shows the simulink structure of this sub-model.

This is probably the most complicated submodel included in this project. It involves calculating the heat transfer coefficient of the heater core as function of the air and coolant flows. This submodel includes three maps that have to be calibrated; evaluation of the air temperature at the entrance of the heater core, coolant flow through the heater core and heat transfer coefficient of the heater core.

The layout of the heating system is shown figure 3.9.

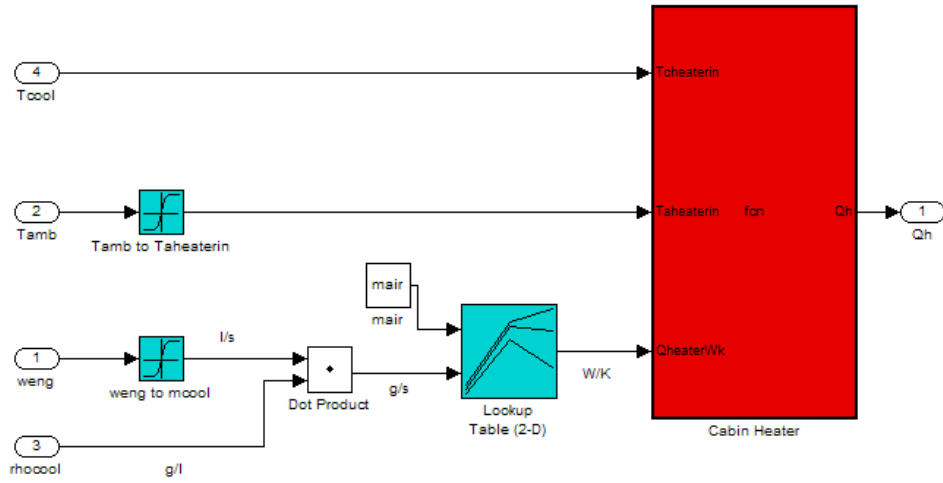


Figure 3.8: The simulink model of the heat to the cabin heater sub-model

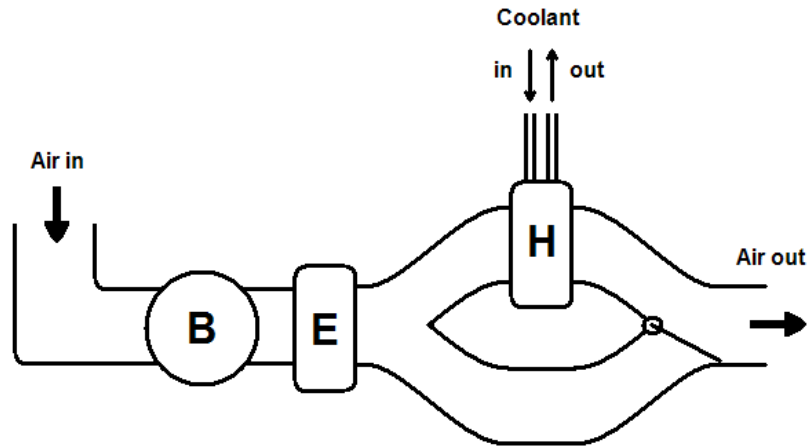


Figure 3.9: The simplified layout of the cabin heater

- B is the blower
- E is the evaporator
- H is the heater core

The air going to the heater first goes through the evaporator (E) that cools down the air in order to dry it or to be able to regulate the temperature in the

cabin. A part of the air is going to the heater core (H) to be warmed up. The cold and warm air are mixed to obtain the requested cabin temperature. In the model, a worst case approach is used. Therefore, all the air is going to the heater core to be warm-up. The cooling down strategy of the evaporator is shown figure 3.10. At temperature below zero, the evaporator is not working to prevent the apparition of ice. Up to an ambient temperature of 10 degrees the air is cooled down to 0 degree. Above 10 degrees the air is gradually less cooled to reach a temperature of 10 degrees to reduce energy consumption and finally above 20 degrees the air is cooled down to 0 degree to reduce the fan noise. The data used to calibrate this map are obtained in the climate system department.

The second map to calibrate is the evaluation of the coolant flow as function of the engine speed (or pump speed in case of an electric pump)see figure 3.11. The cooling system department can give this information.

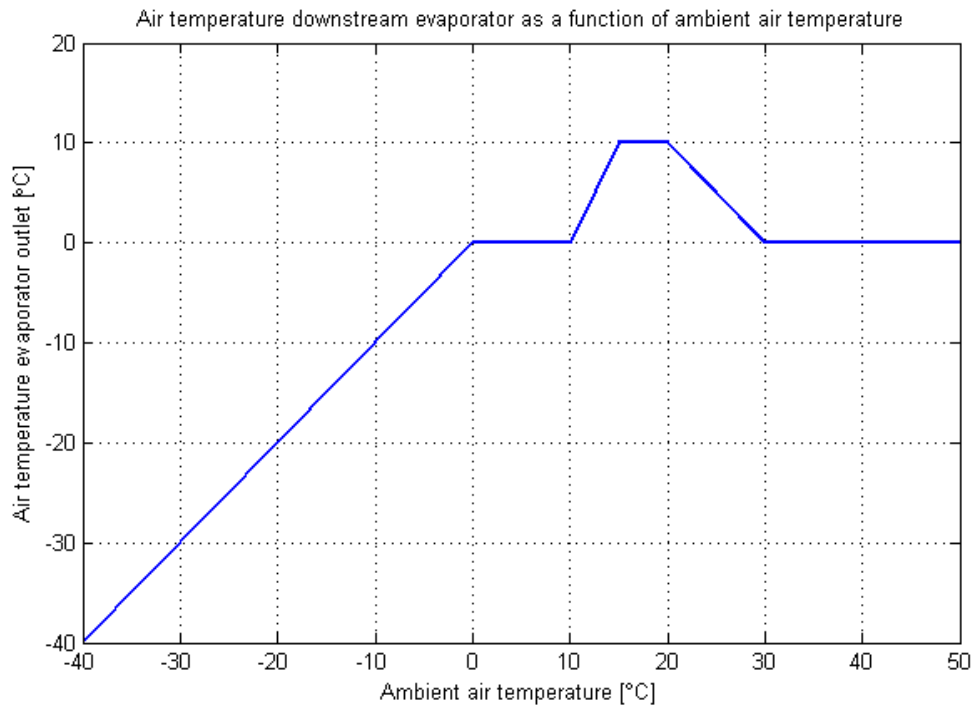


Figure 3.10: The cooling strategy of the evaporator

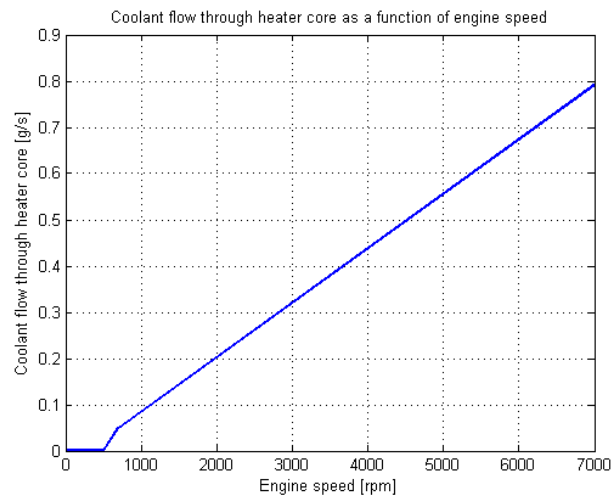
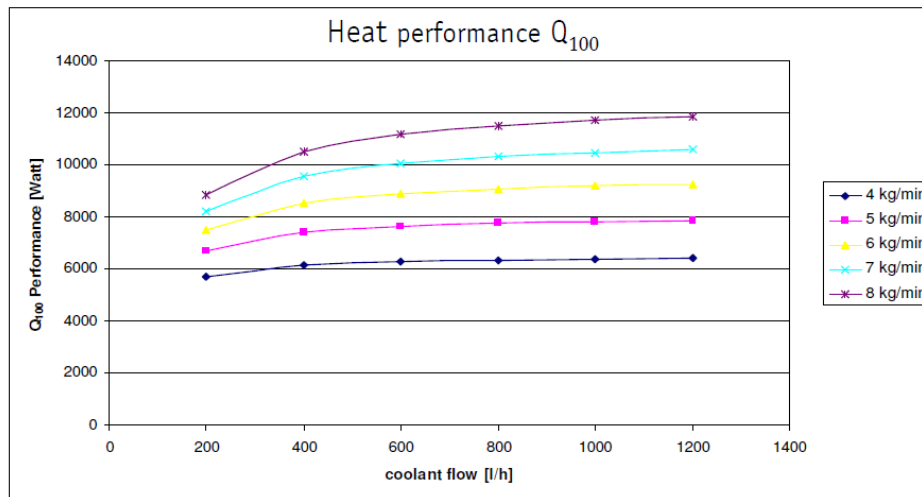


Figure 3.11: The coolant flow through heater as function of engine/pump speed

The last map calculates the heat transfer coefficient of the heater core as function of the air and coolant flows. The supplier of the heater core provides graphs with this characteristic for a 100 degree difference of temperature between the air and the coolant (see figure 3.12). The coefficient is considered as linear with the temperature difference. By extrapolating these curves for a bigger range of flows and converting to the appropriate unit, the map can be filled with the appropriate values (see figure 3.13).



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Heat up. Cool down.

BEHR

Figure 3.12: Q100 of heater core

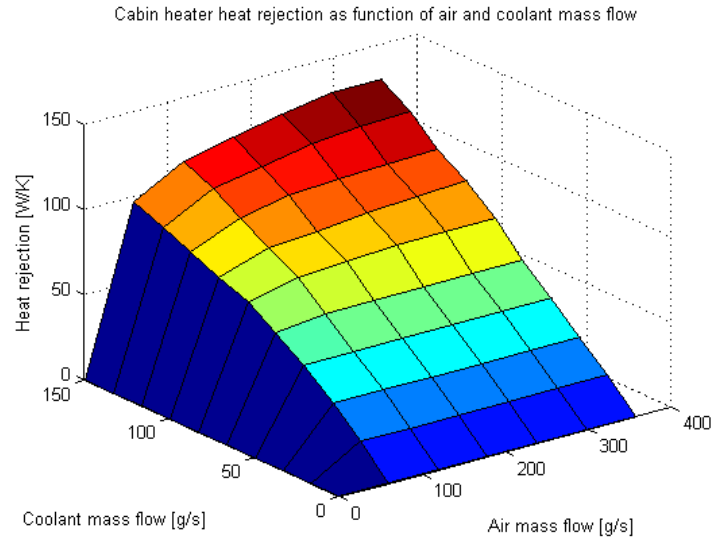


Figure 3.13: The heater core heat transfer map

The evaluation of the heat going to the heater is done with the equation:

$$Q_{heater} = h \cdot (T_{coolIn} - T_{airIn}) \quad (3.11)$$

All the data required for the cabin heater model are acquired in other departments, which means no measurement has to be done.

Error can come from sensor failure but also from a non fully accurate data set in the maps. Since some values are extrapolated, small differences can occur. In the worst case approach, a full blower is used to define the air flow through the heater core. It is not possible to use the blower speed because it is not monitored by the ECU. However, in real conditions, the heater is never used at full load. That can result in an over-estimation of the heat going to the heating system.

3.7 Coolant temperature sub model

This sub-model predicts the temperature of the coolant. The figure 3.14 shows the simulink structure of this sub-model.

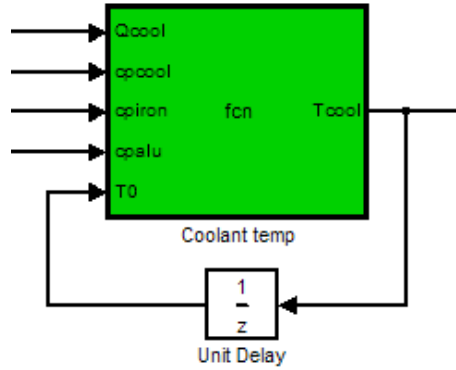


Figure 3.14: The simulink model of the coolant temperature sub-model

The last sub model is used to predict the coolant temperature itself regarding the amount of heat flowing to it. The equation used is the definition of the heat flow (eq 3.12).

$$T_{cool} = T_0 + \frac{1}{c_{p,total}} \cdot Q_{cool} \cdot t_{sample} \quad (3.12)$$

In this project the coolant is lumped together with the oil and block as one mass. The global heat capacity is evaluated using the materials characteristics and the dimensions of the engine. As can be seen in figure 3.15, the biggest part (77%) of the heat capacity comes from the block. The coolant itself is only 13% of the total heat capacity. Figure 3.16 shows the heat capacity of the different parts in the block. As can be seen, the main part of the heat capacity is due to cast iron parts, 62%.

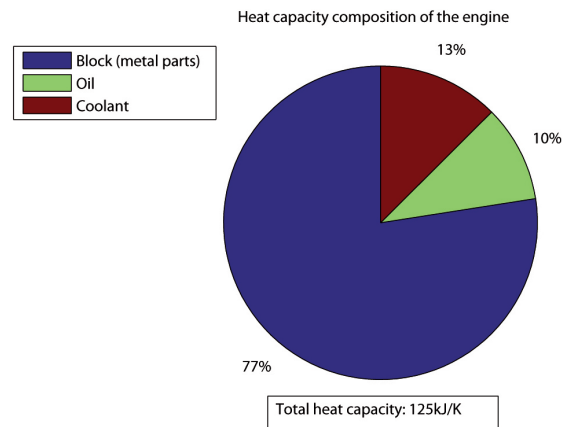


Figure 3.15: The heat capacity repartition between the oil, the coolant and the block

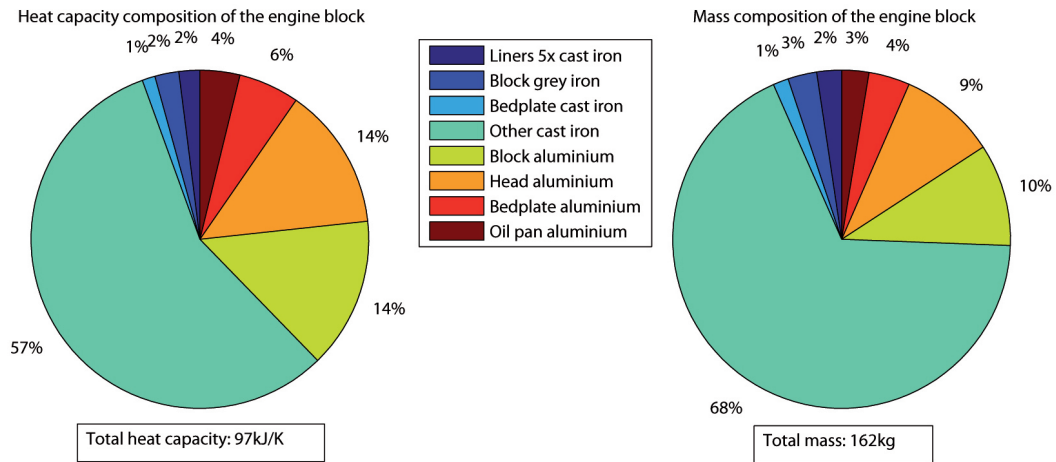


Figure 3.16: The heat capacity repartition between the different block parts

The initial temperature for every step is the previously calculated temperature. Every time the heat is added or removed from the previous state, resulting in a fully dynamic model. The driving style is taken into account at every step.

3.8 Constants map

Some physical properties, like coolant density and heat capacity, vary with temperature. Since the temperature can vary from 0 to 100 degree C during the warming up of the engine, it is important to take these effects into account. Therefore, several maps are implemented to provide the appropriate value to the other submodel and then, improve the accuracy of the model.

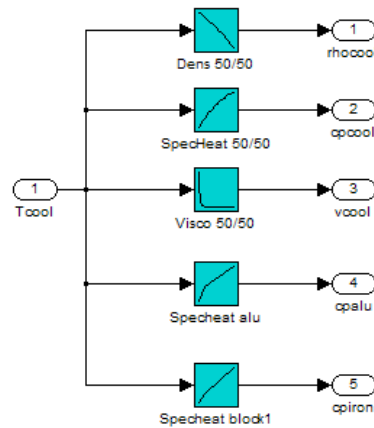


Figure 3.17: The different constant maps used in the model

Chapter 4

Diagnostics

4.1 Algorithm

Different approaches are considered in this study. The modelled and measured temperature can be compared at a single point (figure 4.1) or on an interval (figure 4.2 and 4.3). The first solution is slightly easier but the risk to do the comparison at an unrepresentative point is too high. At some moments the model can be inaccurate due to for example interaction effects of sensor errors. When the algorithm is executed at such a moment, it may give a false diagnosis. Hence, only measured on an interval should be used. Two possibilities are studied: the integral (figure 4.2) and the average (figure 4.3). The integral method adds the temperature difference between the model and the measured value of each time step (in a certain interval). The average method calculates the average temperature difference over a certain interval. In both cases the interval is taken between two predefined temperatures of the model. They are similar regarding the risk of mistake. It is more difficult to find the threshold value for the sum of the difference in the integral method. The average method is easier to use and understand since the threshold will be fixed as a temperature difference. So, the average method is chosen.

This algorithm calculates the average temperature difference within an interval of temperature. If this average is too high, the thermostat is considered as leaking. The interval of temperature is chosen where the model has the highest accuracy. An accuracy study of the model on different temperature intervals is done (figure 4.4). The average difference between measured and modeled temperature is calculated on different time interval for every data set in order to see its accuracy. The model is the best between 70 and 80 degrees in most of cases. This is why the chosen interval used for the diagnostic is 70 to 80 degree. During the literature study other methods were found but were not suitable to this model.

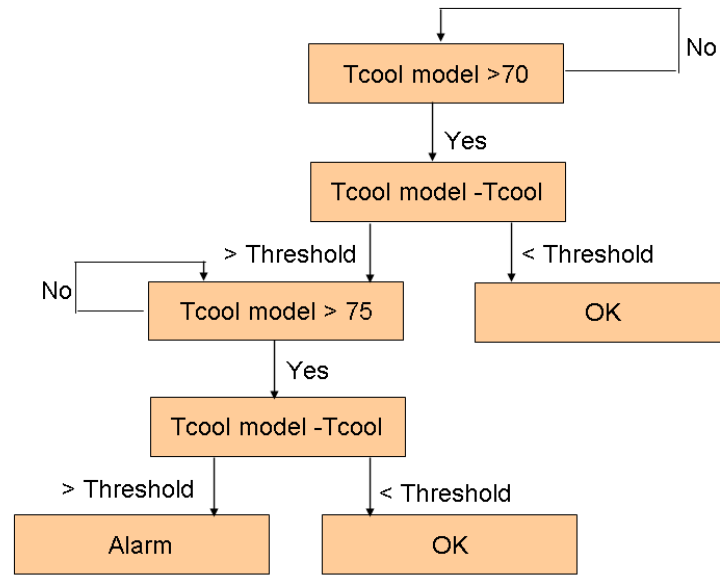


Figure 4.1: Flowchart of the algorithm checking the thermostat status by measuring the temperature at a certain point

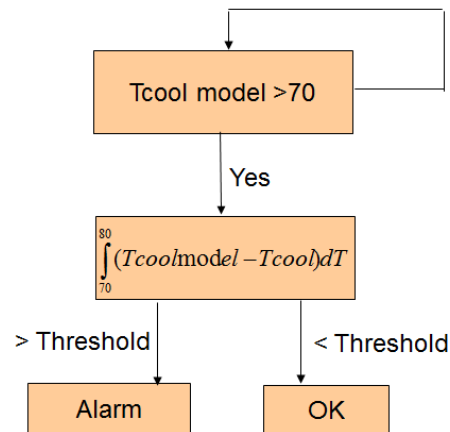


Figure 4.2: Flowchart of the algorithm checking the thermostat status by calculating the sum of the error on a temperature interval

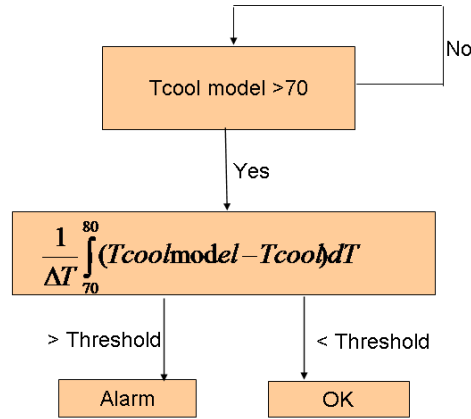


Figure 4.3: Flowchart of the algorithm checking the thermostat status by calculating the average error on a temperature interval

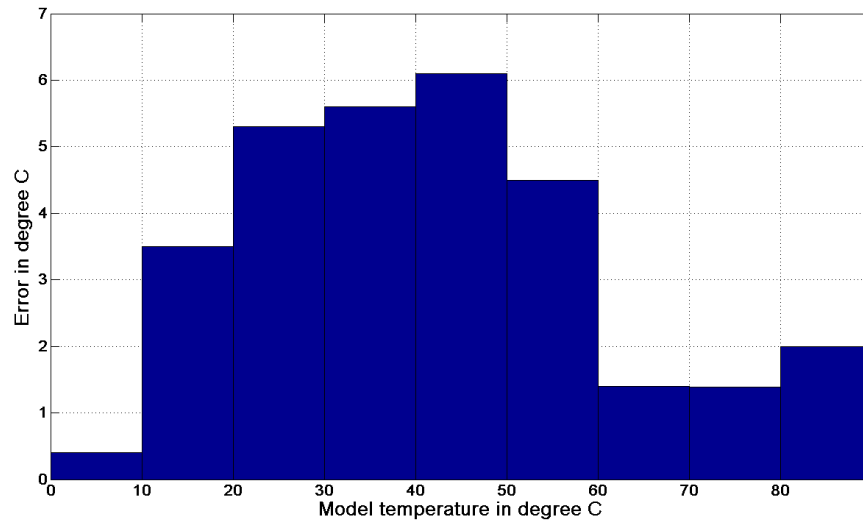


Figure 4.4: Accuracy of the model between 0 and 90 degree C

4.2 Sensitivity study

4.2.1 Model sensitivity

The sensors used as input in the coolant model can be inaccurate or fail. Also the used constants can vary in real life use, like for example fuel properties. In order to study the impact of such problem, a sensitivity study is performed. Every input variable is varied separately for 5, 10 and 20% error. Using every data set

of measurement concerning the I5T engine, an absolute average error is calculated between 70 and 80 degree C (the interval where the leaking thermostat is tested). The results can be seen in the figure 4.5.

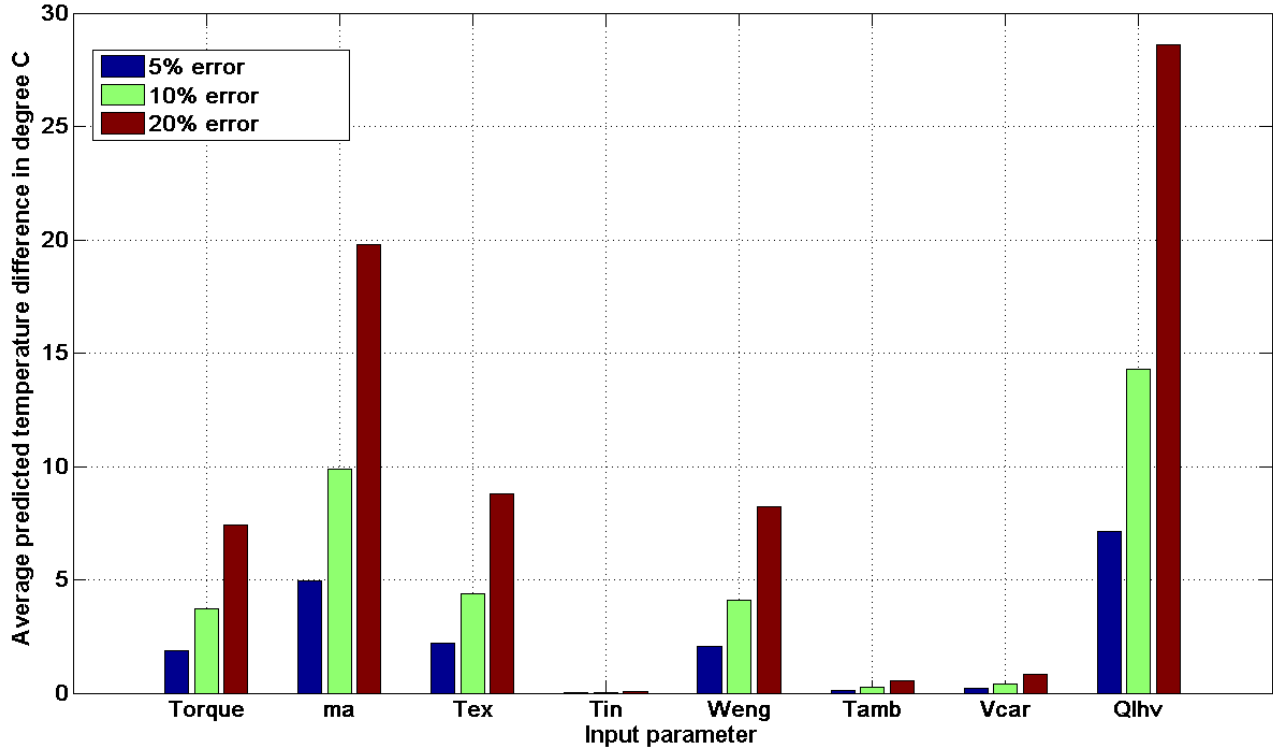


Figure 4.5: Impact of input data error on the predicted temperature in degree C

As can be seen the three less important parameters are the inlet temperature (T_{in}), the ambient temperature (T_{amb}) and the speed of the car (V_{car}). That can mainly be explained by the size of these values and the relative importance of the sub model they are used in. The torque ($Torque$), the engine speed (W_{eng}) and the exhaust temperature (T_{ex}) have a similar importance. The torque is obtained from a model present in the ECU. The accuracy of this model varies between 5 (at low load) and 20 Nm at some operating points.

As can be seen, the two most sensitive parameters are the lower heating value (Q_{LHV}) and the mass air flow (ma) with respectively 14 and 10 degree difference for a 10% error. These parameters are used to calculate the heat produced by the combustion. It becomes important to consider the accuracy and reliability of the air flow sensor. The lower heating value is a characteristic of the fuel and can vary function of the region, the season, the legislation and fuel choice of driver... The statistics regarding the US fuel are used to evaluate the risk of wrong diagnostic due to these variations. Two parameters are influenced by the

fuel quality in the model: the lower heating value and the stoichiometric air/fuel ratio. For instance when the fuel contains a certain amount of ethanol both the lower heating value and the stoichiometric air/fuel ratio will decrease. The figure 4.6 shows the influence of varying the air/fuel ratio and the lower heating value on the result of the model. The average error between the model and the measured data is evaluated between 70 and 80 degree C. The stoichiometric air/fuel ratio of E10 and conventional gasoline are used. The extremes values of the US fuel and of the fuel used by Volvo are used regarding the Q_{lhv}.

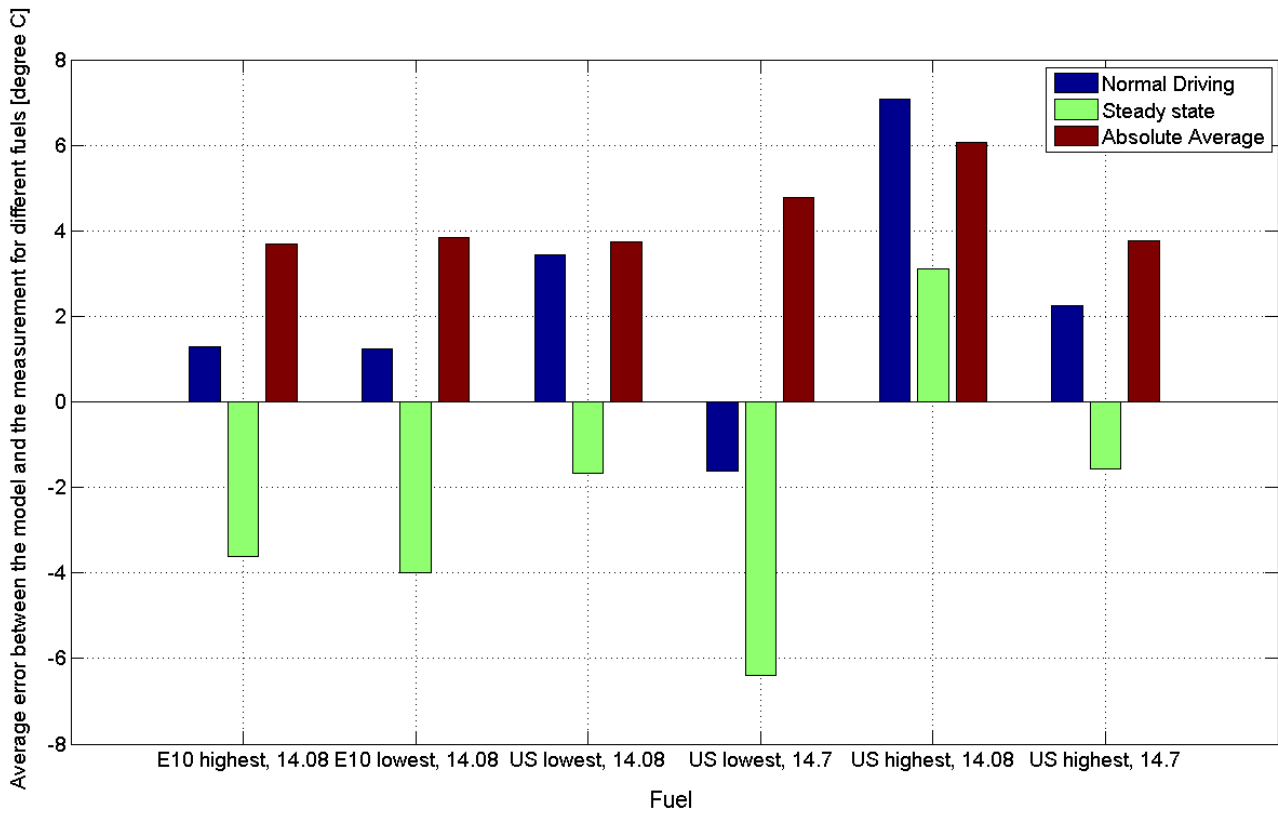


Figure 4.6: Impact of the fuel on the predicted temperature in degree C

The difference between the predicted temperature for the two extremes heat value are shown in table 4.1. Considering the accuracy required for the detection of a leaking thermostat, the small inaccuracy or variation will not perturbate the monitoring system.

Table 4.1: Difference of the predicted temperature (in degree C) between the lowest and the highest heating value for different fuels and stoichiometric air/fuel ratio

Fuel	Difference in degree C
US stoich=14.7	1
US stoich=14.08	2.3
E10 stoich=14.08	0.14

4.2.2 Diagnostic sensitivity

Due to the model sensitivity to input error, the diagnostic algorithm can fail to detect a leaking thermostat. A sensitivity study is done to study the impact of input inaccuracy on the failing thermostat detection. The methodology is similar as the one used for the model sensitivity but instead of measuring the difference between the model and the real temperature, the wrong diagnoses are counted for different input errors.

There are two main diagnostics error states that have to be considered: when the algorithm detects a leaking thermostat which is working properly and when the algorithm doesn't detect a leaking thermostat when it is leaking.

By putting an error on the sensor that increases the model temperature, the algorithm could detect leaking thermostat even if it is not. This study shows that the only critical parameter is the mass airflow. This value needs to have an error less than 8% to be able to perform valid fault detection.

By reducing the model temperature, leaking thermostat could be misdetected. Two parameters are critical in that case: the airflow and the lambda value. A 10% accuracy is required on the mass airflow sensor to avoid misdetection while 8% is required on the lambda value.

Since the accuracy requirement is similar in both error state, the threshold minimize the risk of error in the detection. Increasing the threshold would increase the risk of misdetection and reducing the threshold would increase the risk of detect a normal thermostat as leaking.

Chapter 5

Results and Discussion

5.1 Model validation

5.1.1 Steady state

To validate if the model is accurately predicting the coolant temperature, a couple of steady state warm up measurements is done. These steady state measurements are chosen at various operating points. For the steady state validation, the varied parameters are engine load, engine speed, vehicle speed and ambient temperature. In the text below, a couple of the measurements are explained. The rest of the measurement can be found in appendix A. All the measurements are performed with maximum cabin heating, maximum cabin heater fan speed and with the air conditioning switched on. The figures 5.1 to 5.5 show the results of some steady state comparisons between the new physically based model, the old model and the measured temperature.

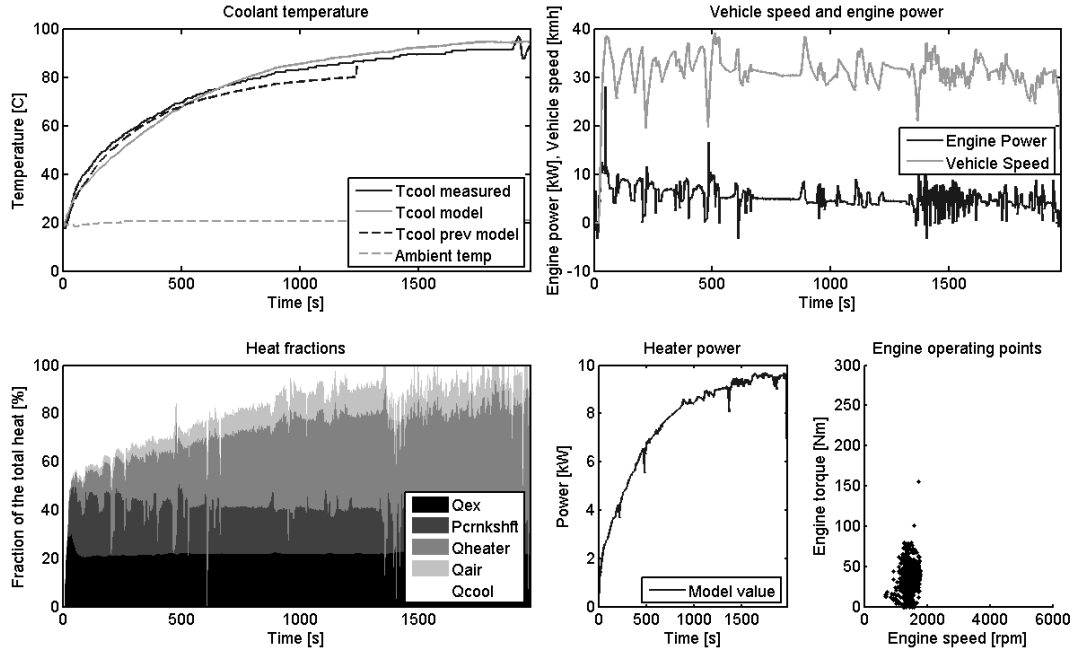


Figure 5.1: Measurement at 30kph and 20Nm

The left top subplot is the main graph and shows the coolant temperatures according to the old model, the new model and the sensor. The right top sub plot shows the vehicle speed and engine output power during the measurement. The left bottom figure shows the energy distribution from the fuel over the different sub models as a function of time. The bottom middle figure indicates the power of the cabin heater. The bottom right figure gives an indication of the engine operating points measured every five seconds. Figure 5.1 shows an engine warm up at low engine load and low vehicle speed. As can be seen from the top left sub plot, the new model is more accurate than the existing model over the whole temperature range. The old model is always considered until the point when it switches to the sensor value after the diagnosis is done. At higher coolant temperatures the temperature is increasing very slowly. This is because at these low engine loads the cabin heater power and heat losses to the surrounding air are relatively big. As coolant temperature increases the heater power and losses to surrounding air also increase. This leaves little heat left to further warm up the coolant.

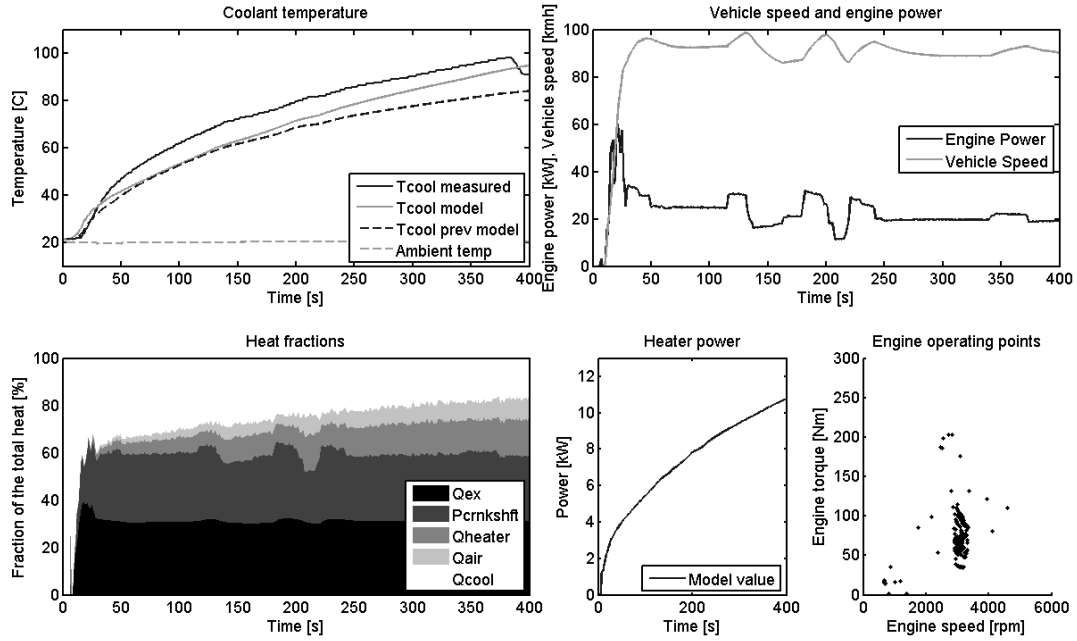


Figure 5.2: Measurement at 90kph

Figure 5.2 shows an engine warm up at higher load and high vehicle speed. The time needed to warm up the engine to operating temperature is much shorter. The new model predict the temperature more accurately than the old model. One can see that the heater power and losses to surrounding air are relatively low when the load on the engine is high.

Figure 5.3 shows a warm up at low load, low vehicle speed and low ambient temperature (-10°C). In these conditions the coolant temperature doesn't exceed 70°C . This is due to the low engine load and low ambient temperatures. The amount of heat withdrawn from the coolant by the cabin heater is extra high due to the low air inlet temperature. The heat losses from engine block to the surrounding air are higher due to the cold surrounding air. This in combination with the low engine load makes the coolant temperature not to rise above 70°C . The predicted temperature by the new model is close to the real temperature though. This is because the above mentioned factors are included in the model.

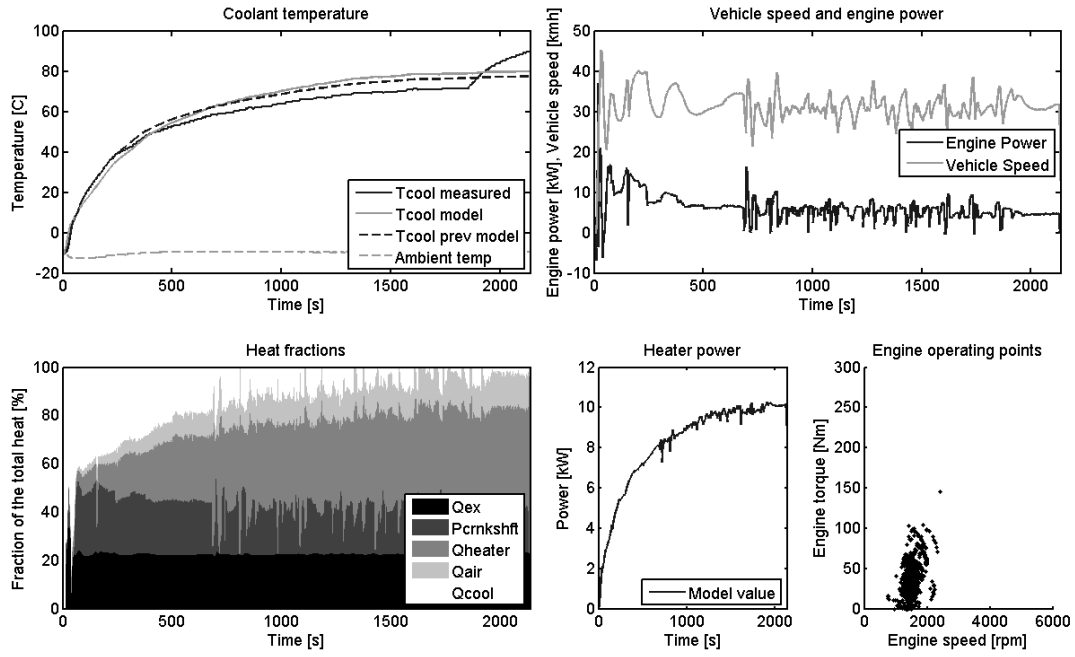


Figure 5.3: Measurement at 30 kph 1500rpm 20Nm -10C

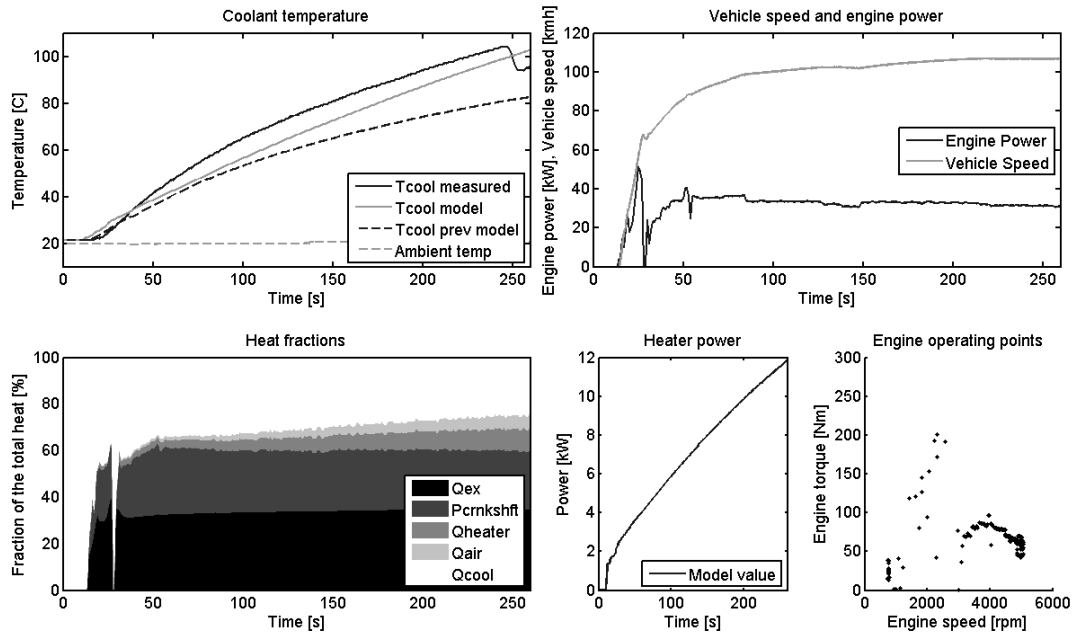


Figure 5.4: Measurement 45Ga 20C 4500rpm 30Nm

A warm up at high engine speed is shown in figure 5.4. Here, the coolant temperature is a bit underestimated but has a smaller error than the original model. The same measurement with low ambient temperature shows the same trend of underestimating the coolant temperature (see appendix A).

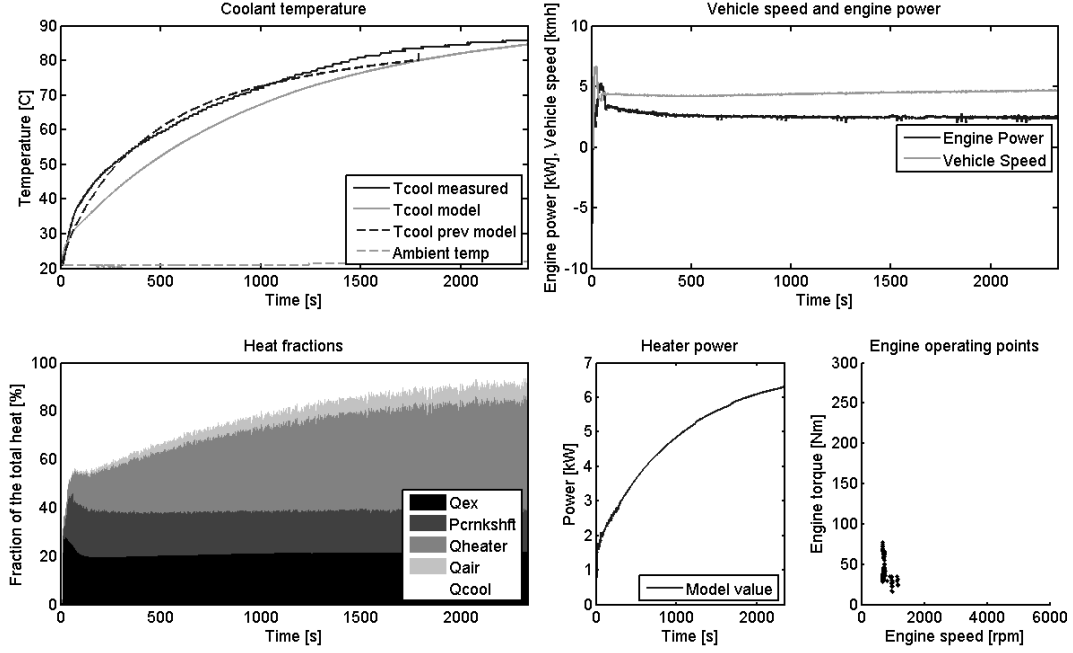


Figure 5.5: Minlast 700rpm 10Nm 20C

Figure 5.5 shows an engine warm up at minimum engine load. This measurement represents driving at engine idle speed and first gear engaged, similar to driving in a traffic jam. The model's prediction in the first part of the warm has quite an error but at higher temperatures the model estimates the coolant temperature more correctly. The new model is a slightly less accurate than the existing model. One can again see that a big part of the energy from the fuel is consumed by the cabin heater.

As can be seen from the figures, the new model fits adequately to the measured values without the use of any correction factors. On average between 70 and 80C, the new model is 3.7 degree better than the old one for these step load measurements (see table 5.1). The average is calculated regarding all the available data. The figures also confirm that the model is the most accurate at the interval between 70 and 80C.

5.1.2 Driving

To validate if the model is accurately predicting the coolant temperature in real life, a couple of warm up measurement during vehicle driving are carried out. These measurements are done with different driving styles and starting temperatures. The driving styles include aggressive driving, normal driving, driving through urban traffic and highway driving. The coolant temperatures with which the measurements were started vary between approximately -10 and +50C. Additionally, measurements were done with many start-stops to verify if the model will work well on future start stop systems or in hybrid power train applications. The figures 5.6 to 5.11 show the results of the comparisons between the new physically based model, the old model and the measured data.

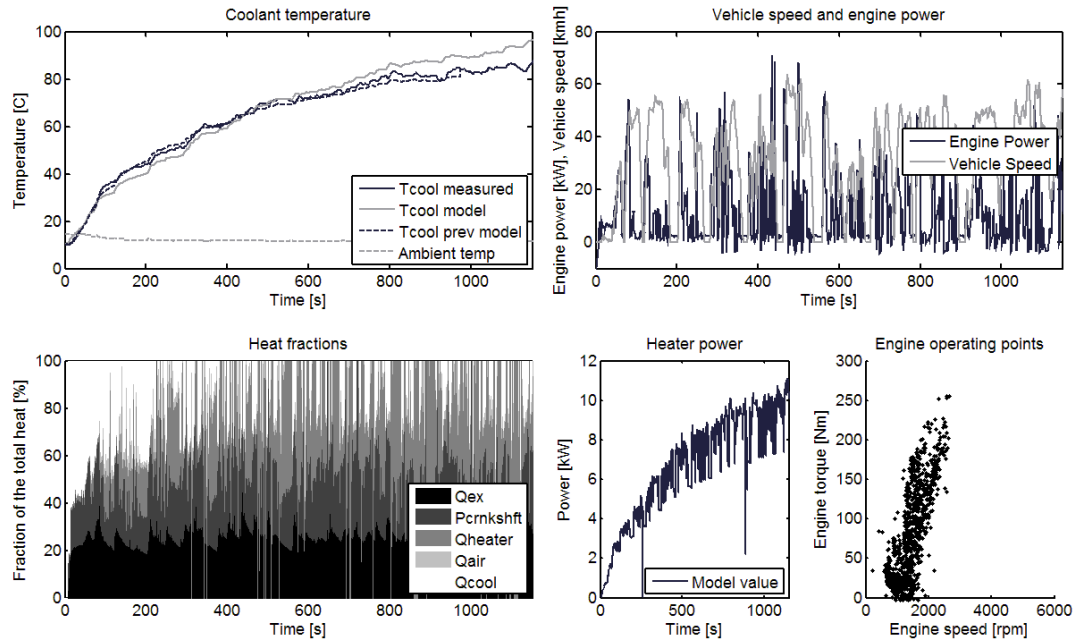


Figure 5.6: City driving

Figure 5.6 shows an engine warm-up while driving through a city. The load on the engine is relatively low and the speed hardly exceeds 60kmh. Because the load on the engine is low, the coolant warms-up slowly. One can see the losses through the cabin heater and to surrounding air are relatively low.

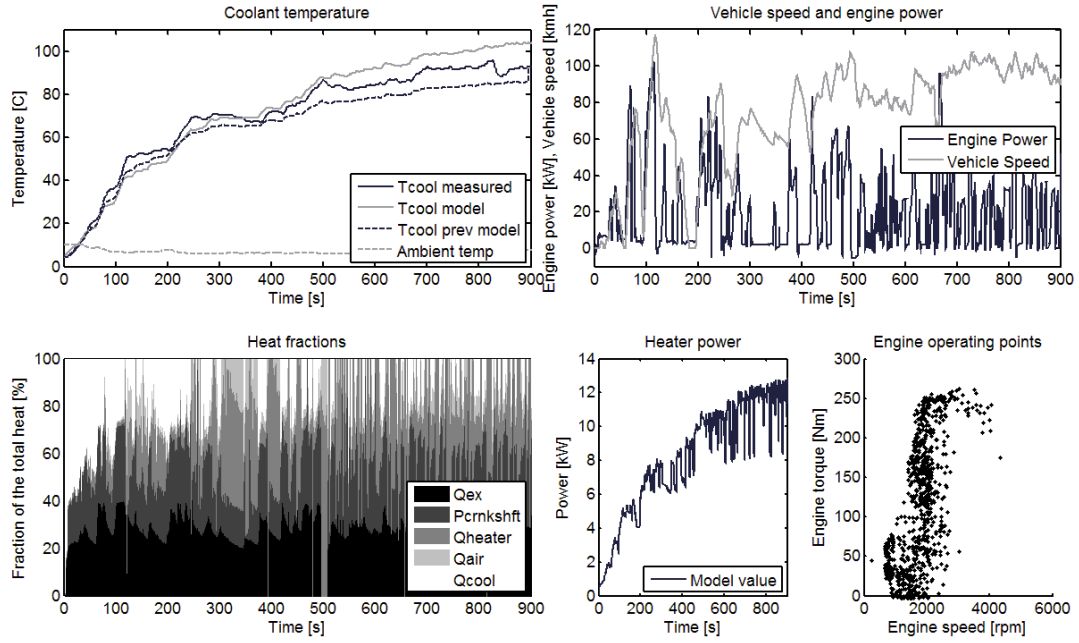


Figure 5.7: Highway driving

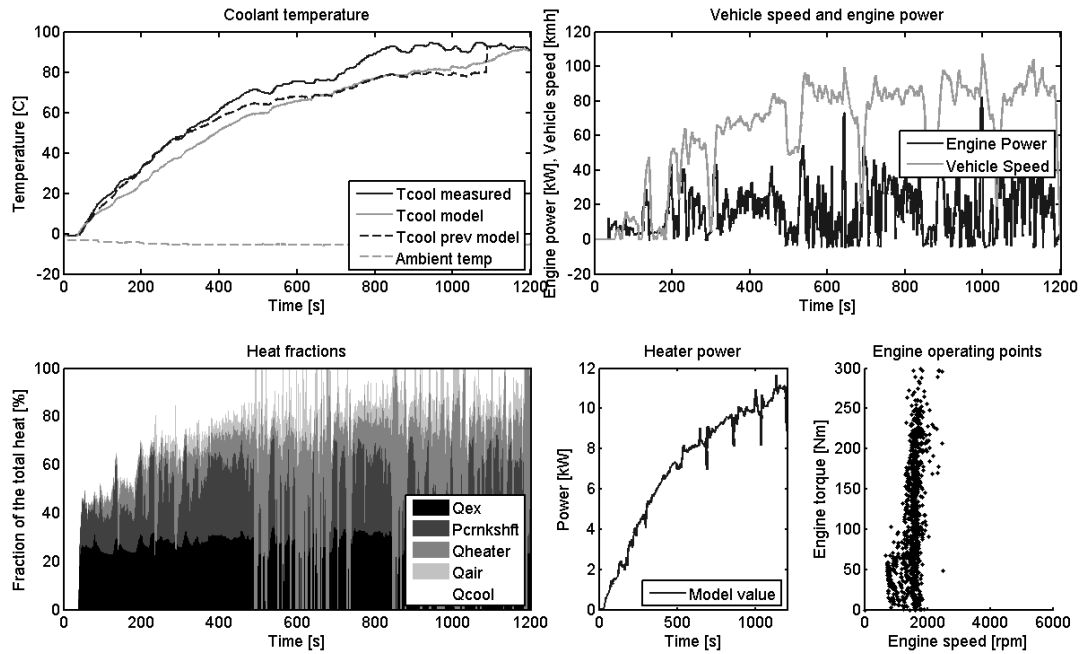


Figure 5.8: Rural driving

Figure 5.7 shows an engine warm-up where shortly after the engine was started, the vehicle was driven on the highway. As can be seen engine load is considerably higher than the load during city driving. This causes the engine to warm up a lot quicker. One can also see that the heat losses to the cabin heater and to surrounding air relatively small due to the higher engine load.

Figure 5.8 shows a warm-up while driving in a rural area. Here the vehicle drives with speeds between 80 and 100kmh with few stand stills. The model is not as accurate as in other measurements.

The graphs in figure 5.9 show a warm-up at low ambient temperature. The temperature is as low as -20C but the coolant temperature is still predicted well. The engine is used at high loads though, which is an advantage for the accuracy of the model.

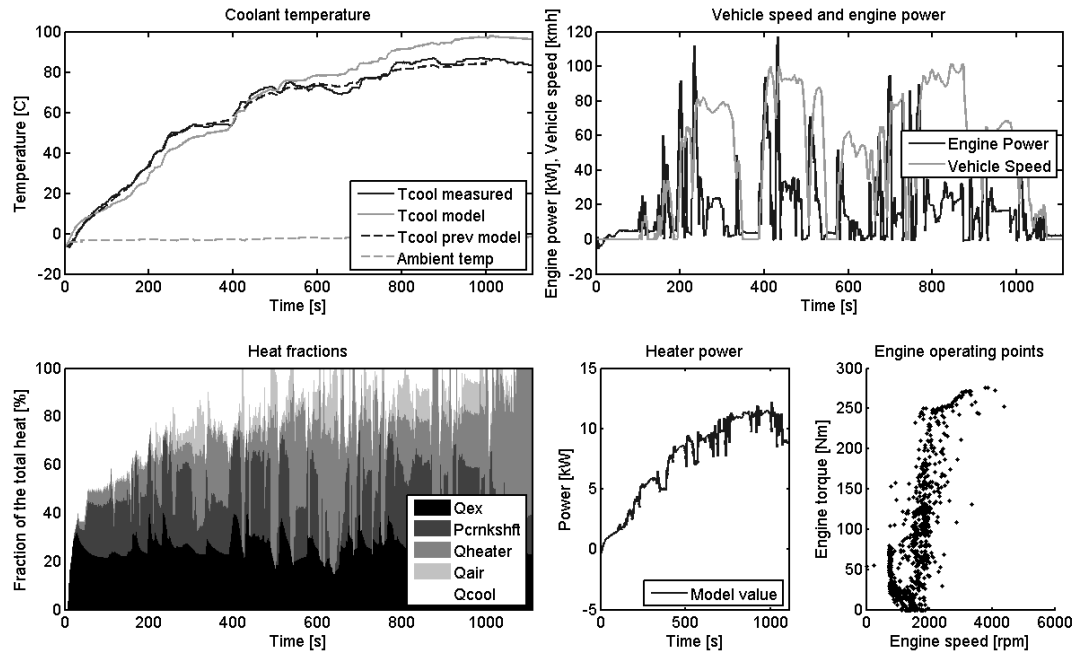


Figure 5.9: Low ambient temperature start

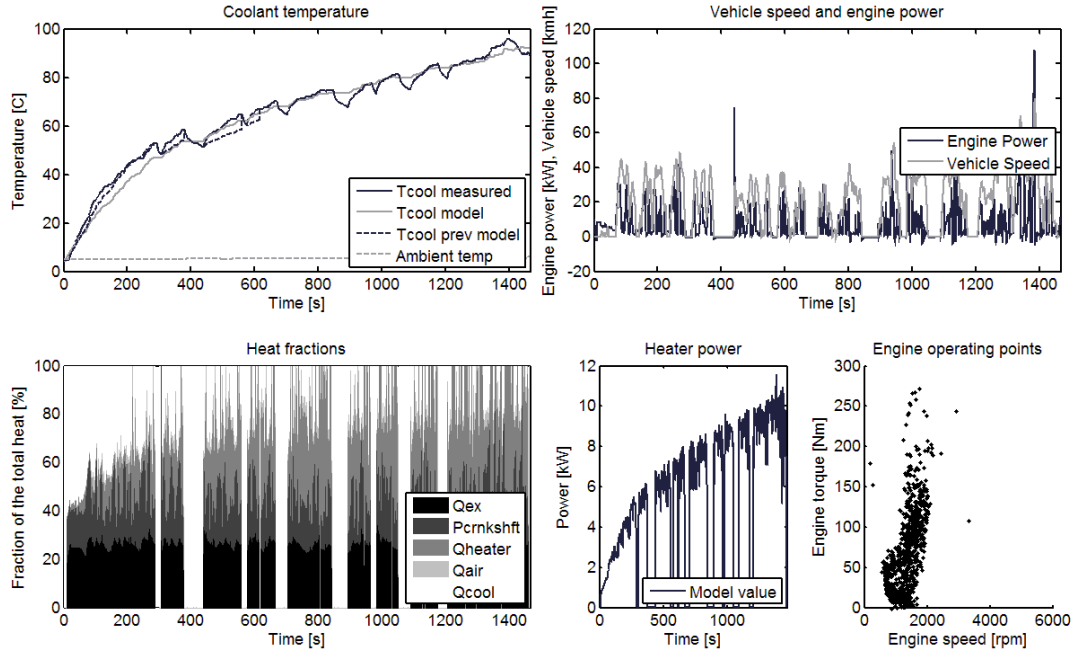


Figure 5.10: City driving with start stop

Figure 5.10 shows how the model reacts on a vehicle with a start stop system. This system stops the engine whenever the car is brought to a stand still. When the brake pedal is released the system starts the engine automatically and as soon as the driver's foot reaches the throttle pedal the engine is running again. One can see that the coolant temperature is predicted very well. One can also see that whenever the engine is stopped, the measured coolant temperature drops very quickly. When the engine is restarted, the temperature rises quickly to its original value though. This is explained by the fact that cold coolant from the external cooling circuit is flowing down into the thermostat housing through the bypass hole. The coolant temperature sensor is mounted close to the thermostat, therefore these temperature "drops" appear. When the engine reached its operating temperature (and the thermostat opens), these "drops" are not present anymore because then the external circuit is filled with hot coolant. In case the heater is on, the temperature also decreases at engine stops but only marginally. By positioning the sensor closer to the engine, the problem will be avoided. In case it remains at the current place, the diagnostic should be disabled when the engine is stopped and enabled a few seconds after the engine starts again.

Figure 5.11 shows a warm-up that starts with a coolant temperature of 40 C. In this case the model works well too, which is expected because it takes a sample from the coolant temperature sensor to determine the start temperature. When this sample is taken at a high temperature, the model has less time to deviate

from reality.

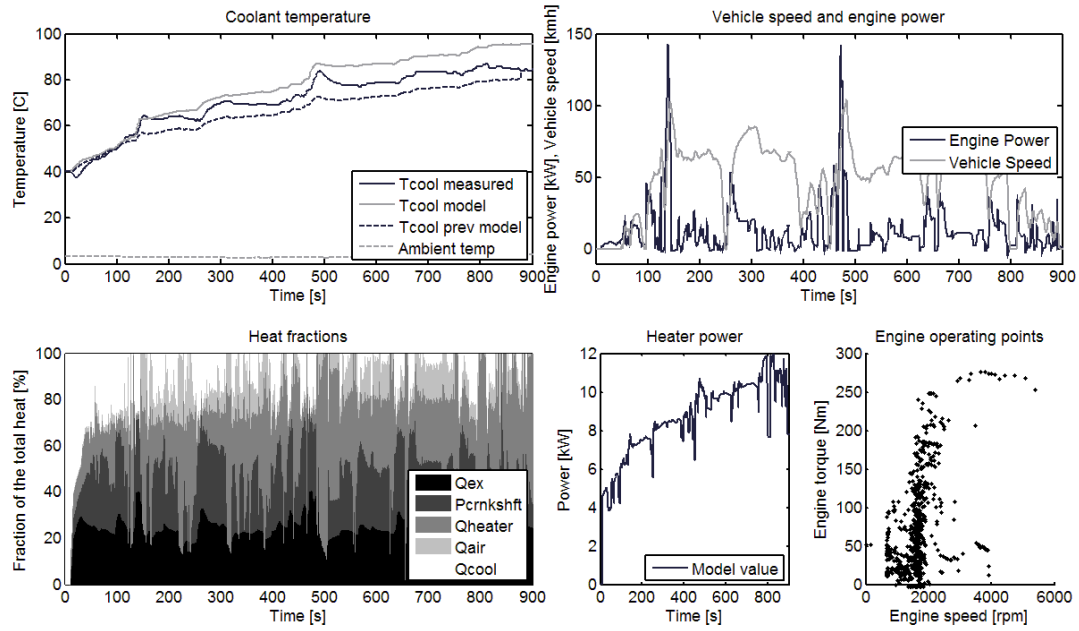


Figure 5.11: Soak start

As can be seen from the figures, the new model fits adequately to the measured values without the use of any correction factors. It is also confirmed that the new model is more accurate (in average for all the measurements) or equivalent (between 70 and 80 degree C) than the old model (table 5.1) in this specific interval.

Table 5.1: Comparison between the new and the old model (improvement in C)

	Between 70 and 80 degree C	Between 0 and 90 degree C
All measurements	1.6	0.5
Driving	-0.2	-1.1
Steady state	3.7	2.6

5.2 Diagnostic validation

Figure 5.12 shows a warm-up during an FTP cycle with both a good thermostat and a leaking thermostat. The solid blue line is the measured coolant temperature with a well functioning thermostat and the dashed blue line the measured coolant temperature with a leaking thermostat. One can see from the graph that in case the thermostat is functioning well, the real coolant temperature and the modeled temperature are almost equal. When the thermostat is leaking though, the measured temperature is a lot lower than the model due to the extra heat loss in the radiator.

The diagnostic function detect the failing thermostat at the vertical dash line. That correspond to a model temperature of 80 degree C.

In figure 5.13 is shown how the engine warms-up over a Unified Cycle Driving Schedule (UCDS). The model appears to estimate the coolant temperature rather accurate. One can see that when the thermostat is leaking, the coolant temperature remains at a low temperature which can also be seen in the FTP cycle.

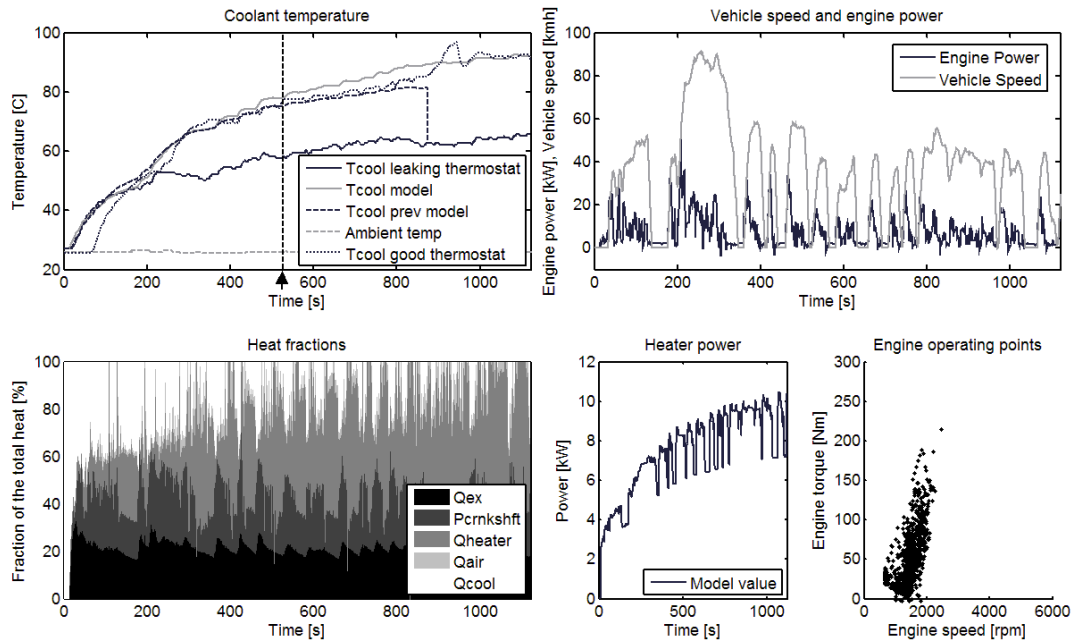


Figure 5.12: FTP driving

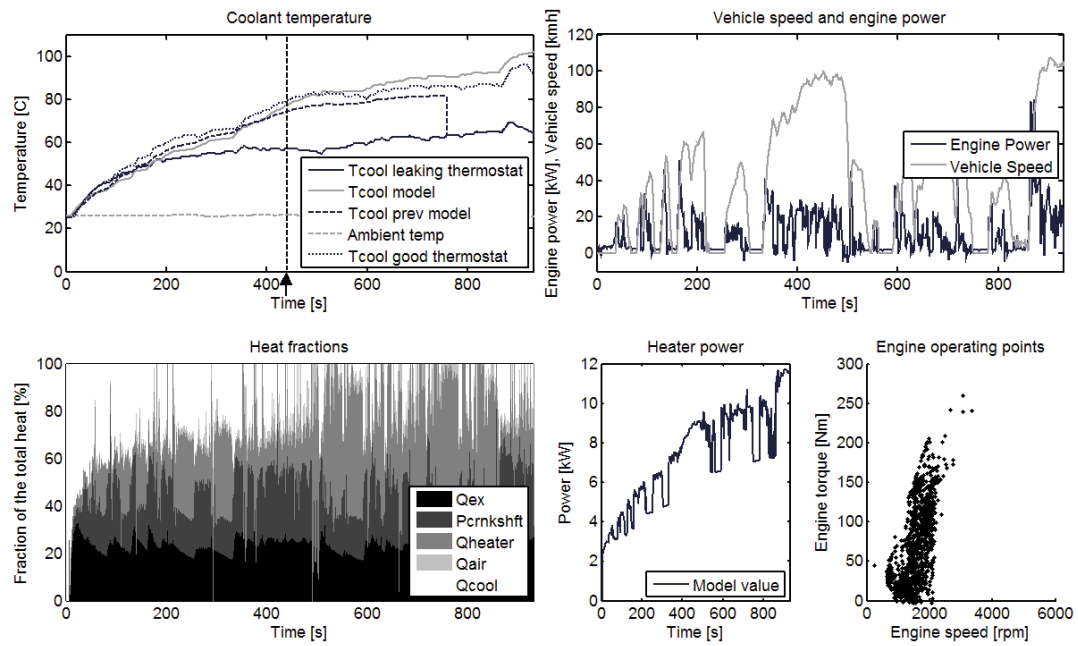


Figure 5.13: UDC driving

Chapter 6

Conclusion

The physically based model to predict the coolant temperature which is developed in this thesis work has proven to be a big step forward in the thermostat diagnostics. The model structure that is chosen turned out to be a good balance of accuracy, complexity, calibratability and applicability.

In chapter 3 is shown that the calibration can be done almost entirely using only component specifications. Only some simple measurements on the vehicle are needed to complete the calibration. This makes the calibration process much easier and saves a lot of time and money.

Chapter 5 shows that the model is more accurate than the current model used and that it performs well in varying environment conditions, customer usage and power train applications.

The model's sensitivity to sensor errors and other noise factors is shown to be low enough in chapter 4. The diagnostic function for detecting a failing thermostat is proven to have a low rate of misdetection. The proposed threshold for the diagnostic function is shown to be the best value to both minimize failing to detect a leaking thermostat and diagnose a good thermostat as leaking.

The modular structure allows the model to be adapted for hybrid power train applications which will be more crucial in the future. This adaptation of the model to a hybrid power train will be the next challenge of thermostat diagnostics.

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List of Symbols and Abbreviations

Abbreviation	Description	Unit
A	object surface	m^2
C, c	heat capacity	$J.g^{-1}.K^{-1}$
c_p	heat capacity constant pressure	$J.g^{-1}.K^{-1}$
d	hydraulic diameter	m
g	gravitation constant	$m.s^{-2}$
h	heat transfer coefficient	$W.K^{-1}$
H_{evap}	heat needed for evaporation of injected fuel	J
k	conduction coefficient	$W.K^{-1}.m^{-1}$
l	length of body	m
L	characteristic length	m
LHV	lower heat value	$J.g^{-1}$
m	mass	g
$\dot{m}_a$	air mass flow	$g.s^{-1}$
$\dot{m}_{exhaust}$	exhaust gas mass flow	$g.s^{-1}$
$\dot{m}_f$	fuel mass flow	$g.s^{-1}$
Nu	Nusselt number	-
P	power	W
P_{out}	mechanical output power	W
Pr	Prandtl number	-
Q	added heat	J
q	local heat flux	J
$Q_{comb(ustion)}$	heat from combustion	W
$Q_{cool(ant)}$	heat to coolant	W
$Q_{coolantToBlock}$	heat from coolant to engine block	W
$Q_{coolToair}$	heat from coolant to surrounding air	W
$\dot{Q}$	heat flow	W
$Q_{exh(aust)}$	heat to exhaust	W
$Q_{friction}$	heat from internal friction	W
Q_{fuel}	heat from fuel	W

Abbreviation	Description	Unit
Q_{heater}	heat to cabin heater	W
$Q_{manifoldToblock}$	heat from exhaust manifold to engine block	W
Q_{oil}	heat to engine oil	W
$Q_{oilToblock}$	heat from engine oil to engine block	W
$Q_{oilToCool}$	heat from engine oil to coolant	W
$Q_{surrounding}$	heat engine block to surrounding air	W
$Q_{toblock}$	heat to engine block	W
Q_{turbo}	heat from turbo water cooling	W
Q_{wall}	heat into cylinder wall	W
Re	Reynolds number	-
S	engine outer surface	m^2
$stoich$	stoichiometric air/fuel ratio	-
t_{sample}	Sample time	s
T	engine torque	nm
T_0	temperature coolant previous iteration	K
T_{∞}	temperature far from wall	K
T_{air}	engine's surrounding air temperature	K
T_{airin}	temperature air heater core inlet	K
T_c	coolant temperature	K
T_{c1}	temperature coolant at time 1	K
T_{c2}	temperature coolant at time 2	K
T_{coolin}	temperature coolant heater core inlet	K
T_{env}	environment temperature	K
T_{exh}	exhaust gas temperature	K
T_{in}	inlet air temperature after CAC	K
T_s	surface temperature	K
U	internal energy	J
V	mean flow speed	$m.s^{-1}$
W	work	J
α	thermal diffusivity	$m^2.s^{-1}$
β	thermal expansion coefficient	$m.K^{-1}$
ϵ	emissivity of material	-
η	combustion efficiency	-
λ	excess air number	-
μ	dynamic viscosity	$N.s.m^{-2}$
ν	kinematic viscosity	$m^2.s^{-1}$
ρ	fluid density	$kg.m^{-3}$
ω_{eng}	angular velocity engine	$rad.s^{-1}$

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Appendices

Graphs

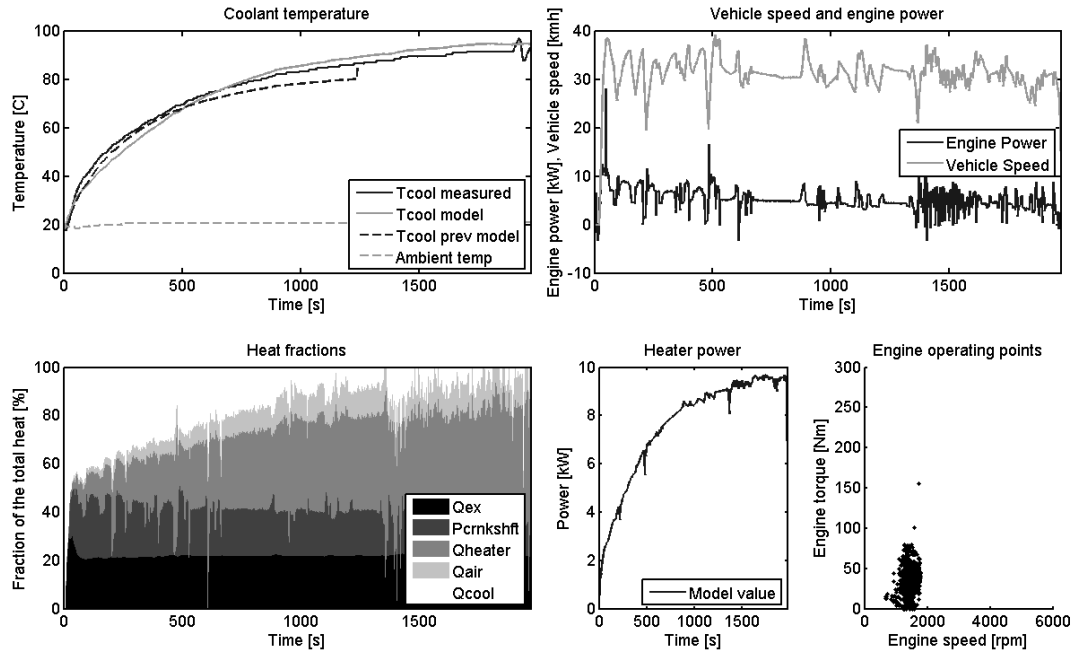


Figure 1: Measurement at 30kph and 20Nm

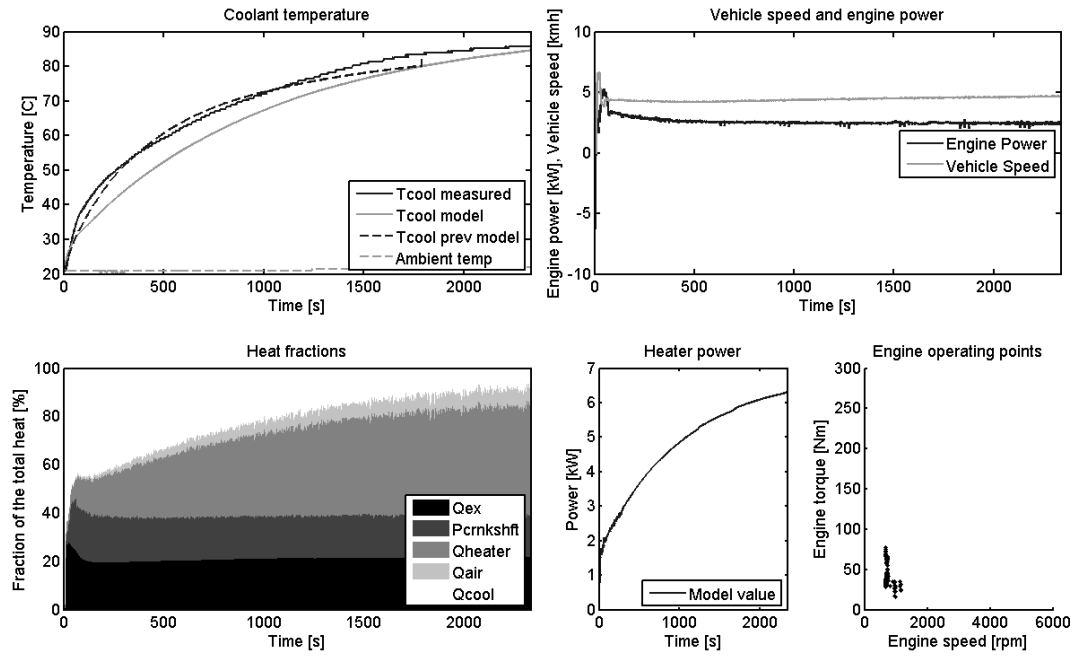


Figure 2: Measurement at idle minimum load

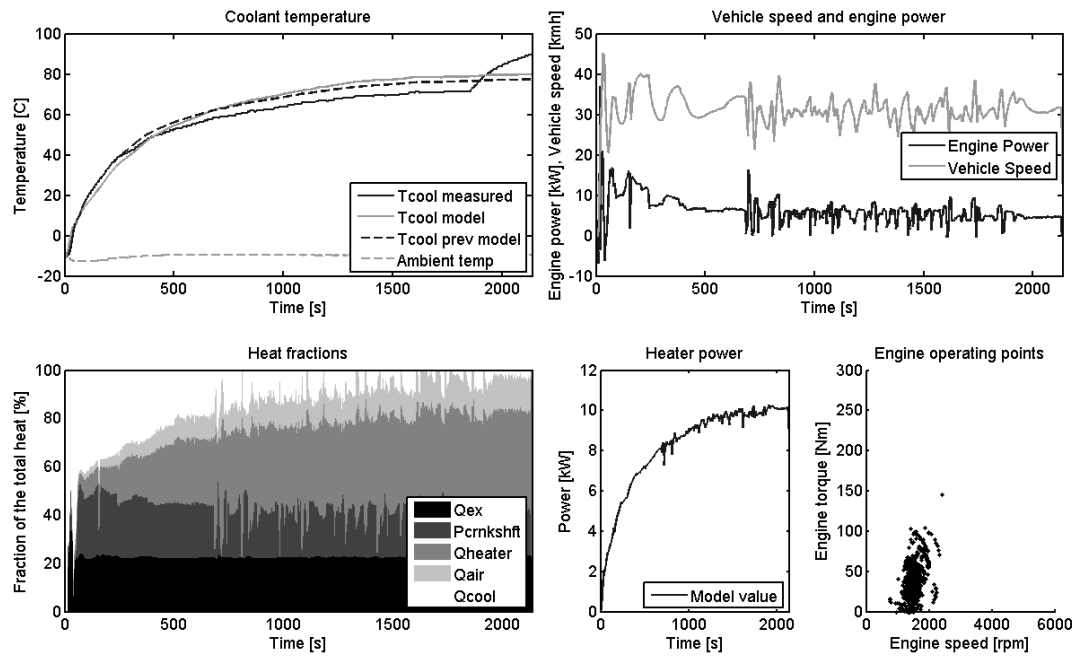


Figure 3: Measurement at 30kph and -10 degree C

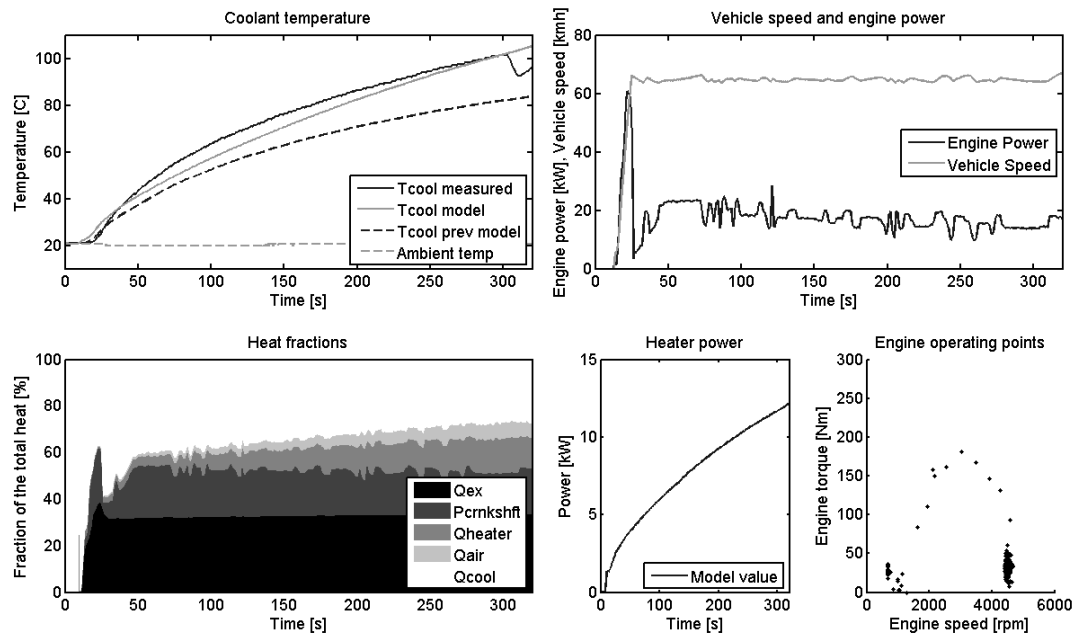


Figure 4: Measurement at 4500 rpm

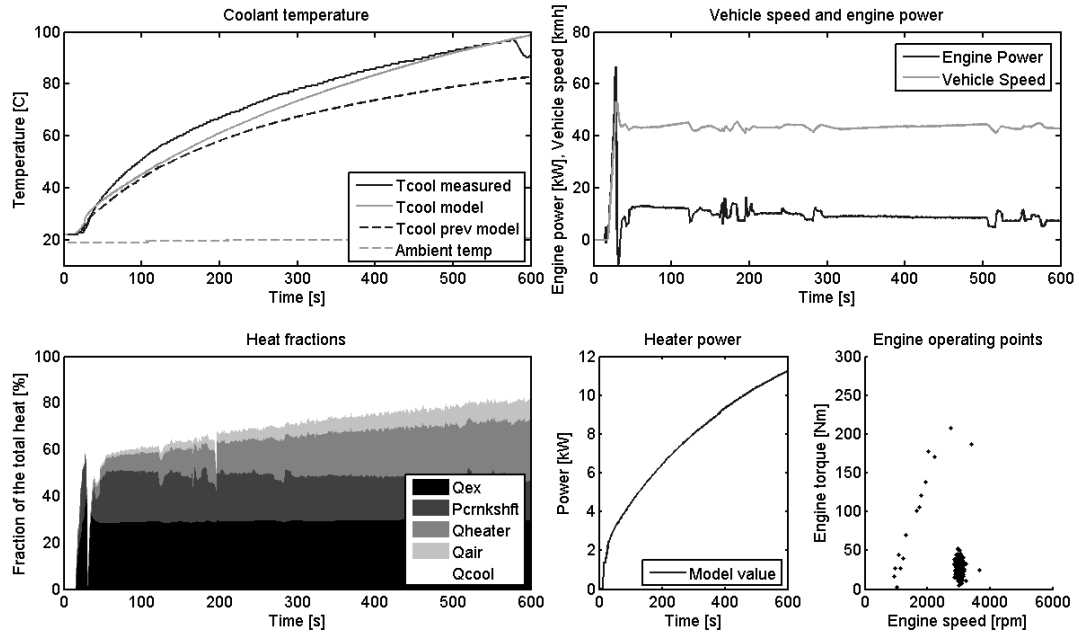


Figure 5: Measurement at 3000rpm

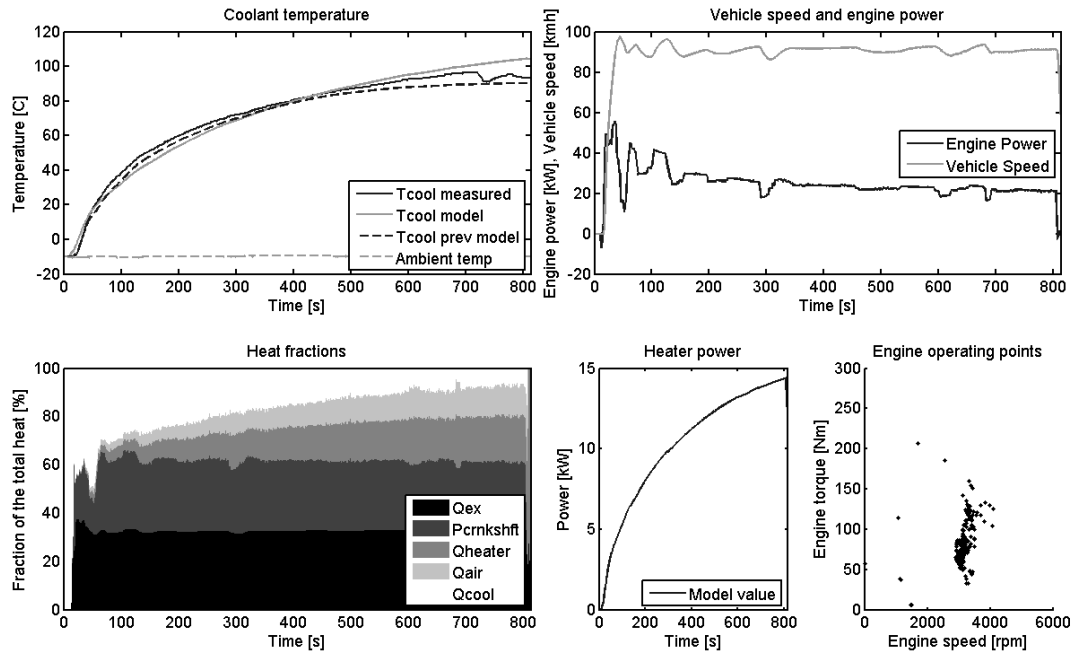


Figure 6: Measurement at 90kph and -10 degree C

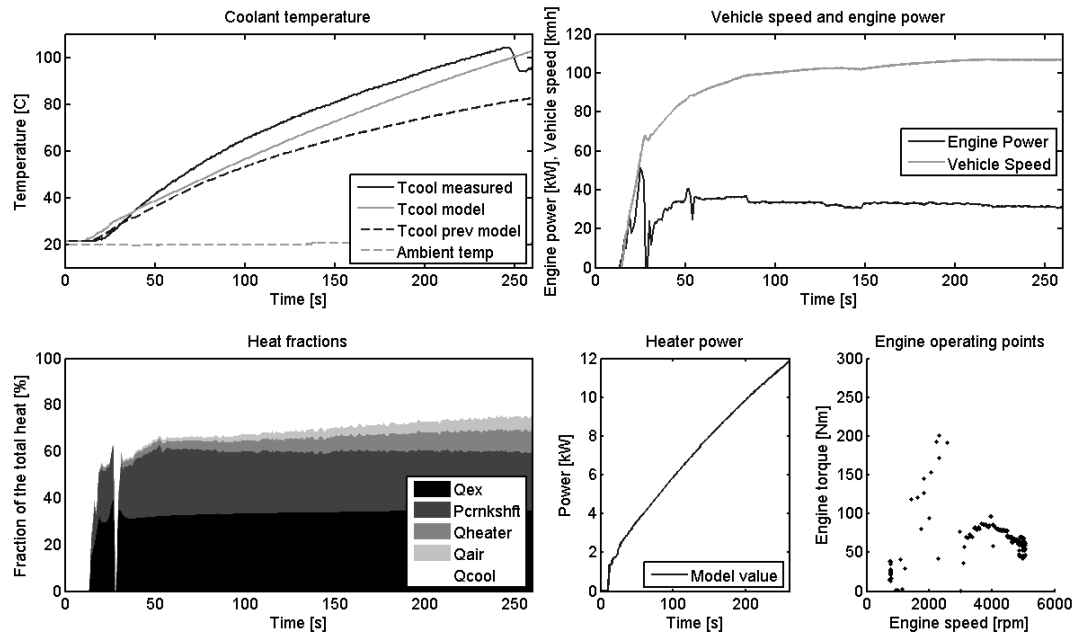


Figure 7: Measurement at 100kph and medium load

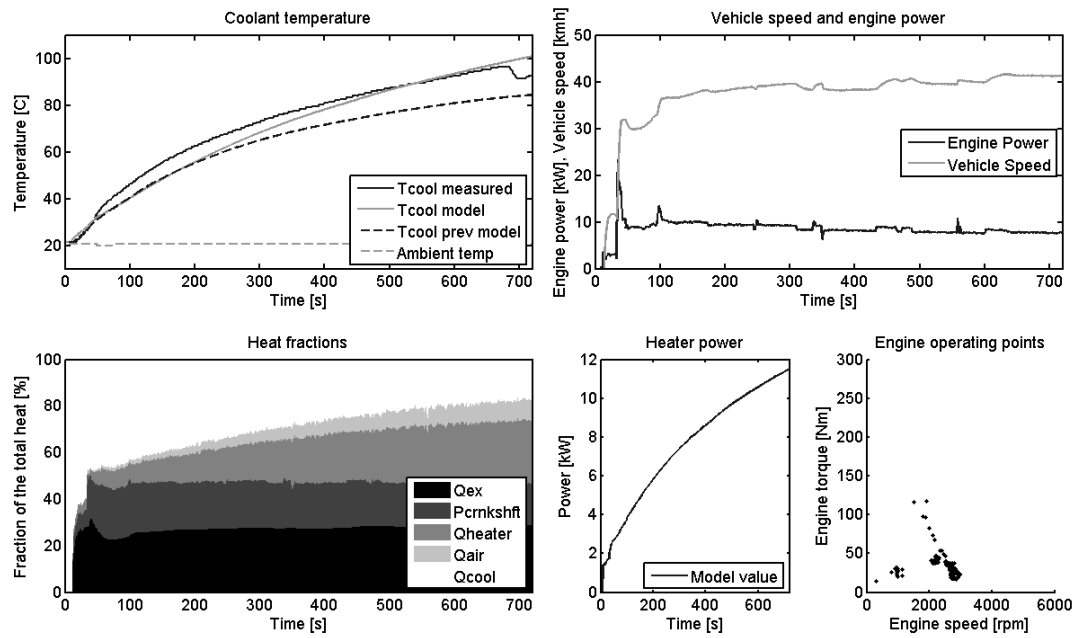


Figure 8: Measurement at 40kph and 40Nm

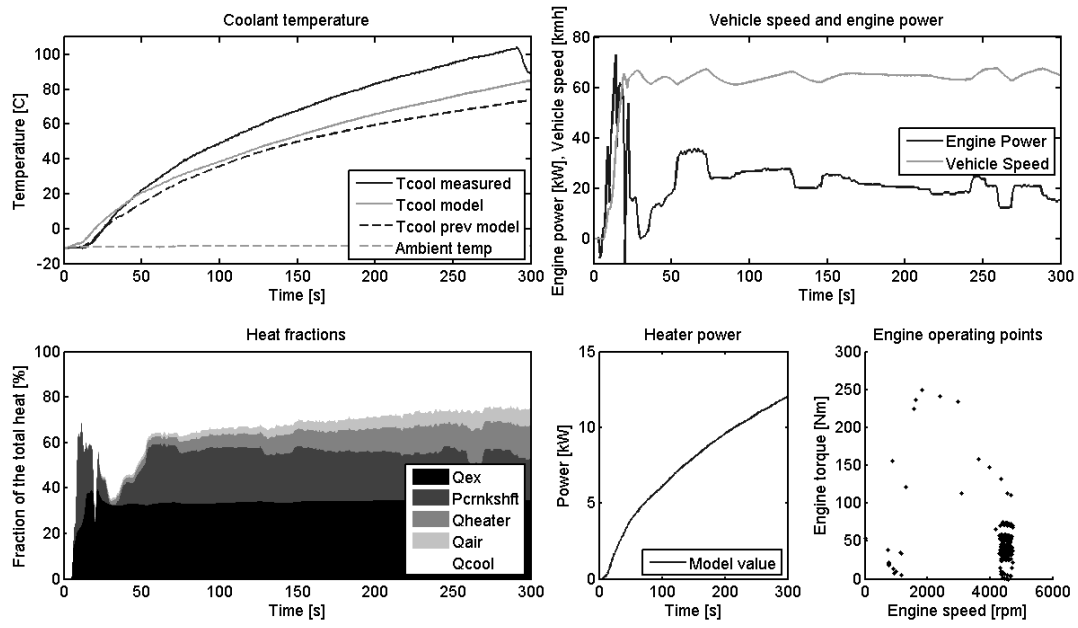


Figure 9: Measurement at 4500rpm and -10 degree C

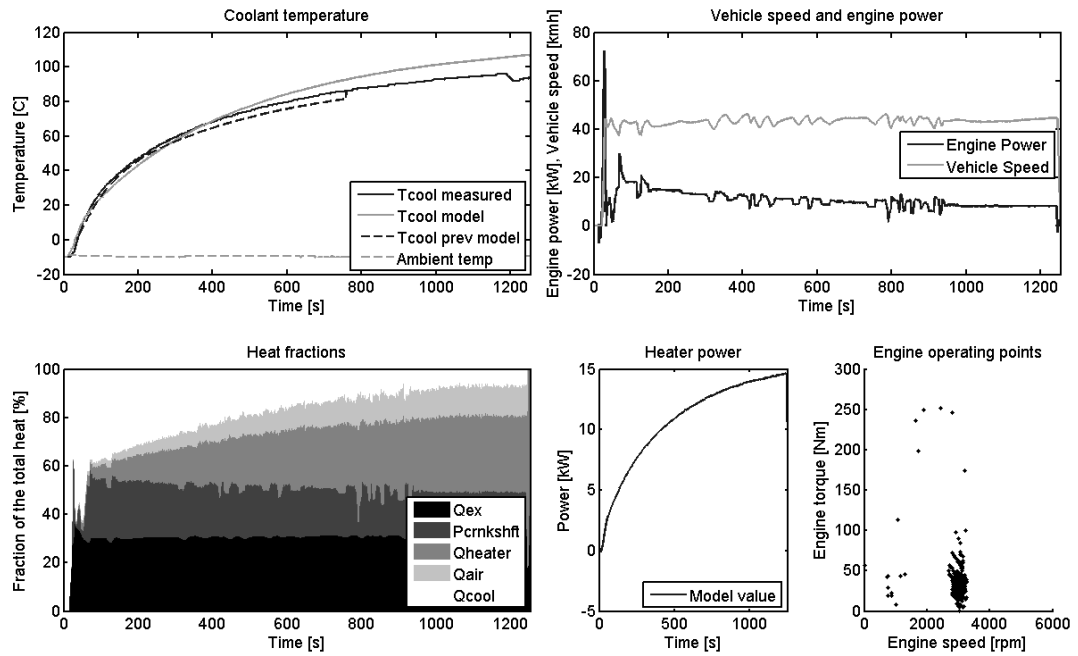


Figure 10: Measurement at 3000rpm and -10 degree C

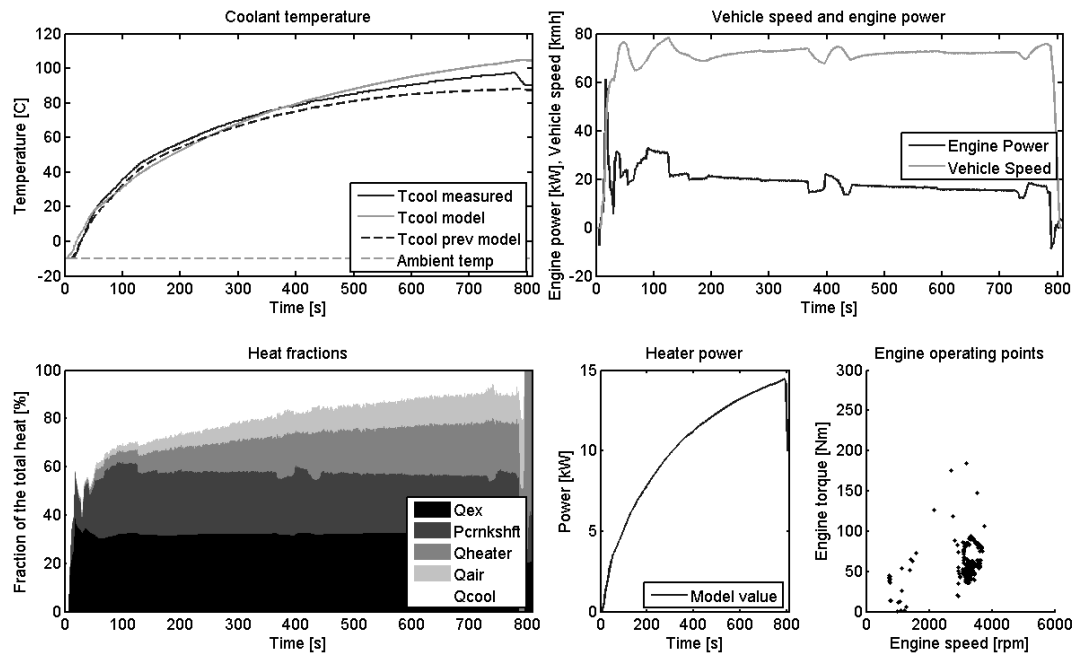


Figure 11: Measurement at 70kph and -10 degree

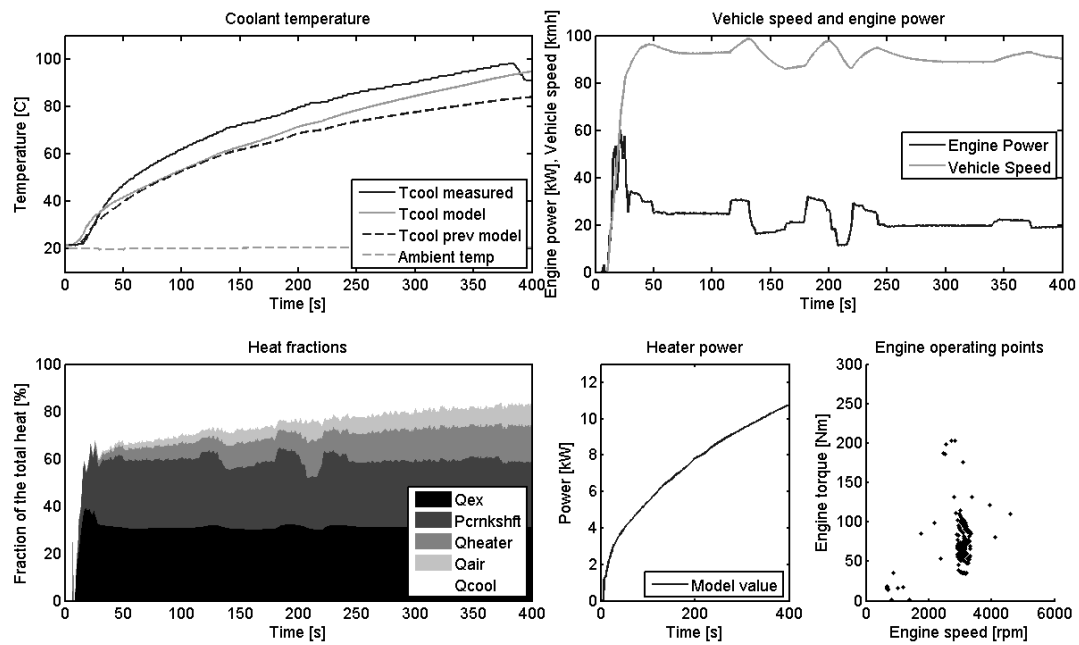


Figure 12: Measurement at 90kph

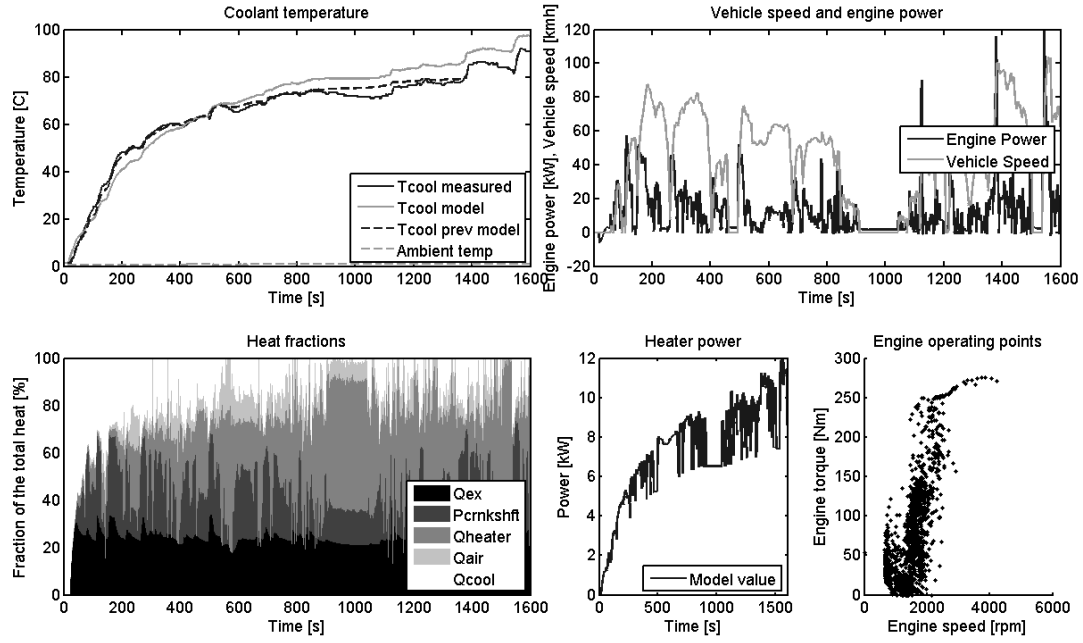


Figure 13: Measurement in driving condition

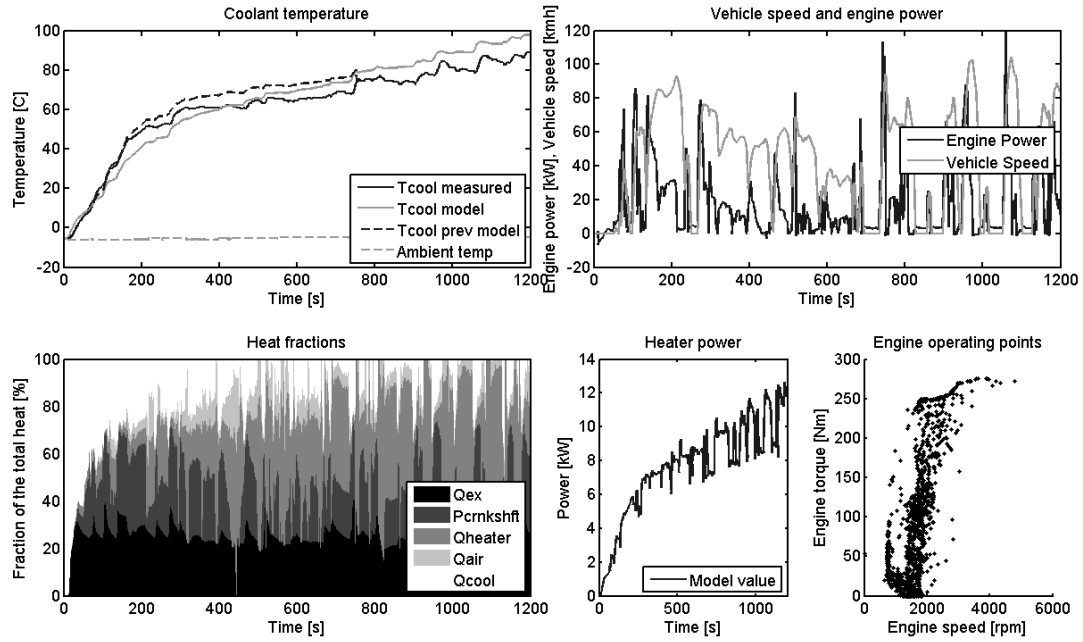


Figure 14: Measurement in driving condition

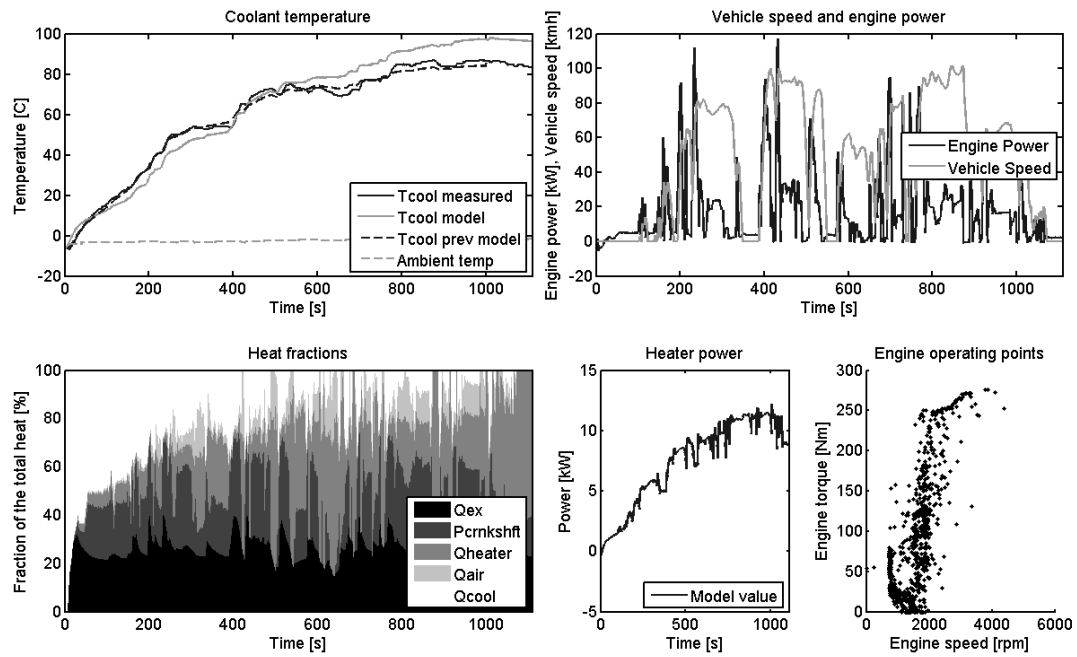


Figure 15: Measurement in driving condition

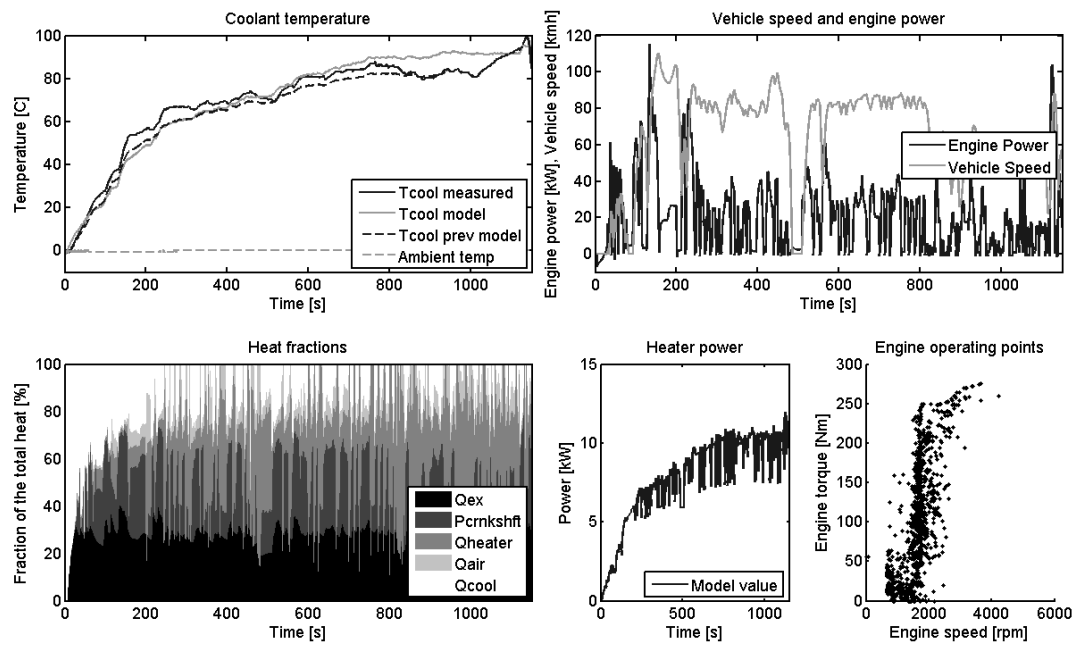


Figure 16: Measurement in driving condition

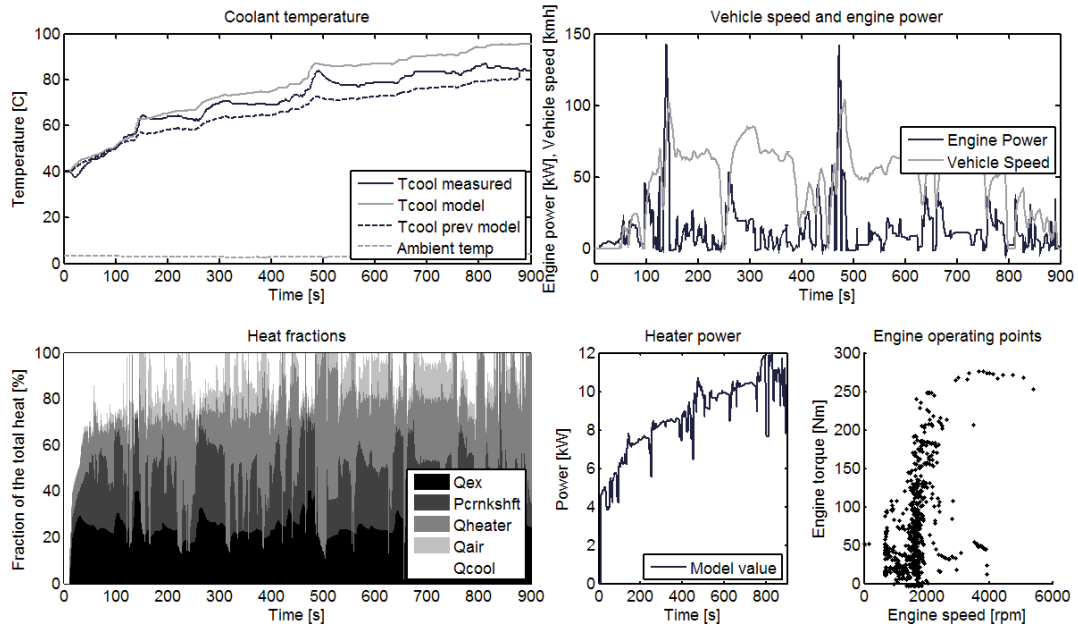


Figure 17: Measurement in driving condition

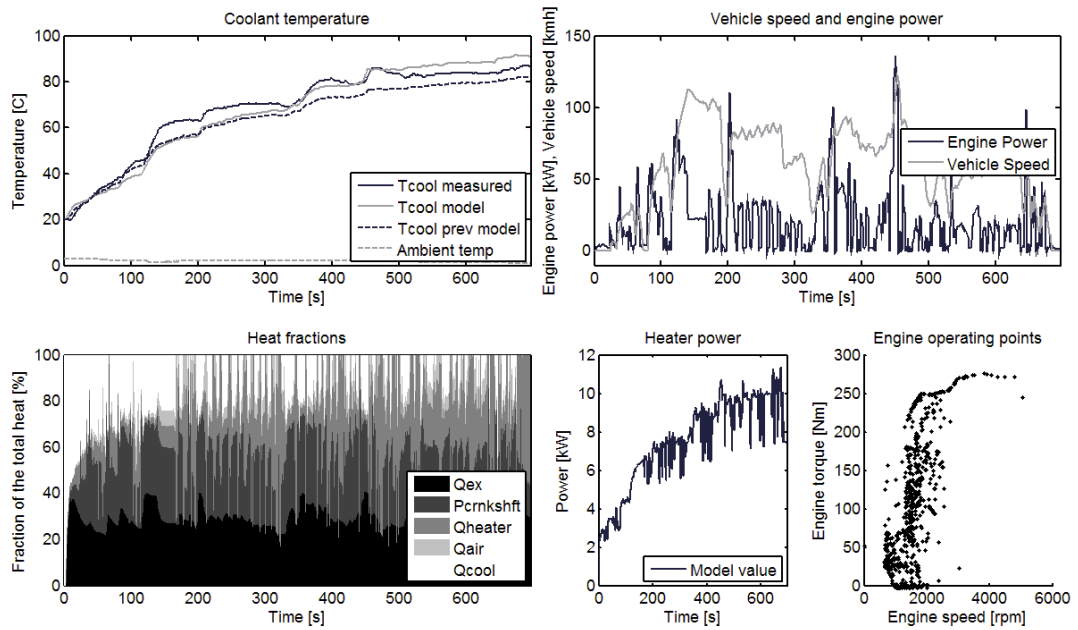


Figure 18: Measurement in driving condition