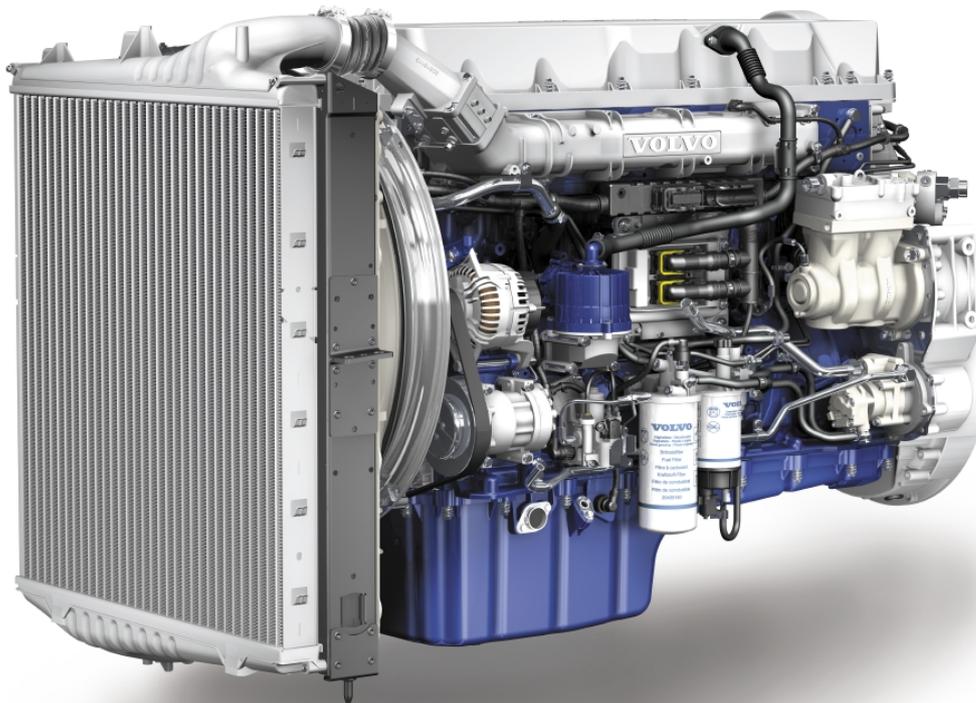


CHALMERS



Study of Software System Integration for Transient Simulation of Future Cooling System for Heavy Truck Applications

Master's thesis in Automotive Engineering

BLAGO MINOVSKI

Department of Applied Mechanics
Division of Vehicle Engineering and Autonomous Systems
CHALMERS UNIVERSITY OF TECHNOLOGY
Göteborg, Sweden 2013
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Volvo E5 13L engine

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ABSTRACT

The work investigates the integration between tools for analysis and simulation of cooling system at Volvo Group Trucks Technology. At the same time it is a consequent step in evaluating GT-SUITE as a tool for analysis and simulation of cooling systems.

The interaction between the truck's cooling system and the engine has so far not been modeled in its entirety in a completely integrated, detailed model capable of simulating transient runs. This work marks the first steps in this direction.

A number of different methods for modeling cooling system performance exist. This project focuses on 1D simulation tools, which are generally preferred in the context of transient simulations of power train and engine installation systems.

The Cooling Analysis and Simulations group at Volvo Group Trucks Technology uses KULI as 1D simulation tool for analysis of cooling performance. Volvo Powertrain, on the other hand, uses GT-SUITE for engine simulations. It is expected to improve the quality of the simulation, (i.e the accuracy of the results) and improve system integration by using one tool for both areas of simulation.

GT-SUITE is a powerful tool for modeling the fluid-dynamics and heat-transfer phenomena, which occur in the cooling system. The work includes a detailed model of the main coolant circuit since this is vital for the authenticity of the transient simulation.

This thesis is a natural continuation of a project performed by the author during the summer of 2012, which evaluated GT-SUITE as a tool for steady-state simulation of cooling systems[13]. The basic models of the cooler package were developed and calibrated during this project.

This work delivers two transient models of FH 13liter Euro 6 cooling system integrated with a predictive engine model, provided by Volvo Powertrain.

As a first step, all necessary models and control algorithms were obtained from different technical units within the organization and were further reworked and refined to fit the purpose of this work. Such are the engine model obtained from BF66360 System Analysis and Simulation at Volvo Powertrain, the fan control model provided by BF72362 Cooling Systems at Powertrain Installation and the coolant pump control model from BF69317 Vehicle Functions. Component performance data was as well acquired for different components within the coolant circuit: thermostat valve, coolant pump, engine oil cooler, etc.

GEM3D was used to ease the process of creating authentic models of the coolant logistic components (pipes, hoses, flowsplits, etc)

The first model produced in this work, delivers a basic representation of the physics in the system and its main aim is to prove the technical feasibility of the concept. It includes all necessary functional features : working fan control, coolant pump mode control, function for temperature compensation for the effect of air recirculation. The model was used to obtain critical cooling system-related parameters from Hamburg-Kassel drive cycle and critical parameters were compared to measurements from tests. The results, acquired from the first model have satisfactory consistency with the data from test. Average coefficient of determination achieved by the model is $R^2 > 0.85$, which for most of the parameters is higher than the results given by the currently available program for transient simulations. The rate of execution of the model was successfully increased to 1 x Real Time.

The second model implements two-way communication between the engine model and the cooling system model: the temperature on the outlet of the CAC and torque consumed by coolant pump and fan are fed back to the engine model and therefore the interaction between the performance of the two separate systems is partially accounted for. Average coefficients of determination achieved from this model are similar to the ones from the model with single connectivity. Execution time rate did not alter significantly in comparison to the previous model. The implemented interaction between the subsystems allows for a predictive model of the boost temperature and investigations on fuel economy.

The work has proven the feasibility and the integrability of an engine model and a model of a cooling system in GT being controlled in Simulink environment. This paper can be seen as a comprehensive manual to building, tuning and executing such a model.

Keywords: 1D cooling simulations, transient simulations, GT-SUITE, Matlab, Simulink, engine models, trucks, calibration

PREFACE

This master thesis work is performed from January to June 2013 at Volvo Trucks Group Technology in Gothenburg, Sweden as a final work completing a Master's degree in Automotive Engineering at Chalmers University of Technology. The project is done in BF72363 Simulation and testing department at Volvo Group Trucks Technology in cooperation with BF66360 System Analysis and Simulation at Volvo Powertrain and BF72362 Cooling Systems at Powertrain Installation.

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I would like to thank all specialists at Volvo Trucks, who provided support to this project: Elias Johansson for his valuable help with setting up the engine model, Christoffer Bäckvall for his help with the fan control software, Stina Karlsson, Anna Rimark, who provided me with useful data about the coolant pump and coolant circuit, Anders Ottosson and Björn Andre for their advice regarding physical testing and Dennis Engberg, who solved many of my IT issues.

A special thanks goes to my examiner at Chalmers, Prof. Lennart Löfdahl for the swift responses to my questions, his trust and vision.

Last, but not least I would like to thank Gamma Technologies and specially Jan Böbel for providing extremely competent support and advice.

ABBREVIATIONS

CAC	Charge air cooler
ECU	Electronic control unit
EMS	Engine management system
FMEP	Friction mean effective pressure
IMEP	Indicated mean effective pressure
RTW	Real time workshop
TMCD	Tool for measuring consistency of datasets
vEMS	Virtual engine management system

CONTENTS

Abstract	i
Preface	iii
Acknowledgements	iii
Abbreviations	v
Contents	vii
1 Introduction	1
1.1 Background	1
1.2 Project formulation	1
1.3 Limitations	2
2 Theory	3
2.1 Heat transfer theory	3
2.1.1 Conduction	3
2.1.2 Convection	3
2.1.3 Radiation	3
2.2 The cooling system of a truck	4
2.2.1 Radiator	5
2.2.2 Charge Air Cooler (CAC)	7
2.2.3 Condenser radiator	7
2.2.4 Coolant pump	8
2.2.5 Coolant mixture	8
2.2.6 Thermostat	9
2.2.7 Cooling fan	10
2.2.8 Engine oil cooler	10
2.2.9 Expansion tank	10
2.2.10 Pipes and hoses	11
2.2.11 Other components	11
2.2.12 Effect of hot air recirculation	11
2.2.13 Pressure loss created by the internals of the engine bay	11
2.3 Computer modeling of cooling systems	12
2.3.1 Introduction to GT	12
2.3.2 System integration and software coupling	13
3 Method	15
3.1 Simulation verification	15
3.1.1 A tool for measuring the consistency between dataset results from test and simulation (TMCD)	15
3.1.2 Acceptance criteria	15
3.2 Communication between models and system coupling	16
3.3 System architecture	17
3.4 Description of system, main functional blocks, acquisition and adaptation	19
3.4.1 Engine model	19
3.4.2 Cooler package model	20
3.4.3 Verification	21
3.4.4 Fan model and fan control	21
3.4.5 Coolant pump model and pump control	23
3.4.6 Thermostat model	24
3.4.7 Coolant circuit model	24
3.4.8 Model for compensating the effect of hot air recirculation	27

4 Models and results	28
4.1 Model with one-way coupling between engine and cooling subsystems	28
4.1.1 Description	28
4.1.2 Characteristics of the model	28
4.1.3 Results and comparisons with measured test data	29
4.1.4 Runtime	32
4.1.5 Summary	32
4.2 Model with double-directional coupling between engine and cooling subsystems	33
4.2.1 Description	33
4.2.2 Characteristics of the model	33
4.2.3 Results and comparisons with measured test data	33
4.2.4 Runtime	36
4.2.5 Summary	36
4.3 Investigation of fuel consumption	36
4.3.1 Method	36
4.3.2 Results	36
4.4 Summary of results	37
5 Discussion	39
5.1 Acceptance criteria	39
5.2 Simulation time step and runtime	39
5.3 Recommendations for choice of software for transient analysis	40
6 Conclusion	41
7 Future developments	42
A Appendix - System architecture	43
B Appendix - Block diagrams	46
C Appendix - Implementation	47
D Appendix - Results from Another tool for transient analysis	48
E Appendix - Torque consistency	51
F Appendix - FlowPDropTableRef object	52
G Appendix - Computer system characteristics	53
References	54

1 Introduction

1.1 Background

Since the advent of the digital age computer models and simulations have been used in engineering to recreate phenomena in systems in a comprehensive, accurate and cost efficient manner in order to obtain performance parameters of interest. Due to the fact that any model is always built around certain assumptions and simplifications it is rarely perfectly accurate, however models could still be useful, provided that the right set of assumptions is used. Usually there is a trade-off between model accuracy and execution time rate. As both these parameters are obviously preferred high, a simulation engineer must make a compromise. This often involves deciding which physical phenomena in the system to capture, to what extent and level of refinement (discretization), what method to use for this, and in the context of transient simulations, time discretization or time step.

A wide range of physical phenomena occur in the cooling system of a heavy duty vehicle. As far as results from cooling simulations are concerned, the main parameters of interest are temperatures, pressures, mass and energy flows in the system. Therefore, thermal-fluid mechanical phenomena are the primary objects of cooling system models.

There is a number of methods for modeling cooling systems, which differ mainly in their technical implementation, but they are all based on fundamental laws in thermal-fluid mechanics such as the laws of conservation of energy and momentum and others.

Both 1D and 3D CFD simulations are used to model completely or partly cooling systems, depending on their specific application. Widely used 1D commercial codes in today's automotive industry are GT-SUITE, KULI, AMESim, FlowMaster. GT-SUITE is a powerful tool for engine performance simulations, which is currently used by Volvo Powertrain.

1.2 Project formulation

The project is a consecutive step in the evaluation of GT-SUITE as a tool for analysis of cooling systems for heavy duty vehicles. The main focus of this work is the capability of the tool in terms of transient analysis and its integrability with the currently used softwares for simulation of power train and power train installation systems. This is done in order to allow for studying the complex interaction between different vehicle subsystems in a transient cycle with prospect of optimizing the overall energy consumption of the vehicle.

When working with transient simulations some of the assumptions, which were possible within the context of steady state analysis are no longer valid and for the sake of accuracy and proper operation of the model, it is necessary to provide it with more functionality, subsystems and data inputs. Naturally, this requires collaboration with a number of units within the organization.

An important aspect in the work is to consider the current structure, typological and complexional diversity of the different simulation models, provided by the respective departments in the organization. Therefore, it is an important practical aspect to keep the modularity of the simulation environment with vision for easy adjustability, updatability and flexibility.

In this thesis a 1D simulation model of Volvo FH truck cooling system will be built in GT-SUITE and will be coupled with an engine model and system control modules to provide the capability for transient analysis with drive cycle data inputs.

The object of the simulation is the FH-844, 13L 460hp Eu6 eSCR. Images of a similar FH truck are shown on Figure 1.1. This particular vehicle is chosen because of the availability of test data, which would allow a comparison between simulation and test results.

The process will begin with collecting all necessary control and functional blocks. Engine model, fan control, coolant pump control will be acquired and adapted to the specifics of the simulation. A model of the coolant circuit will be made, including a functional model of the thermostatic valve. All the blocks will be integrated in a common software environment and the model of the coolant system will be calibrated according to test results.

A two-way interaction between the models will be implemented by feeding the CAC outlet temperature and resistive torque from fan and pump back to the engine model. An investigation of fuel consumption will be demonstrated using this model.



Figure 1.1: *Volvo FH truck (Volvo Trucks Image Gallery)*

The results from the simulations and their consistency with test data will be compared to results for the same drive cycle run by the currently used software tool for transient analysis. More information about the specific criteria for the comparison follow in section Method.

1.3 Limitations

Since the work does not focus on manipulating control strategies, no attempt will be made to improve existing fan or coolant pump control strategies. They may be adapted to fit the needs of the simulation environment, but will not be improved in terms of performance.

Same applies for the engine model, whose function will be verified against test data, but no attempt will be made to modify performance-related parameters within the engine model.

A certain system architecture will be chosen, which will satisfy the requirements stated in the project formulation. Due to the limited time frame other architectures will not be analyzed.

2 Theory

This chapter introduces the reader to the theoretical foundation related to this project. It begins with a short account of the basic mechanisms of heat transfer and continues with a concise description of a truck's cooling system, where the basic components and their functions are summarized. The topics of computer aided modeling of cooling systems is also covered here.

2.1 Heat transfer theory

The section shortly presents the mechanisms of heat transfer.

2.1.1 Conduction

Conduction is a heat transfer mechanism, where energy is transmitted by microscopic diffusion and collision of particles within a body as a result from a temperature gradient. The microscopic particles, which transfer the thermal (also known as internal) energy could be atoms, molecules, electrons and others. Conduction occurs in solids, liquids, gases and plasmas. In the absence of any external drives conduction would equalize the temperature gradient as heat is always transferred from regions of high temperature to regions of low temperature. The magnitude of the transferred heat is proportional to the temperature gradient. The heat flux \dot{q} , is given by Equation 2.1

$$\dot{q} = -k\Delta T \quad (2.1)$$

where k is coefficient of thermal conductivity in $[\frac{W}{mK}]$ and ΔT is the temperature gradient [K]. A good example of a cooling system component, where pure conduction takes place, is the wall of any heat exchanger, across which thermal energy is transferred from one media to another without them being in physical contact. The heat transfer through the material of the wall is driven by conduction.

2.1.2 Convection

Convection is a mechanism of heat transfer, where physical movement of a medium facilitates the transfer of thermal energy. It is the most common heat transfer mode in fluids and it is said to comprise of heat diffusion (conduction) and heat transfer by bulk fluid flow, known as advection. Convection can be forced - when the movement of the medium is caused by an external drive, f.ex a pump, or natural - when fluid buoyancy is the only driving force for the motion of the surrounding fluid.

In a cooling system, convection occurs on both sides of any heat exchanger and in the coolant circuit itself. With prospect of increasing the efficiency of the energy transfer, convection in cooling systems is most often forced by a pump or a fan. Equation 2.2 provides a mathematical expression for the convective heat flux.

$$\dot{q} = h(T - T_w) \quad (2.2)$$

where h is convective heat transfer coefficient in $[\frac{W}{m^2K}]$, T_w is the surface temperature and T is the temperature of the medium [K].

2.1.3 Radiation

All matter, that has temperature greater than the absolute zero emits energy in the form of electromagnetic waves. In this prospect radiation can be seen as a conversion of thermal energy into electromagnetic energy as a result of the oscillation of the micro-particles in matter and the consequent generation of coupled electric and magnetic fields, which emit photons and radiate energy away from the body through its surface. Radiation is a heat transfer mechanism, which does not require the presence of a medium and radiative heat propagates with the speed of light infinitely far unless obstructed.

The intensity of the transmitted radiative heat emitted by a surface of a black body is proportional to the fourth power of its absolute surface temperature. This relationship is defined by the Stefan-Boltzmann law expressed in 2.3

$$E_{rad} = \sigma T_s^4 \quad (2.3)$$

where σ is the Stefan-Boltzmann constant, $5.6704 \times 10^{-8} \left[\frac{W}{m^2 K^4} \right]$ and T_s is the absolute temperature of the surface [K].

Generally a body emits only a portion of the energy stated in the ideal case of black body from Equation 2.3. For this reason a non-dimensional coefficient of thermal emissivity, ε in the range $0 < \varepsilon < 1$ is introduced as follows:

$$E_{rad} = \varepsilon \sigma T_s^4 \quad (2.4)$$

The net radiative heat flux is give in Equation 2.5

$$\dot{q}_{rad} = \varepsilon \sigma (T_s^4 - T_\infty^4) \quad (2.5)$$

where T_∞ is the temperature of the surrounding environment.

2.2 The cooling system of a truck

The main purpose of a cooling system as a part of an internal combustion engine is to reject the heat from the engine block and other engine peripherals to the environment and thus maintain optimal temperature of operation. Heywood [12] points out, that in modern diesel engines the cooling system rejects within the range of 16 to 35 % of the fuel heating value depending on the engine's construction and mode of operation.

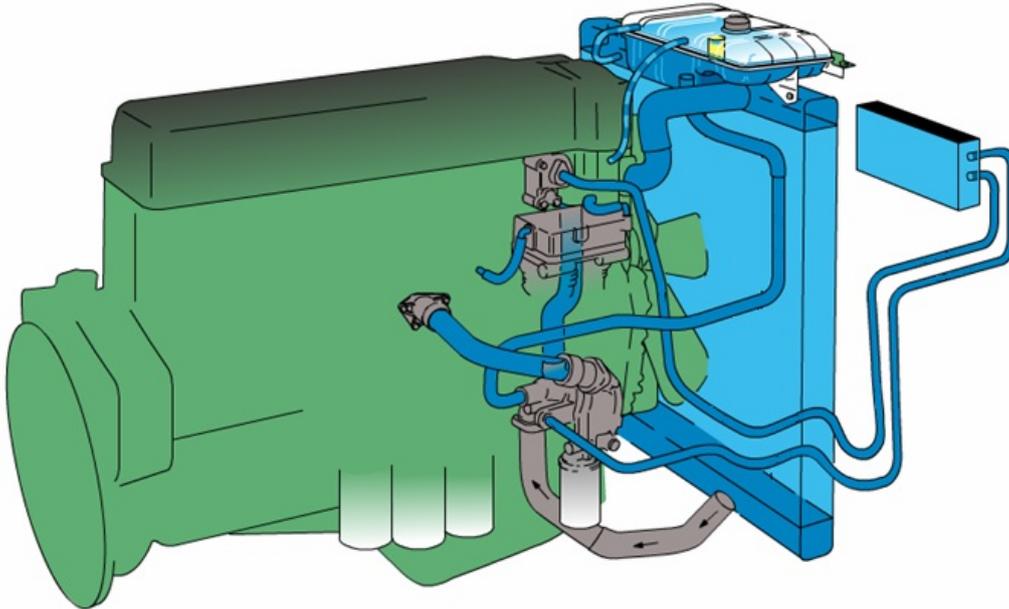


Figure 2.1: *Cooling system of a truck*

A schematic of a very basic truck cooling system is shown on Figure 2.2. It presents a cooling package consisting of a condenser, charge air cooler (CAC), radiator and a fan.

For simplicity of analysis the coolant side of the cooling system can be considered a closed thermodynamic system, where mass remains constant. This is an assumption, which is valid for most operating conditions. In reality coolant can exit the cooling system through the pressure cap once the internal system pressure exceeds a certain limit. In order to perform its function the cooling system needs to have a thermal interface with the surroundings, which is most effectively provided by the heat exchangers. However, heat transfer occurs not only through them, but also through the walls of the coolant transportation system (pipes and hoses) as well as through other components including the engine block itself by a combination of convection and radiation.

Practically all the parts of the cooling circuit take part in some form of heat transfer and in other phenomena, which have a direct influence on it (f.ex. pressure drops influence fluid flow rate, which determines the convective

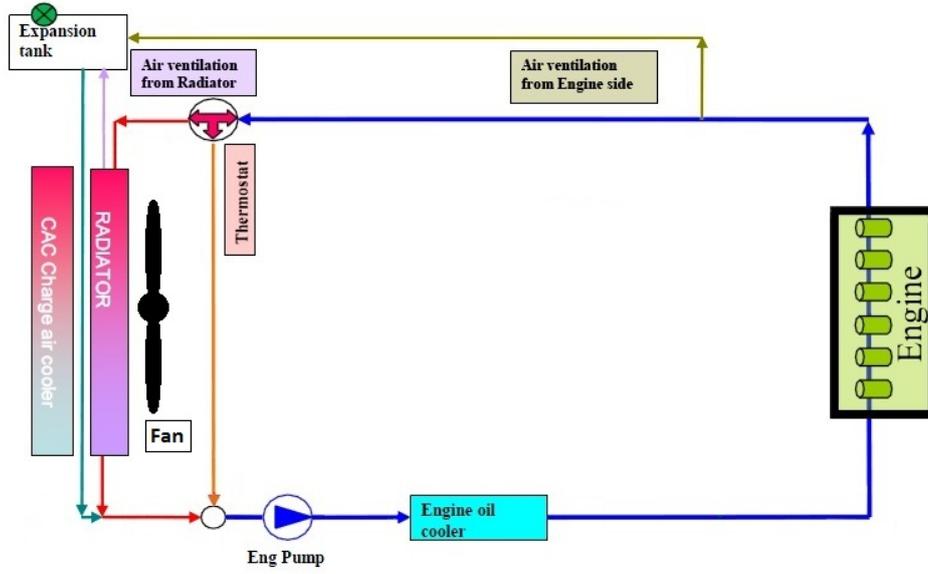


Figure 2.2: Schematic of a truck's cooling system (basic components)

heat transfer coefficient). Therefore, there are many physical phenomena one could model in order to capture the system behavior most thoroughly.

Under normal operating conditions the coolant temperature would be maintained below 105 C°. The most direct mechanism to control the coolant temperature is by adjusting the speed of the fan, which is of suction impeller type. As in most automotive cooling systems the thermostatic valve ensures quick arrival at operating temperatures and it is one of the thermal controls on the liquid side together with the coolant pump control. The following subsections give a short account for each basic component in the cooling system.

2.2.1 Radiator

The Radiator is an air-to-liquid heat exchanger of cross-flow type usually located after the CAC in the stack. It is the main effective interface for heat transfer from the coolant to the surrounding environment. There are many types of radiators available on the market for a variety of different applications. The most common technology for modern automotive radiators is the finned aluminum type, where flat tubes held together by header plates at both their ends are stacked parallel with fins in between them. The joining method is usually brazing. An example of an automotive radiator is shown on Figure 2.3

Radiator cores are available in different thicknesses and face area sizes. The dimensions and the number of the tubes, where coolant flows, affect the pressure drop across the liquid side of the cooler. The same applies for the air side, where fin geometry has great influence on the pressure drop and the overall cooling efficiency of the heat exchanger.

Heat is normally transferred from the hot fluid to the wall by convection, through the wall by conduction and to the cold fluid by convection. The thermal resistance network of this process involves two convection and one conduction resistances. The conduction heat resistance can be expressed as shown in Equation 2.6

$$R_{wall} = \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi kL} \quad (2.6)$$

where k is the coefficient of thermal conductivity, L is material thickness and D_o and D_i are the external and internal equivalent tube diameters.

The total thermal resistance becomes

$$R = R_{total} = R_i + R_{wall} + R_o = \frac{1}{h_i A_i} + \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi kL} + \frac{1}{h_o A_o} \quad (2.7)$$

where A_i is the area of the inner surface of the cooler, A_o is the outer surface and h is the respective convective heat transfer coefficient.



Figure 2.3: *Automotive radiator with inbuilt oil cooler in the outlet header (Courtesy Nissens A/S [1])*

Usually in the analysis of heat exchangers it is more convenient to unite all thermal resistances in a single resistance R and to express the rate of heat transfer between the two fluids as:

$$\dot{Q} = \frac{\Delta T}{R} = UA\Delta T = U_i A_i \Delta T = U_o A_o \Delta T \quad (2.8)$$

where U is the **overall heat transfer coefficient** [$\frac{W}{m^2K}$], which has the same units as the convective heat transfer coefficient. Canceling ΔT and neglecting R_{wall} for small wall thicknesses and high conductive coefficients Equation 2.8 reduces to:

$$R = \frac{1}{h_i A_i} + \frac{1}{h_o A_o} \quad (2.9)$$

It is important to mind the fact that the conductive heat transfer coefficient on the gas side of the radiator, h_o is normally lower than h_i and it is therefore the major contributor to the total thermal resistance of the radiator.

Selecting type and size of a radiator is an important design task in automotive engineering. Modern methods for heat exchanger sizing involve a great amount of computer simulations in order to reach an optimum compromise satisfying the specific design targets and requirements. However, these simulations would not be possible without knowing the performance characteristics of the radiator, which are normally provided by the supplier or by an independent testing facility and include performance parameters such as *pressure loss* [Pa], *cooling performance* [W] given in tables for different flow rates for both sides of the heat exchanger. The dry mass of the core and the internal fluid mass content are important for transient analysis as they influence the dynamics of the system. An example of plots of raw test results are shown in Figure 2.4

Measurements of static pressure loss are normally performed at constant temperatures of both media in order to avoid the effects of varying fluid densities and viscosities. Cooling performance is usually mapped for different mass flows on both sides of the heat exchanger.

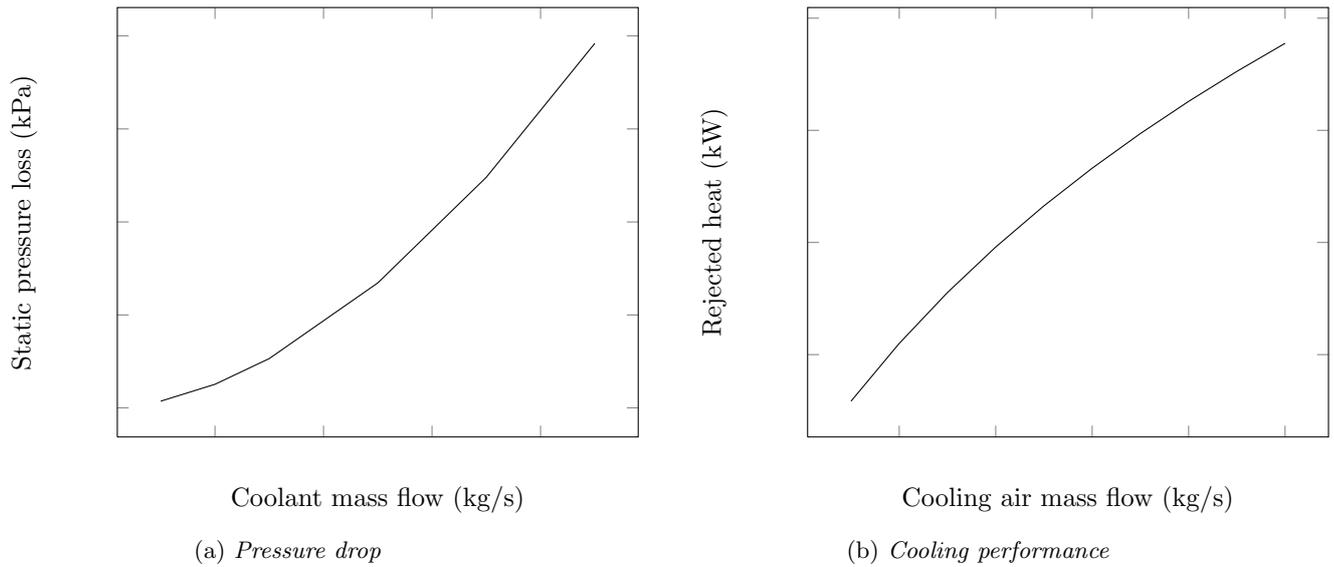


Figure 2.4: *Radiator performance data*

2.2.2 Charge Air Cooler (CAC)

The CAC is an air-to-air heat exchanger of cross flow type typically mounted first or after the condenser radiator in the heat exchanger stack of a truck as shown on Figure 2.5. Its function is to cool down the hot compressed air downstream the turbo or supercharger in order to decrease its specific volume and therefore improve the volumetric efficiency of the engine. In its construction the CAC is very similar to the radiator with the main differences being the size and geometry of the tubes and fins. Some CACs have inbuilt fins in the charge-air side aiming at increasing the internal convective heat transfer coefficient. This, however results in increased levels of pressure loss.

Equations 2.6, 2.7, 2.8 and 2.9 apply for the CAC as well.

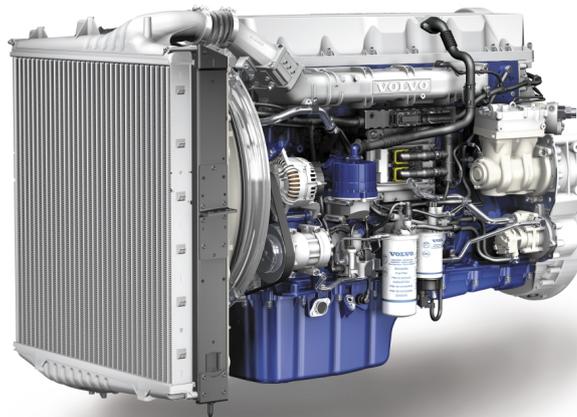


Figure 2.5: *Charge air cooler mounted in a cooler stack on a Volvo E5 13L engine*

2.2.3 Condenser radiator

The condenser radiator, Figure 2.6 is typically mounted before the CAC in the heat exchanger stack and it is a part of the truck's climate control system. The internal medium of the condenser is refrigerant. It enters the condenser downstream of the climate compressor as superheated gas. As it travels through the tubes of the

condenser it rejects heat. During this process the internal energy of the superheated refrigerant decreases until it condenses into liquid phase.

As this work assumes pre-defined constant heat addition from the condenser (typically zero for most test drive cycles), it does not attempt to model and simulate the heat transfer between the refrigerant and the environment. Therefore multiphase heat transfer will not be modeled, but the condenser radiator will be present in the model of the air path with its pressure loss characteristics and constant pre-defined heat addition.

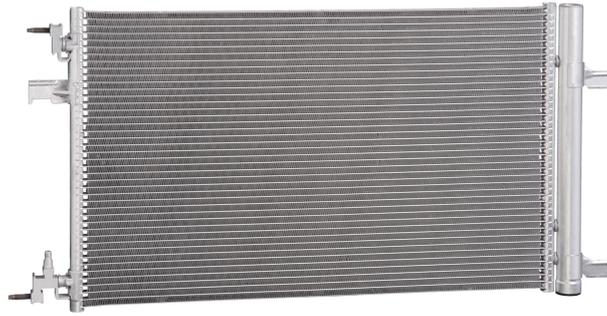


Figure 2.6: *Condenser radiator* [2]

2.2.4 Coolant pump

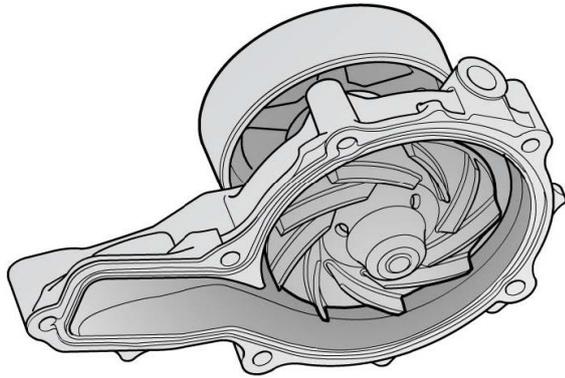
Efficient heat transfer on the liquid side of any modern water cooled automotive cooling system occurs thanks to forced convection. The main drive of this phenomenon is the pressure rise (head) created by the coolant pump, Figure 2.7a, which leads to coolant circulation. Most automotive coolant pumps are of centrifugal type, located on the low-temperature side of the coolant circuit downstream the radiator. The coolant pump is usually permanently coupled to the crankshaft by some type of torque transfer arrangement: a belt or a gear drive, which imposes a fixed gear ratio. For reasons related to fuel efficiency some modern automotive coolant pumps are coupled to the drive by an electromagnetic or viscous clutch. This flexible control allows to interrupt the direct connection of the pump impeller to the pump input shaft and reduce the speed of the impeller in relation to the engine speed, which consequently reduces the energy consumption of this auxiliary component. A control strategy implemented in the ECU activates the clutch depending on parameters as coolant temperature, engine torque, engine speed, etc.

The coolant pump is modeled with help of input data delivered by test. An example of pump performance map for different speeds and flows is shown on Figure 2.7b.

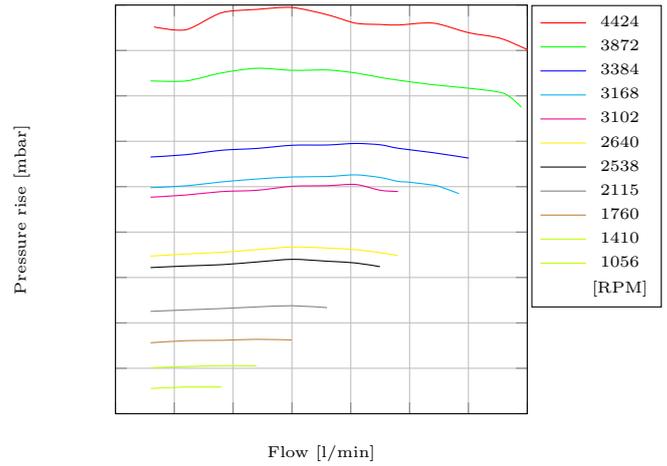
The vehicle analyzed in this work utilizes a two-speed pump with electromagnetic coupling.

2.2.5 Coolant mixture

Coolant mixture is circulated through the coolant circuit into the engine block and its cooling channels, through the channels in the cylinder head and through all other heat exchanges connected to the liquid side of the system. It is the main internal medium for heat transportation. In most automotive applications the coolant mixture consists of water, ethylene glycol and other additives mixed in a certain proportion. The addition of ethylene glycol aims to reduce the freezing temperature and increase the boiling temperature of the mixture, which prevents the coolant from freezing on cold winter days and allows for utilizing higher temperature differences and therefore improved efficiency on hot days. However the specific heat capacity of the mixture also decreases with adding glycol. Important fluid characteristics such as density, specific heat capacity, thermal conductivity, temperature of freezing, temperature of boiling and others vary with glycol concentration in the mixture and are provided in data sheets by the supplier of the coolant.



(a) *Coolant pump [11]*



(b) *Pump performance map*

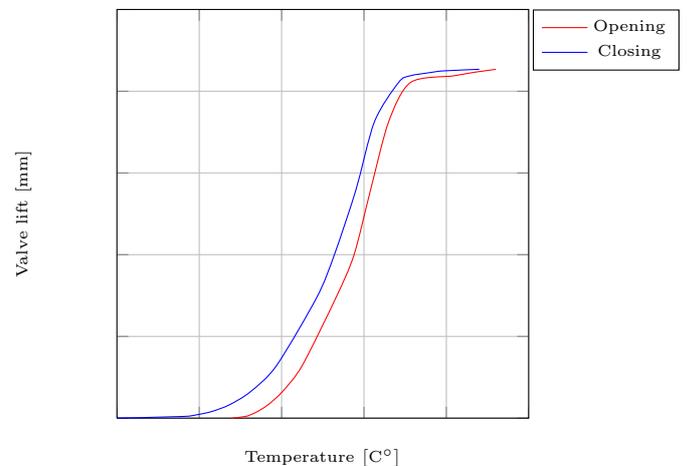
Figure 2.7

2.2.6 Thermostat

The thermostat, Figure 2.8a is a controllable valve, which can be adjusted to bypass coolant flow away from the radiator through a parallel line in order to faster reach and maintain the recommended temperature range for engine operation. Most automotive thermostatic valves are controlled by an in-built thermo-sensitive wax-filled mechanical actuator. Correct modeling of the behavior of the valve and the resultant pressure drop on the radiator side and on the bypass side are critical for the authenticity and accuracy of the transient model. Therefore, time response for closing and opening, hysteresis and pressure drop levels across the valve housing are necessary for the transient model. Time response for closing and opening are measured in a step-response test, where the valve is instantly submerged into a large bath of liquid of constant temperature and the time for the valve to react to it is measured according to a certain testing procedure. Hysteresis is an indication of the difference in the valve's reaction to increasing and decreasing temperatures and is measured in a similar test setup. A plot representing hysteresis of a thermostatic valve is shown on Figure 2.8b.



(a) *Thermostatic valve [3]*



(b) *Thermostat hysteresis*

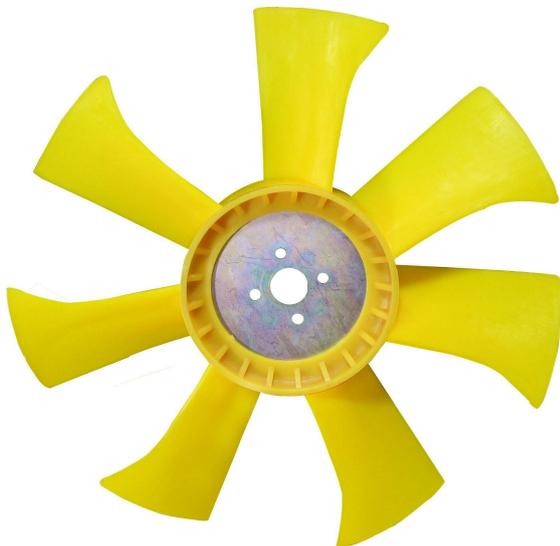
Figure 2.8

2.2.7 Cooling fan

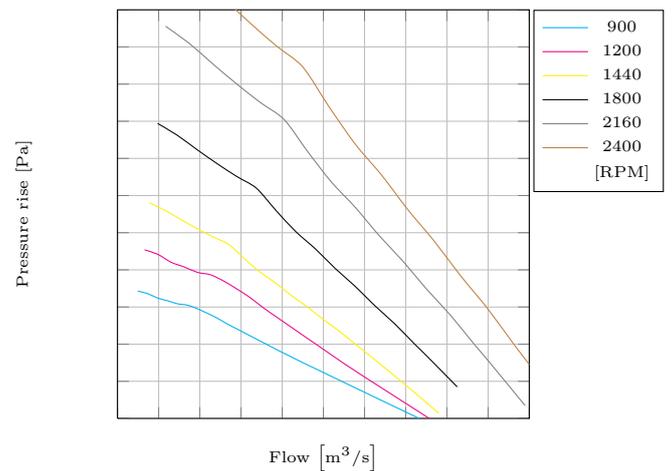
As discussed earlier in Subsection 2.2.1, the Radiator's external coefficient of heat transfer is very sensitive to the amount of air flowing through its core. As the vehicle moves forward, static pressure builds up in front of the radiator generating a natural flow through the core referred to as ram air effect. This is, however not sufficient for all operating ranges of the engine and it is therefore necessary to provide a mechanism, which would drive extra air mass through. This is done by the cooling fan, Figure 2.9a, which is most commonly of axial impeller type with a shroud in order to ensure relatively even mass flow through the entire surface of the core.

There is a number of ways to actuate the fan: by a mechanical connection to the engine, by an electric or hydraulic motor. A prevailing trend in heavy truck industry is to couple the fan mechanically to the crankshaft. Similar to the coolant pump the fan can be connected to the crankshaft through a viscous clutch for optimization of energy consumption and more flexible temperature control. The viscous clutch is controlled by the ECU based on a dedicated control strategy with similar inputs to the ones for the pump control, Subsection 2.2.4. Fan performance maps provide data for the operation of the fan at different speeds and mass flows. An example is shown on Figure 2.9b.

In the context of transient analysis it is vital for accuracy and reliability to implement an adequate fan control strategy in the model.



(a) Cooling fan [4]



(b) Fan performance map

Figure 2.9

2.2.8 Engine oil cooler

Cooling is a major function of engine oil. As it travels through the oil channels in the engine block and as it cools down the piston-cylinder assembly it receives heat, which must be rejected. This is done through a dedicated heat exchanger referred to as engine oil cooler, Figure 2.10a. Different automotive applications have different configurations of engine oil heat exchangers. Some use liquid-to-air coolers, others use liquid-to-liquid setups, where oil heat is dissipated into the engine coolant and thereafter rejected through the radiator. The vehicle analyzed in this work uses an oil-to-water heat exchanger mounted downstream the coolant pump, Figure 2.2.

2.2.9 Expansion tank

The expansion or accumulator tank is a depot for coolant mixture located at the highest point of the coolant system. Its functions are to fill the system with coolant mixture, to continuously bleed the system from air, which decreases the cooling performance and to serve as a volume buffer when the density of the coolant



(a) *Engine oil cooler [5]*



(b) *Expansion tank [6]*

Figure 2.10

mixture fluctuates as a result of varying system temperatures. A picture of an expansion tank is shown on Figure 2.10b

2.2.10 Pipes and hoses

Also called coolant logistic system, pipes and hoses are the connecting structures between major blocks of the cooling system. As their function includes transportation of hot coolant mixture in the rather hostile environment of the engine bay, they are designed to withstand high temperatures, pressures and mechanical vibrations.

In the context of transient simulation it is important to provide the model with an adequate representation of the coolant logistic system, as it creates pressure losses, which influence coolant flow. Furthermore, the quantity of coolant contained in the pipes and hoses acts as a thermal mass, which influences the transient behavior of the system. Heat rejection through the walls of the hoses may be considered too.

2.2.11 Other components

Depending on the specific installation of the vehicle there may be other components in the cooling system such as EGR cooler, transmission cooler, air compressor cooler, cab heater, urea heater, etc.

2.2.12 Effect of hot air recirculation

Hot air recirculation is a phenomenon, which occurs in the engine bay and its surroundings. It is driven by the increased pressure in the regions after the fan. Under different operating conditions some quantities of hot air from this high pressure region may be pulled towards regions of lower pressure levels in front of the radiator, CAC and in front of the entire cooling package. A principle representation of the phenomenon is shown on Figure 2.11

The effect of hot air recirculation can be best modeled by 3D CFD tools. In fast running 1D simulation it is represented by a simplified model usually including preheating of the inlet ram air by a heat addition object or by simply imposing a certain increase in ambient temperature. Magnitudes of ambient air temperature increase for compensating the effect of hot air recirculation are usually acquired from test and calibration for steady state mode of operation.

2.2.13 Pressure loss created by the internals of the engine bay

All components in the engine bay which are in direct contact with the cooling airflow, including the grill and the engine block, exert resistance against the air flow, which in 1D simulations is represented by a single pressure loss component and is calibrated against test data. The pressure loss created by the internals of the engine bay is also referred to as built-in-resistance.

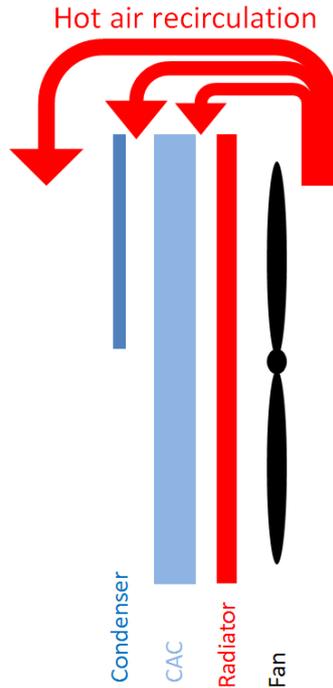


Figure 2.11: *Effect of hot air recirculation*

2.3 Computer modeling of cooling systems

2.3.1 Introduction to GT

GT-SUITE is a set of software tools for simulating different technical areas in a vehicle. These tools are unified under a common interface and this allows system integration as a number of different areas can be simulated simultaneously. The work focuses on GT-SUITE's Cooling system module, however other modules are also used in this relation. GT is currently used by Volvo Powertrain for engine modeling. For a more detailed introduction to GT see [13], [16] and [7].

The GT-ISE environment, Figure 2.12 consists of a template library (to the left), where standard component templates are grouped and ready to be picked and defined. Once this is done they become objects in the project map, which can be seen as a list of all components used in the project. Once these objects are dragged onto the project map, they become parts. The basic idea in GT-ISE is that templates are provided which contain the unfilled attributes needed by the models within the code. The templates are made into objects, and when component and connection objects are placed on the project map, they become parts. These objects and parts may call reference objects. During the course of building the model reference templates will be used; however, these will be automatically imported into the project at the time they are first called. This functionality of the environment makes it very effort-optimal and flexible.

Applications

GT-Suite can be used for a wide range of analysis related to thermal management. Typical applications include: (Cited from *GT user manual* [7])

- Analysis of the entire cooling circuit
- Transient or steady state operation
- Radiator sizing
- Pipe and orifice sizing
- Engine warm-up

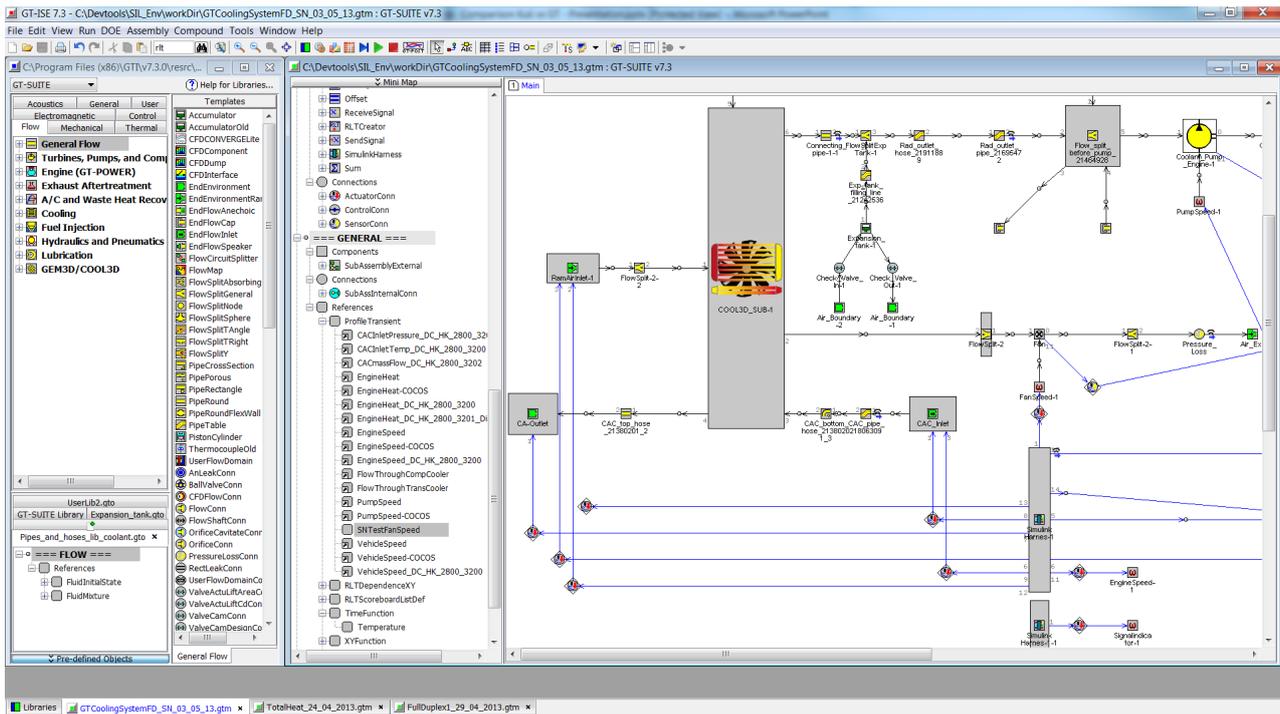


Figure 2.12: GT user interface

- Thermostat specification
- System operation during a prescribed vehicle drive cycle (e.g. Hamburg-Kassel)
 - engine speed
 - engine torque
 - vehicle air speed
 - ram air temperature
 - ambient pressure

2.3.2 System integration and software coupling

Due to the fact that engine thermal management tasks concern interaction between multiple disciplines and subsystems, system integration will be of key importance in this work. Specific attention will be paid to modularity and flexibility of the simulation environment, which would ensure quick and effort-free updatability of the simulation. This naturally requires parallel operation of two or more softwares in the same environment and more importantly, efficient communication between the different software units within the simulation. The two main softwares, which will be used in this work - Matlab Simulink and GT-SUITE, do offer possibilities for such a coupling. GT-SUITE can send simulation outputs (e.g. temperatures, pressures, flows, etc.) to a harness, which transmits this information every time step to Simulink, which processes these inputs and can afterwards output control commands back to GT-SUITE.

There are two modes of the coupling mechanism described as follows:

Running coupled simulations from GT-SUITE

This feature allows users to create their Simulink models using the “GT-SUITE Model RTW” S-function block, and use Real-Time Workshop to generate C-code from the model and compile it as a .dll (or a .so for Linux). This dll can then be pointed to in the ‘SimulinkHarness’ in GT-SUITE. Up to 5 Simulink models can be loaded dynamically by a single GT-SUITE model.

Using this method has several advantages over running the model from Simulink. As with non-coupled simulations, this allows the flexibility of using multiple cases. The GT-ISE DOE tool can now be utilized to analyze systems,

which is not possible when running from Simulink. Additionally, Simulink models can now be simulated without having a Matlab or Simulink license – a GT-SUITE user needs only the .dll/.so file to run their coupled simulation through GT-SUITE.

Cited from GT-Manual, Controls Coupling[7]

Running coupled simulation from Simulink

In this option Simulink operates as a "master", where a GT .gtm or .dat file is pointed to. It allows execution of only one case at a time unless there is a dedicated piece of code in Simulink to switch between multiple cases. For more information refer to GT-Manual, Controls Coupling[7].

3 Method

This chapter introduces the methods and approaches utilized in this work. It begins by familiarizing the reader with measurable criteria for simulation verification. It continues with a detailed account on the simulation methods employed for each functional block of the simulation.

3.1 Simulation verification

3.1.1 A tool for measuring the consistency between dataset results from test and simulation (TMCD)

Readings for parameters of interest are logged during a transient simulation normally each time step. A specially developed code for the purpose of scientific and measurable comparison between large datasets from test and simulation is produced. Its functions are to:

- acquire the data from test and simulation and calculate residuals, coefficient of determination R^2 and mean residual \bar{R} .
- plot and display readings of interest as fan speed, radiator inlet temperature, radiator outlet temperature, coolant flow through cooler, etc. both from test and simulation.
- plot residuals and display values for R^2 and \bar{R} .

The coefficient of determination R^2 is a numerical representation of the consistency of two datasets. Its meaningful range varies between 0 and 1, where 1 represents perfect fit between the elements of the compared datasets. Its mathematical definition follows in Equation 3.1.

$$R^2 = \frac{\sum (A_i - B_i)^2}{\sum (A_i - \bar{A})^2} \quad (3.1)$$

where A represents a dataset acquired from test and B represents a dataset acquired from simulation.

A residual is a dataset, whose elements are the result of a subtraction performed by the corresponding elements of two datasets. If dataset A represents measurements of a certain parameter of interest from test and dataset B represents results from an ideally accurate simulation for the same parameter, the elements of residual dataset Rs would be equal to zero. The mathematical definition of a residual follows in Equation 3.2.

$$Rs_i = A_i - B_i \quad (3.2)$$

where i is the index of an element in each array.

The mean residual \bar{R} represents the arithmetic average of the elements in the residual dataset Rs. It is in other words an indication of the general time-weighted accuracy of the simulation, but should not be relied on as it does not account for the instantaneous deviations, which can be best reported graphically or by the coefficient of determination.

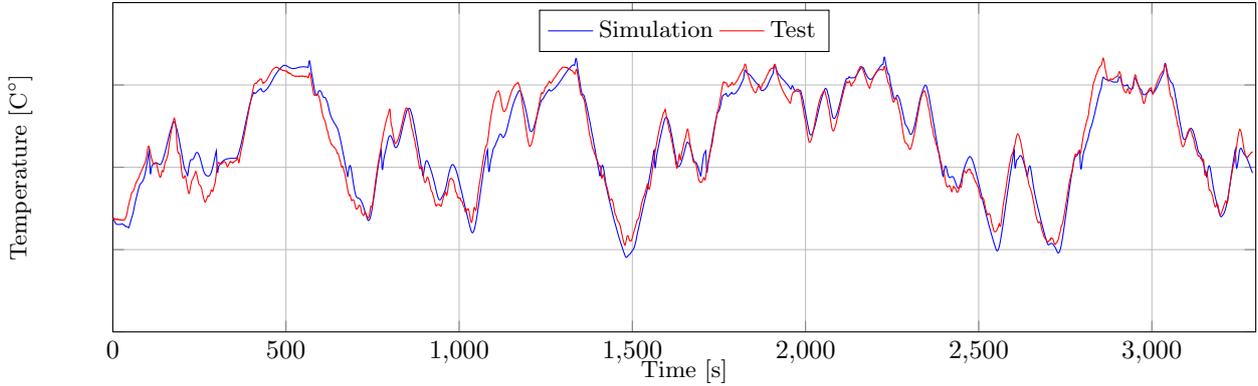
An example of the product delivered by the tool for measuring the consistency between test and simulation is shown on Figure 3.1.

3.1.2 Acceptance criteria

The following areas for correlation between test and simulation have been proposed for validation of results produced by another tool for transient simulations of coolant systems, previously developed and currently used by the company. It has been chosen to adopt these criteria since they would verify if the currently developed tool is at least as accurate as the one being in use at the moment.

- Engine
- Cooling system
- Fan clutch and control

Radiator inlet temperature, $R^2=0.94106$, $\bar{R}=0.0205$



Radiator inlet temperature residuals

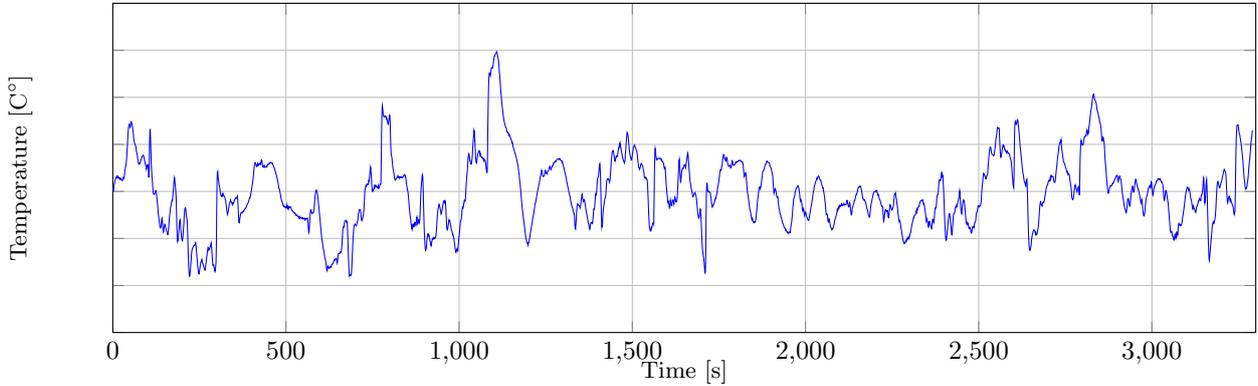


Figure 3.1: *Example of a plot produced by TMCD*

- Complete engine and cooling system installation

For more detailed description of the criteria see Table A.1 in Appendix A.

After an analysis of the results delivered by another tool for transient simulation, Appendix D it has been concluded that the average accuracy achieved by it is $R^2 \approx 0.762^1$. Based on this evaluation of the current status a new quantitative demand for minimum average accuracy to be delivered by the model developed in this work is set to $R^2 = 0.80$.

3.2 Communication between models and system coupling

As the presented task involves parallel running of two or more combined models, the current section will elaborate on the different methods for coupling and communication between the separate models and the consequences from this.

Figure 3.2 presents a block diagram of a system comprising two models with one way communication between them. This type of linking does not allow for mutual interaction between the models, because there is no feedback to the first model. In such a system changes in the first model would affect the results from the second model, but eventual changes in the second model would not affect the results from the first model. If such feedback was enabled there would be two-way communication between the models, Figure 3.3. This would allow for mutual interaction between them and a more realistic representation of the physical phenomena would

¹Average coefficient of determination is calculated as the mean of the first four readings in Table 4.1

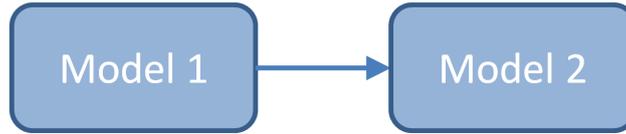


Figure 3.2: *Coupled models with one-way communication*

be achieved. However, it must be pointed out, that this may increase the processing time, because a number of iterations may need to be performed until a balance is reached.

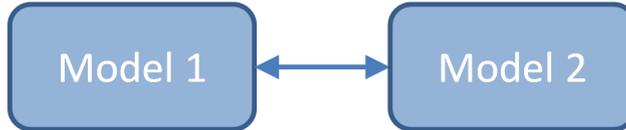


Figure 3.3: *Coupled models with two-way communication*

3.3 System architecture

The term system architecture is used to denote the basic functional arrangement of the software units used to model the system. The following paragraph introduces the reader to the basic structure of the software units used to model the system and the options for their functional integration.

The system consists of two main models: Engine model and a model of the cooling system, which can function independently from each other.

As the engine control is provided in Matlab Simulink and the cooling system is modeled in GT, there are a number of options as system arrangement is concerned. Both software tools provide possibilities for importing foreign models in their environments, therefore any of them could be chosen to function as the main driver (master) or as a subordinate (slave). There are five propositions for system architecture as shown in Table 3.1

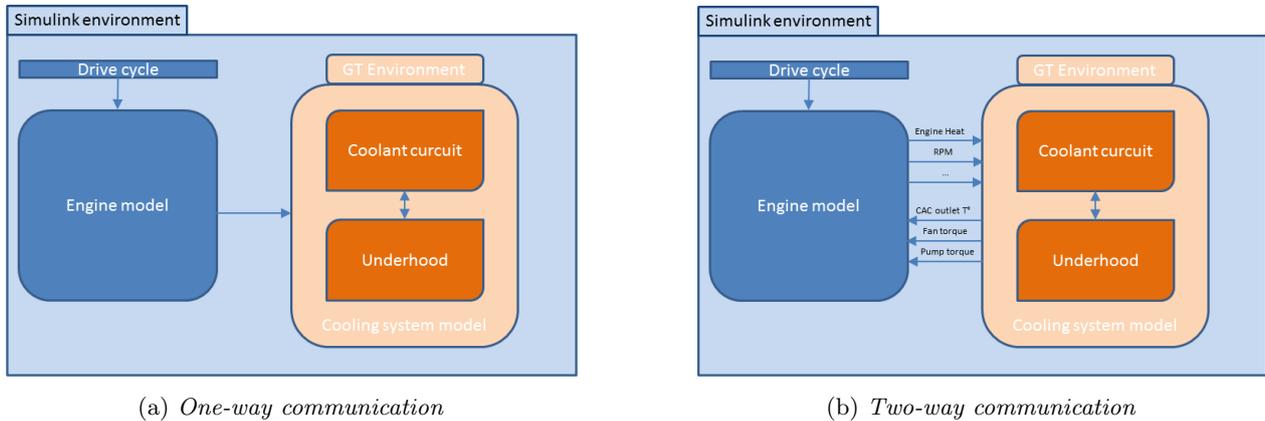


Figure 3.4: *Block diagram: Simulink - master*

Regardless of which choice is made the results and the simulation time would be the same for the first four options as the amount of data transfers is similar. The fifth option is not recommended because of reduced flexibility and susceptibility to failure. The decision should therefore be based on the particular needs of the company and on the advantages and disadvantages stated in Table 3.1.

Considering the characteristics of each option and the main priorities stated in the project formulation, namely the modularity of the simulation environment, adjustability, updatability and flexibility, the first option in Table 3.1 provides the best conditions to satisfy all needs. It is so, mainly because this option maintains the cooling system model and the engine model as separate functional blocks, which would facilitate the updatability and flexibility of the configuration, i.e. it would be easy to interchange complete functional blocks, rather than

Table 3.1: System architecture

Option	Description	Advantages	Disadvantages
Simulink runs as master, GT runs as slave Figure 3.4	The main environment in this case is Matlab. A model of the cooling system is imported into the Simulink environment and run with a time step and control parameters defined by Simulink.	No evident challenges in terms of connectivity. Cooling system model implemented in GT, can use GT-post and make and save plots.	Runs one case per run unless there is a dedicated piece of code, which shifts cases.
GT – master, entire engine model imported Figure A.1	In this option the entire engine model is imported as a .dll file and pointed to in GT by a special object for communication in Matlab.	Both models integrated in GT, can use GT-post and make and save plots, can take advantage of case-engine.	It may be problematic or impossible to export the engine model as a .dll file.
GT – master, mean engine value model run in GT, vEMS imported from Simulink Figure A.2	In this option the vEMS and other peripherals are added to a .dll file and imported to GT through a special object for communication with Matlab.	Both models exist in GT, can use GT-post and make and save plots, can take advantage of case-engine.	Requires extra work from Powertrain’s side, who have to adapt the currently provided model.
Both Software tools run separately. Data exchange between them is done manually. Figure A.3	The engine simulation is first run in Simulink and a file with the input data necessary for the GT part is recorded, then fed to GT.	Easier for cooling simulation engineers, don’t have to deal with Simulink in case Powertrain supplies just the heat trace.	Reduced flexibility as cooling simulation engineers may have to request data files from Powertrain simulation for each particular drive cycle and engine configuration.
Simulink runs as master. Cooling system and engine model implemented in a single file in GT. Figure A.4	Fan control and vEMS run in Simulink. Cooling system implemented in the same .dat file as the engine model. Extra inputs and outputs added to the already existing engine RT object.	Possibility for faster runtimes.	Reduced flexibility and increased probability for failure: It becomes harder to quickly interchange different engines and cooling systems, because each new combination needs to be recompiled into a new .dat file and reimported in Simulink.

having to integrate different models separately every time a new engine model is released. Furthermore, the main controls necessary for the simulation already exist in Simulink and would therefore require minimum efforts to be adapted to the specific needs for the simulation. This feature would improve the efficiency of communication between different departments.

3.4 Description of system, main functional blocks, acquisition and adaptation

The system to be modeled is presented on Figure 2.2. The simulation will provide predictive functional models for the following areas:

- Engine
- Cooler package
- Fan and fan control
- Cooling circuit
- Pump and pump control
- Thermostat

The capability to provide authentic representation of the interaction between these areas is of major importance for the accuracy of the transient simulation. Each of these areas will be explained in the following subsections with focus on their functionality and influence on the dynamic behavior of the simulation.

3.4.1 Engine model

Description

The engine model is implemented in GT and coupled to additional engine control blocks (EMS) in Simulink environment. It is a fast running mean value model, which delivers results with satisfactory accuracy and considerable advantages in terms of runtime of the order $0.5 - 1 \times RT$. It is based on neural networks², which provide data for volumetric efficiency, IMEP, FMEP, Exhaust temperatures, and NOx levels. It is built as a simplification of a detailed engine model, whose runtime is much longer as it provides detailed models for more physical phenomena.

The input to the engine model is a drive cycle, which is a time-indexed table containing the following fields:

- Time [s]
- Engine torque [Nm]
- Engine speed [RPM]
- Vehicle speed [km/h]
- Ram air temperature [C°]
- Atmospheric pressure [Pa]

Functions

The engine model provides most of the input data necessary for the cooling system model. Most importantly, it delivers the instantaneous heat addition [W] from the engine to the cooling system as well as speed of coolant pump input shaft, charge air mass flow and temperature, inlet manifold pressure. In addition it delivers a number of signals necessary for the fan and coolant pump control blocks. The predictive functionality of the engine model makes it able to capture the transient behavior of the system.

Origin

The engine model is acquired from BF66360 System Analysis and Simulation at Volvo Powertrain.

²A neural network is an information processing system inspired by the way the human brain works, which implies a parallel computing architecture. NeuralNet components can be used in GT-SUITE control systems anywhere that an output must be determined as a function of one or more inputs. Its basic purpose is therefore similar to that of a simple lookup table or map. However the method by which a neural network calculates its output is quite different. *Cited from GT-SUITE manual [7]*

Adaptation

A number of adaptations were implemented to the engine model in order to fit the needs of the project:

- Friction heat

During the execution of the first simulations it has been discovered that the rate of heat rejection from engine to coolant is lower than the one measured in test by an offset of 10 to 20%. Further investigations showed, that the initially provided engine model sends signal for the heat transferred from the combustion process to the engine block, but does not account for the heat generated by friction. A simple equation has been implemented in the engine model in GT environment to calculate heat from friction. This was done by sensing the Indicated torque T_i [Nm], the Indicated-minus-Friction torque T_{i-f} [Nm] and engine speed ω [RPM] from the "EngineCrankTrain" object in GT and plug them into Equation 3.3.

$$P_f = (T_i - T_{i-f}) \frac{2\pi\omega}{60} \quad (3.3)$$

where P_f is the friction heat.

The friction heat is added to the combustion heat in order to acquire the total heat addition from the engine block to the cooling system:

$$P_t = P_c + P_f \quad (3.4)$$

where P_t is the total heat and P_c is the heat transferred to the engine block from combustion.

The addition of friction heat eliminated the previously experienced offset and significantly improved simulation consistency with test.

- Extra inputs and outputs

The implementation of two-way communication between the engine model and the cooling system model, Figure 3.4b, required additional input signals to the engine model: CAC outlet temperature and torque from coolant pump and fan. These were added using the original method for communicating signals between Simulink and GT-SUITE. For more information see GT object SimulinkHarness in GT-SUITE user manual [7].

3.4.2 Cooler package model

Description

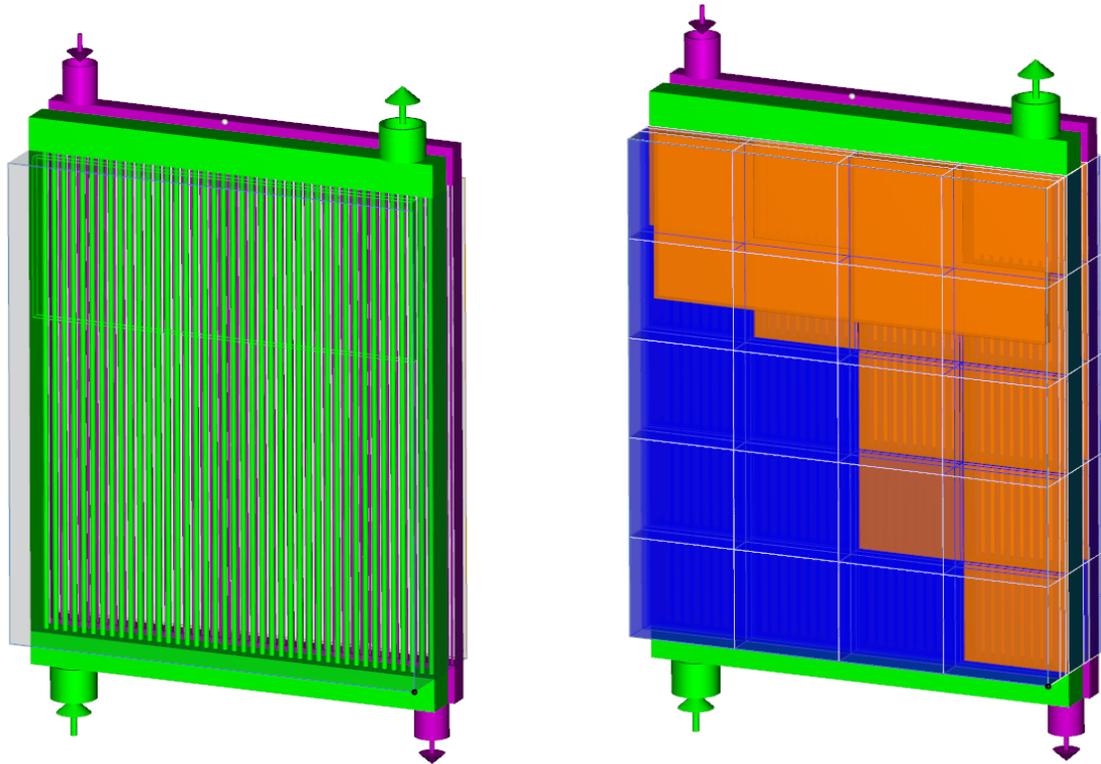
The cooler package model is created in GT-SUITE related tool COOL3D³, which allows external discretization of heat exchangers. The model contains all three heat exchangers as parts of the cooler package: the condenser, CAC and radiator. They are represented by GT-COOL3D specific objects for modeling heat exchangers and contain data for pressure loss and cooling performance as shown on Figure 2.4 as well as data for physical dimensions and position.

The model of the cooler stack gives a predictive quasi-dimensional representation of the heat transfer and fluid dynamics in the cooler stack. Figure 3.5 shows the model in COOL3D.

Functions

The model of the cooler package is used to compute heat transfer between the surroundings and the internal media in the different heat exchangers. It provides a method for computing heat transfer and flow based on a regression of the performance input data.

³COOL3D is intended to aid the user in the model building of the underhood of a vehicle by creating 3-dimensional components. The model can then be discretized in COOL3D to automatically create a GT-SUITE model file that will solve the solution for underhood flow. COOL3D is designed to provide the user a 3D building capability to solve the air flow and thermal distribution in underhood flow caused by heat exchanger stacking to understand the effects on heat exchanger performance and, consequently, cooling system design. In this project COOL3D is seen only as a quasi-dimensional environment, which provides means for external discretization of heat exchangers and is applied for modeling the heat exchanger stack only.



(a) Heat exchanger stack modeled in COOL3D

(b) Discretized model of heat exchanger stack

Figure 3.5

Origin

The model is built by the author in relation to a previous project performed for the company and it is seen fit to be used for the purposes of this work as it concerns the same vehicle. For more information about the model and COOL3D see [13], [16] and [7].

Adaptation

No adaptation is required for the model of the cooler package.

Influence of discretization on simulation runtime

The amount of control volumes modeled in the simulation has a direct influence on simulation runtime. The discretization size is optimized for best runtime.

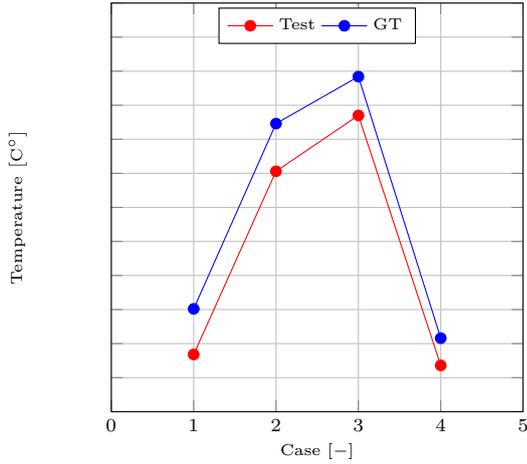
3.4.3 Verification

The cooler stack is calibrated and verified as a part of a previous project. Measurement test points at steady state mode of operation performed as a part of the same test and test setup are used [9]. Built-in resistance and temperatures for compensating the effect of warm air recirculation are tuned in order to reach best fit of radiator and CAC outlet temperatures at imposed inlet temperatures and mass flows from test, Figure 3.6. The calibration method is described in details in [13].

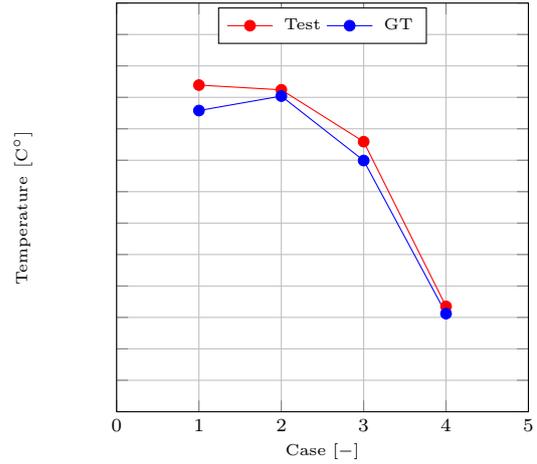
3.4.4 Fan model and fan control

Description

The fan is modeled in GT-SUITE environment by a Fan object, which similar to the heat exchanger object requires fan performance data, Figure 2.9b. A useful feature of the object for modeling fans in GT is that it



(a) Coolant outlet temperatures after calibration



(b) Charge air outlet temperatures after calibration

Figure 3.6

provides an indication of the torque applied to the fan (only in case fan efficiency data is available). This is used in the models, which perform two way communication with the engine model feeding fan and pump torque back to the engine model and applying it on the crankshaft. The fan control is implemented in Simulink.

Functions

The fan control and clutch model have a single clear function: to provide the instantaneous fan speed for the needs of the simulation.

Origin

The fan control is extracted from a currently used software tool for transient analysis.

Adaptation

The fan control blocks are adapted to the needs of the simulation. Time step in the fan control module was kept at 0.1 [s] as the clutch model becomes unstable when running with other time steps. This required the use of Rate Transition blocks in Simulink. Connectivity to GT was implemented by use of the integrated GT object for communication in Simulink.

Simulink block diagram, inputs and outputs

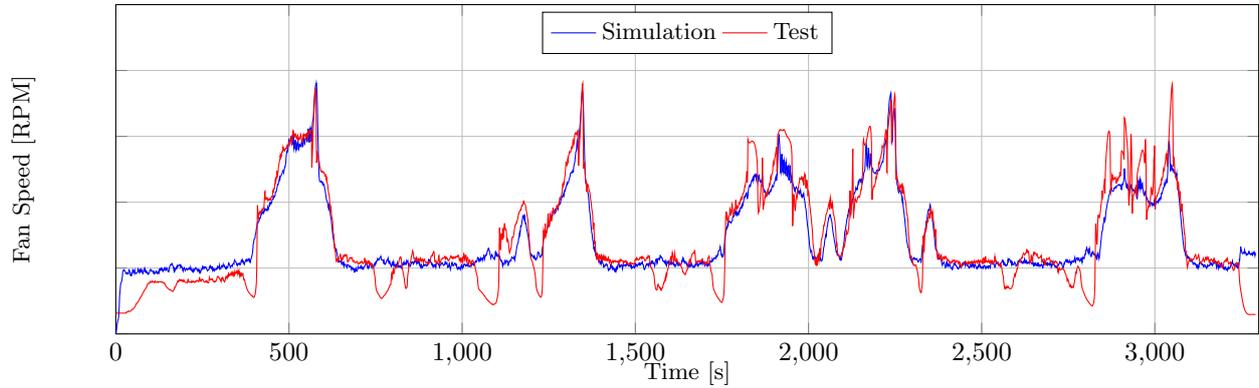
A block diagram with description of all input and output signals of the fan control block is given on Figure B.1 in Appendix B.

Verification

The behavior of the fan control is verified by performing a simulation, where the control block is isolated from the rest of the system. Test data from Hamburg-Kassel drive cycle is used as input to the fan control block. The output is recorded and compared to measured data from the same test using TMCD, Figure 3.7

The coefficient of determination shows satisfactory consistency between simulation and test data, which indicates that the software block responsible for the fan control functions with sufficient accuracy.

Fan Speed, $R^2=0.83295$, $\bar{R}=10.91982$



Fan Speed Residuals

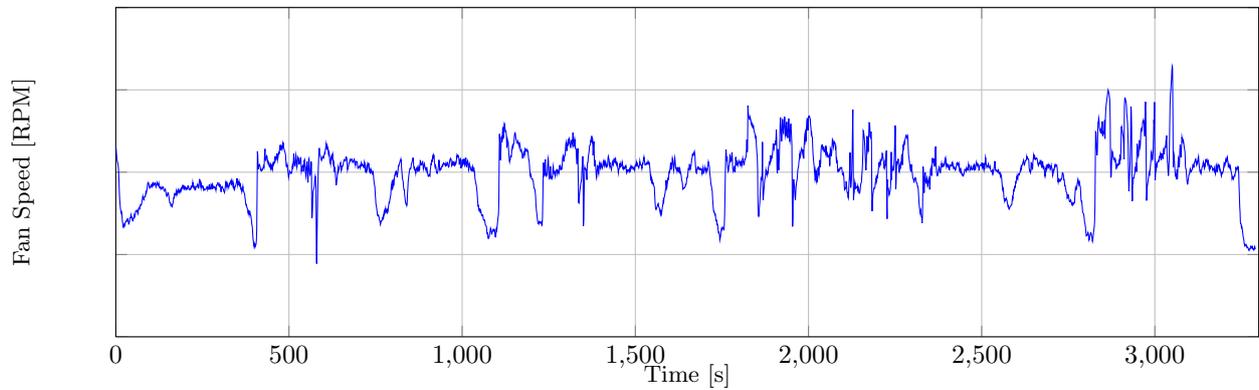


Figure 3.7: Fan control verification

3.4.5 Coolant pump model and pump control

Description

Similar to the fan, the coolant pump is also modeled in GT-SUITE environment by a Pump object, which requires pump performance data, Figure 2.9b. The Pump object also provides the torque consumed by the pump as long as the efficiency input is supplied within the pump input data. The coolant pump used in this vehicle has two speeds of operation switched by an electromagnetic coupling. It is critical for the accuracy of the transient simulation to be able to model pump effects at both speeds of operation. This is implemented by the Pump object, which provides the option to dynamically select a pump map from a list of maps by a control signal given by the pump control block.

Function

The pump control is implemented in Simulink. It consists of a block, that produces a digital signal controlling the mode of pump operation: fully engaged or partly engaged.

Origin

The coolant pump control model was acquired from BF69317 Vehicle Functions.

Adaptation

The pump control block requires no adaptation except for implementing the necessary connectivity. The pump control block is set-up directly in the GT model of the cooling system by use of a GT-SUITE Model (RTW) block in Simulink and exported as a .dat file into GT object SimulinkHarness.

Simulink block diagram, inputs and outputs

A block diagram with description of all input and output signals of the pump control block is given on Figure B.2 in Appendix B.

Verification

The coolant pump model and the coolant pump control are calibrated and verified as parts of the coolant circuit calibration and the main criteria for their verification is the consistency of the resultant coolant flow through the radiator to measurements from test.

3.4.6 Thermostat model

Description

The model of the thermostat is implemented in GT-SUITE by the objects ValveThermostatMConn⁴ and ValveThermostatSConn. These objects compute the pressure drop across the thermostat as a function of temperature, valve lift and time. They require input data for valve lift temperature dependence, Figure 2.8b, pressure drop across the valve as a function of valve lift and time response for closing and opening.

Function

In the context of transient analysis the thermostat model directly affects the coolant mass flow through the radiator. It is therefore crucial for the accuracy of the simulation to have a properly functioning model of the thermostat.

Origin

The input data for valve lift temperature dependence is acquired from documentation available on the corporate database. Pressure loss data as a function of valve lift is acquired from a report of a CFD simulation[10].

Verification

The thermostat model is calibrated and verified as a part of the coolant circuit calibration and the main criteria for its verification is the consistency of the resultant coolant flow through the radiator to measurements from test.

3.4.7 Coolant circuit model

Description

The term coolant circuit is used to describe all objects which are in physical contact with the coolant mixture considered in the aspect of the pressure losses, which they create and their influence on the flow of coolant mixture through them. A snapshot of the GT-ISE project map containing a block diagram of a coolant circuit model is shown on Figure 3.8. A predictive model of the coolant circuit requires functional models of pressure losses and flow dynamics for all components in the coolant circuit:

- All heat exchangers
- All pipes and hoses
- Engine block

⁴ ValveThermostatMConn is used to describe a heat-activated master thermostat valve. *Cited from GT-SUITE manual [7]*

- Thermostat
- Coolant pump

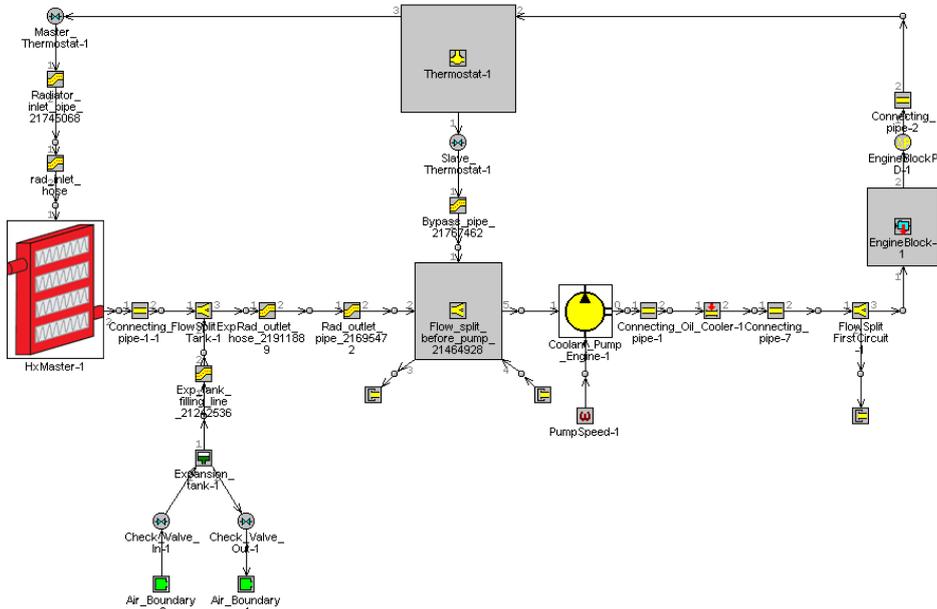


Figure 3.8: *GT model of a coolant circuit*

Pressure loss input data from test is used as input to GT-SUITE objects, which define the pressure loss of flow through the respective components. An example of such an object is the FlowPDRopTableRef. For description see Appendix F.

Function

The model of the coolant circuit has a direct influence on the instantaneous coolant mass flow through the heat exchangers and consequently on the heat transfer rate through them. The main function of the coolant circuit model is to represent the flow and pressure drop phenomena realistically in order to acquire reliable and accurate flow rate quantities. Furthermore, the coolant circuit model should provide a mechanism for heat addition to the fluid from the environment as a result of convective (and radiative) heat transfer from the circuit to the surroundings.

Origin

The pressure loss data as function of flow for the heat exchangers are supplied as a part of their performance data from the respective sources within the corporation. Pressure loss data of engine block is supplied by BF67350.

Method for modeling pipes and hoses

The pipes and hoses are exported from CAD and translated to GT components by a GT specific tool GEM3D⁵ as described by Vdovin[16]. The tool has proven to be useful and time-efficient as it provides an automatic way of translation and discretization of all forms of pipes and hoses from CAD data.

⁵GEM3D is a 3D graphics tool to convert a 3D CAD file into a model suitable to open in GT-ISE. It can be used as a "characterizer", where it can find the effective length, diameters, bends, volume, etc. of a complex 3D shape, and turn them into GT-SUITE-equivalent parts, such as pipes and flowsplits. It also has a model-building capability that is especially useful for building up parameterized mufflers, air-boxes, and plenums, for example with different baffles, perforates, pipe positions, and wools. Cited from *GT manual*[7]

Verification

In order to verify the proper and accurate functionality of the coolant circuit and its integral components a calibration is performed. Since the simplified form of the coolant circuit allows two paths for the coolant: through the radiator and through the bypass line, it is logical to calibrate the circuit by adjusting two pressure loss objects: one on each of these paths. This is done for a set of operating points selected from Hamburg-Kassel drive cycle at points of time, where the flows in the circuit are relatively steady. Calibrating points are selected for both modes of coolant pump clutch operation: directly connected and electromagnetically engaged, Table 3.2.

Table 3.2: Coolant circuit calibration - Values not displayed due to reasons related to secrecy

Parameter	Units	Electromagnetic engagement			Direct engagement		
		1	2	3	1	2	3
Time	[s]	327	1522	3270	542	630	1393
Flow through radiator	[l/s]	N/A	N/A	N/A	N/A	N/A	N/A
Engine speed	[RPM]	N/A	N/A	N/A	N/A	N/A	N/A
Radiator inlet temp.	[C°]	N/A	N/A	N/A	N/A	N/A	N/A
Radiator outlet temp.	[C°]	N/A	N/A	N/A	N/A	N/A	N/A

The method of calibration involves adjustment of the two pressure loss objects until the coolant flow through the radiator matches the values stated in Table 3.2. This is performed strictly at inlet and outlet coolant radiator temperatures consistent to test measurement in order to reproduce the exact same conditions for the coolant pump control, the thermostat valve and the viscosity of the coolant as in the test.

Figure 3.9 shows the results of a drive cycle simulation performed without calibrating the coolant circuit. The maximum flow levels from simulation exceed those from test approximately with 15%. The reasons for this deviation could be attributed to the simplification in terms of excluding the transmission and compressor coolant sub-circuits and to imperfections of the input data. However, the general pattern of the flow signal from simulation resembles the one from test. Apart from the offset flow level, one can notice a delayed switching of the pump mode, which is a result of one or more inadequate inputs to the coolant pump control block. The coefficient of determination is outside the range of its meaningful values, which indicates poor consistency between test data and simulation results.

Coolant flow though radiator, $R^2=-0.81224$, $\bar{R}=-1.49898$

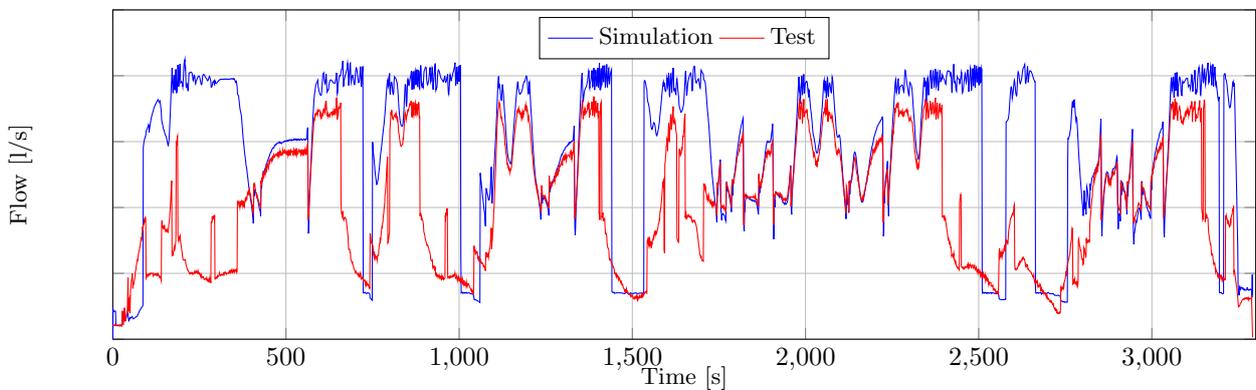


Figure 3.9: Coolant circuit verification: before calibration

Figure 3.10 shows the results of a drive cycle simulation performed after calibrating the coolant circuit. The

calibration has resulted in a clear improvement in the consistency between test and simulated coolant flow through the radiator. There are some residuals primarily at the points when the pump mode is being switched. They are attributed to an imperfection of the simulation concerning the transition between mode of pump operation and electromagnetic engagement. In the simulation this transition is performed instantly, while in the real physical system it happens gradually. Despite the minor deviations the consistency between test and simulation results is satisfactory, reaching coefficients of determination of up to 0.755. The accuracy of the coolant circuit model (including the models of the thermostat and coolant pump) after calibration is considered to be sufficient for the purposes of this project.

Coolant flow though radiator, $R^2=0.75525$, $\bar{R}=0.17378$

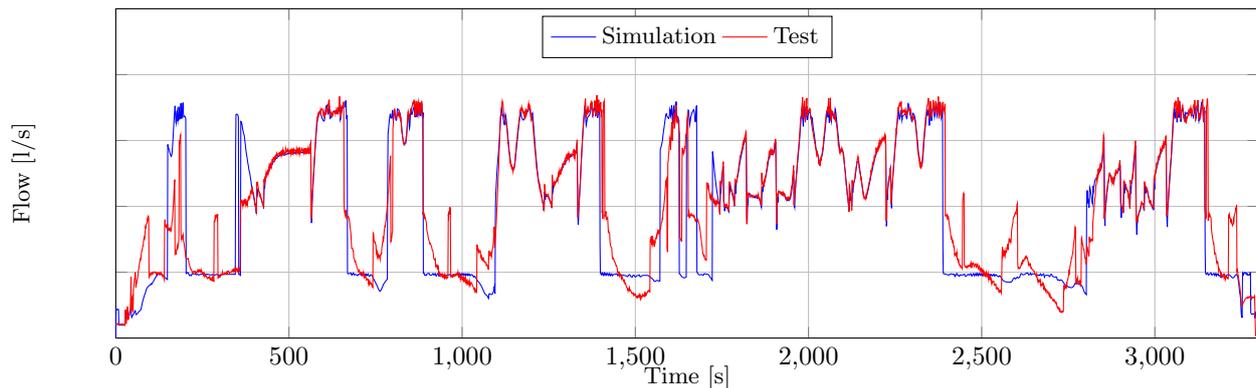


Figure 3.10: *Coolant circuit verification: after calibration*

3.4.8 Model for compensating the effect of hot air recirculation

Description

A Simulink block function used to calculate the ambient preheat temperature increase for compensating the effect of hot air recirculation.

Origin

Values from calibration are used to create a mapping of the preheat temperatures as a function of:

- Fan speed [RPM]
- Rejected heat [W]
- Vehicle speed [km/h]

The resultant map is implemented in Matlab script and used directly in the Simulink code.

4 Models and results

4.1 Model with one-way coupling between engine and cooling subsystems

4.1.1 Description

The first model produced in this work implements single-directional connectivity between the engine and the cooling subsystems as shown on Figure 3.2. As previously explained, in such a model both systems run simultaneously, but the engine subsystem operates independently from the cooling subsystem, i.e. changes in the outputs from the cooling subsystem would not affect the performance of the engine subsystem. The main aim with this model is to prove the basic manageability of the task and to serve as a first step towards implementing the more complex bi-directional connectivity.

The model is made in Simulink environment, belonging to the original engine model, where a functional block containing the cooling system related models and controls is added, Figure 3.4.

Figure 4.1 presents the block diagram of the cooling system model as implemented with the engine model. For detailed description of implementation, inputs and outputs, see Appendix C. Communication with the GT-SUITE model of the cooling system is performed by the GT-SUITE v7.3 (mask)(link) block (top of Figure 4.1). Semi-directional communication between the engine model and the cooling system model is realized by the source Simulink block "From", which forwards signals from the engine output to the GT model of the cooling system.

Most connecting lines in this block diagram carry multiplexed signals. The blocks "Input adaptation" and "Output adaptation" house signal unit translation, routing and multiplexing only. The pump control is integrated directly in the GT-SUITE model of the cooling system using a .dll file compiled and exported to GT with Real-time workshop. For more information see "Controls Coupling" manual, part of GT user manual [7].

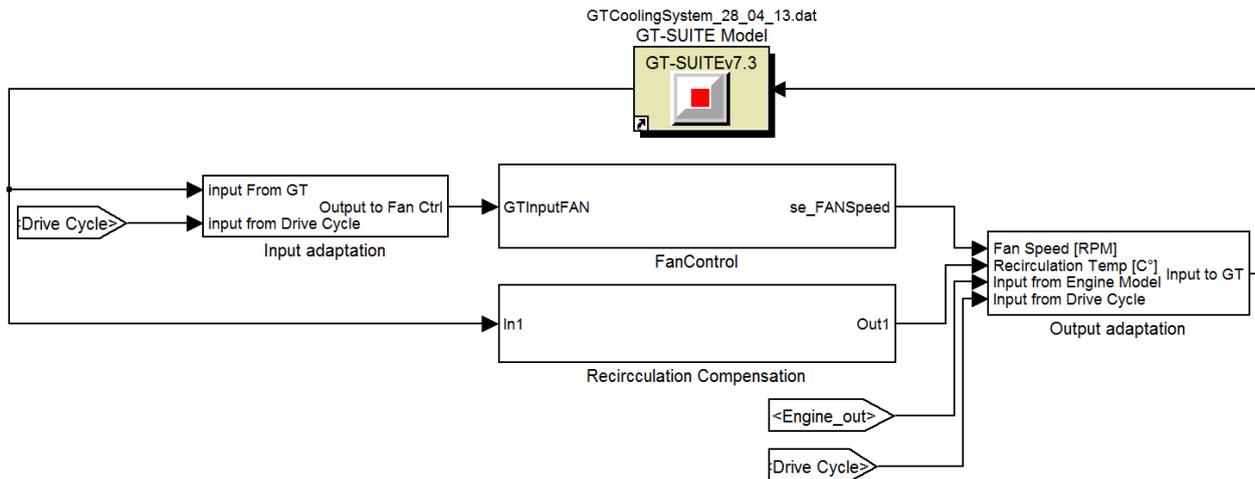


Figure 4.1: Block diagram of model with one way coupling between engine and cooling subsystems

4.1.2 Characteristics of the model

The following list summarizes the predictive models provided in the simulation of the cooling system. For detailed information about the modeled physics and the simulation objects used, see Appendix D.

Coolant circuit

- Detailed one-dimensional model of flow in all components incl. pressure losses and interaction with fluid flow

- Heat transfer from coolant circuit to engine bay environment
- Predictive model of the thermostat with dynamic computation of valve lift, resulting pressure losses and interaction with fluid flow
- Model of the coolant pump with two modes of operation and control strategy
- Representation of the pressure losses and thermal masses in engine block and oil cooler

Cooling package

- Predictive models of CAC and radiator with dynamic computation of heat transfer rate and pressure losses and heat capacitance
- Representation of external pressure loss in all heat exchangers and condenser radiator
- External discretization of heat exchangers performed in a quasi-dimensional environment (COOL3D)
- Model of the cooling fan with control strategy
- Predictive model of the air path by a solution of equations governing the ram air produced by moving vehicle and wind plus the effect of the fan.
- Empirical acquisition of ambient temperature increase for compensating the effects of hot air recirculation

Engine model

- Simplified, fast running, mean value engine model
- Heat transfer from combustion to engine block
- Heat addition from friction
- Engine structure represented by a lumped thermal mass model, which has the heat transfer and pressure loss characteristics of the engine.

Vehicle drive cycle The model is tested with Hamburg-Kassel drive cycle for which data is available from VFL test.

4.1.3 Results and comparisons with measured test data

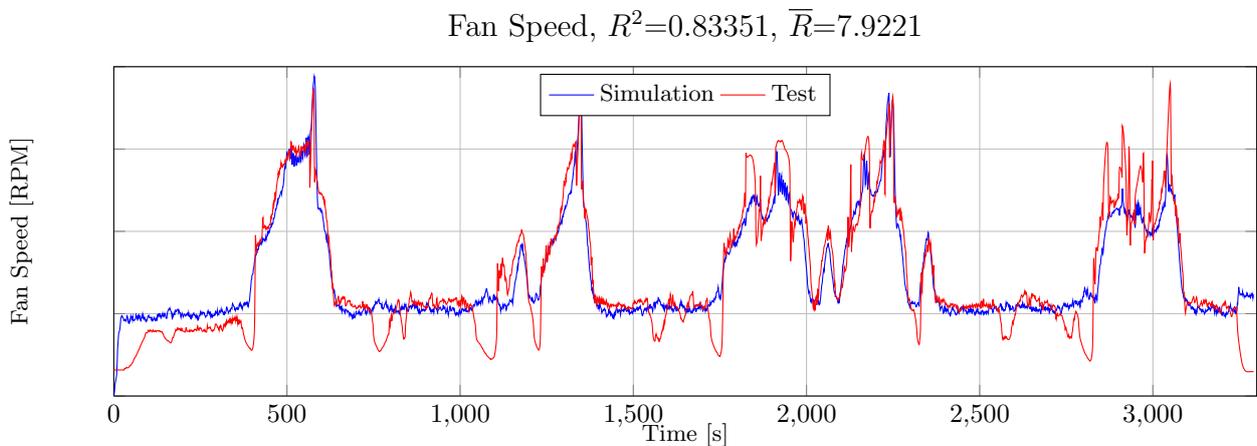


Figure 4.2: Plot: Fan speed

Radiator inlet temperature, $R^2=0.94106$, $\bar{R}=0.0205$

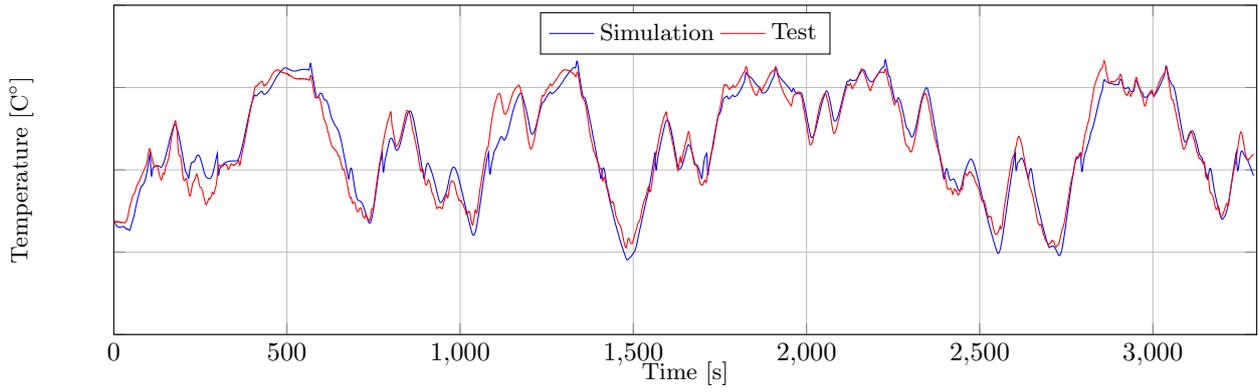


Figure 4.3: Plot: Radiator inlet temperature

Radiator outlet temperature, $R^2=0.90851$, $\bar{R}=0.11661$

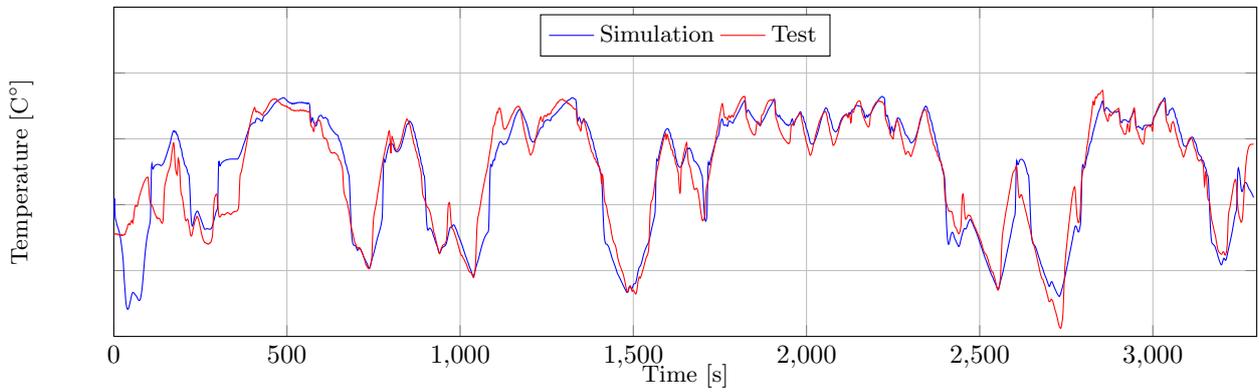


Figure 4.4: Plot: Radiator outlet temperature

Coolant flow through radiator, $R^2=0.70155$, $\bar{R}=-0.12076$

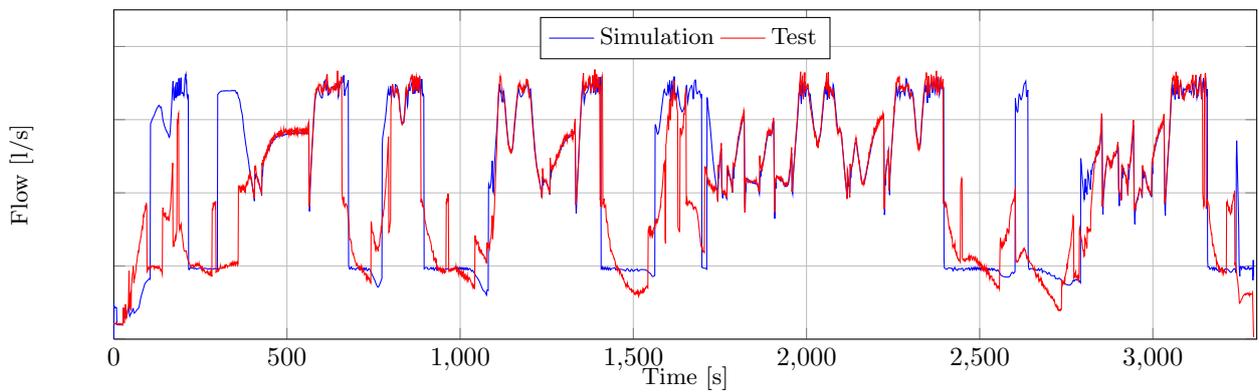


Figure 4.5: Plot: Coolant flow through radiator

The results from the first model, implementing single-directional communication between the engine and cooling subsystems, are generally satisfactory with coefficients of determination reaching 0.94.

Figure 4.2 shows good consistency between measured and simulated fan speeds. The major residuals occur at $t \approx 1100$ s, 1250 s, 1750 s and 2800 s. These deviations result mostly from the imperfections of the fan control strategy, which imposes a low fan speed limit. The impact of the inaccurately simulated fan speed at $t \approx 1100$ s can be observed on Figure 4.3 at the same point of time, where the highest residual of approx. 5 C° occurs. The picture is similar on Figure 4.4, where the residual persists at the same point of time. There is a peaking residual in the beginning of the simulation, which is a result of imperfectly set initial conditions. These inaccuracies are quickly reduced and do not affect the results after $t=500$ s. The simulated coolant flow through radiator, Figure 4.5 closely follows the measured values except for a starting period of approx. 400 s. The behavior of the coolant pump mode-switching is good. However, the coefficient of determination does not exceed the required value of 0.8. This is mostly due to the previously discussed imperfection in the coolant pump model, which does not properly account for the dynamic behavior of the pump pressure rise after switching from high to low speed, which results in a repeating peaking pattern in the behavior of the flow residuals after each switching. This influences the calculations of the heat rejected through the radiator: it introduces noise and consequently lowers the coefficient of determination, Figure 4.6.

Rejected heat through radiator, $R^2=0.40607$, $\bar{R}=-921.86092$

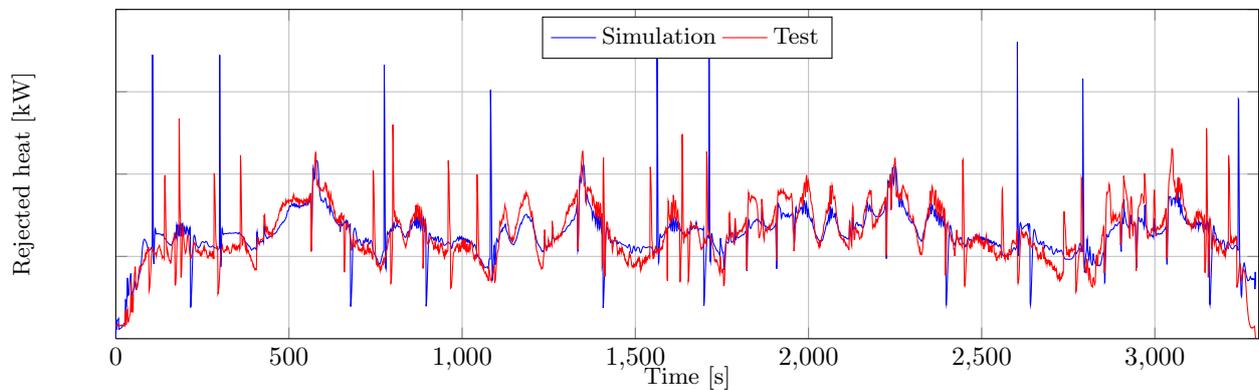


Figure 4.6: Plot: Rejected heat through radiator

Inlet charge air temperature CAC, $R^2=0.81703$, $\bar{R}=-2.21051$

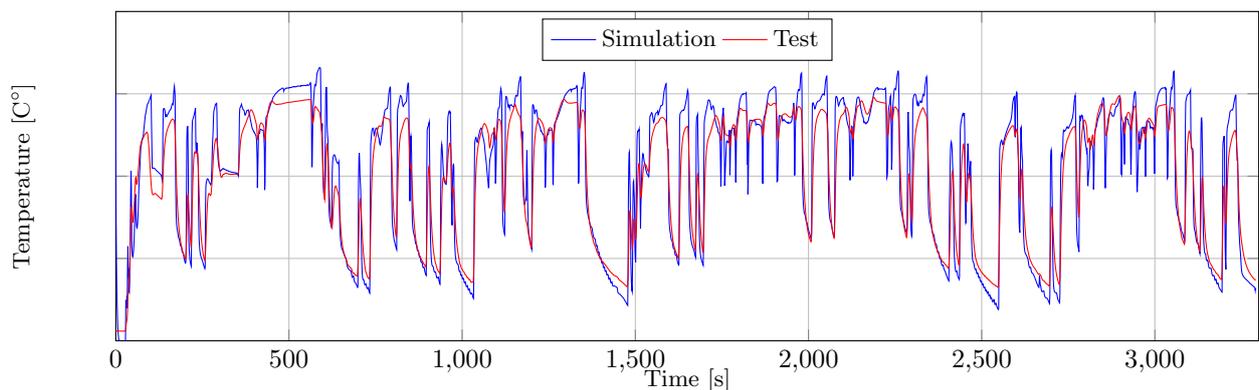


Figure 4.7: Plot: CAC inlet temperature

Figure 4.7 shows satisfactory consistency between the inlet charge air temperatures from test and simulation. R^2 exceeds the required value of 0.8. This is not the case for the results delivered by the currently used software for transient analysis, Appendix D, where the residuals have high magnitudes and levels of noise and R^2 is outside the range of its meaningful values. Similar levels of accuracy apply for the charge air mass flow through

the CAC, which is also supplied by the engine model, Figure 4.9. The advantages of integrating a predictive engine model in the cooling system analysis are clear.

Despite the satisfactory values of R^2 , Figures 4.9 and 4.7 show instantaneous offset of approx 10 % in the high regions. This is taken into consideration, but no attempt has been made to improve the accuracy of the engine model since this is not within the scope of the project and the current accuracy satisfies the acceptance criteria.

Outlet charge air temperature CAC, $R^2=0.72269$, $\bar{R}=-0.27406$

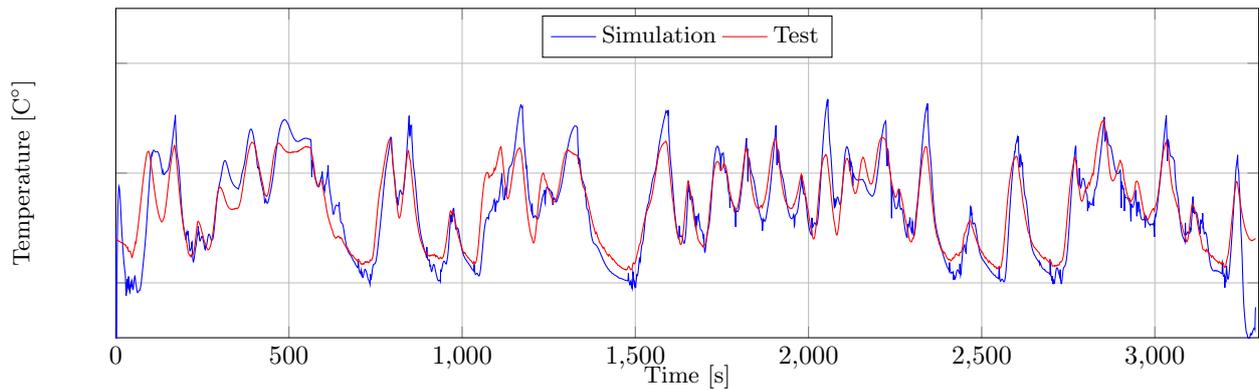


Figure 4.8: Plot: CAC outlet temperature

Charge air mass flow through CAC, $R^2=0.82802$, $\bar{R}=-0.01662$

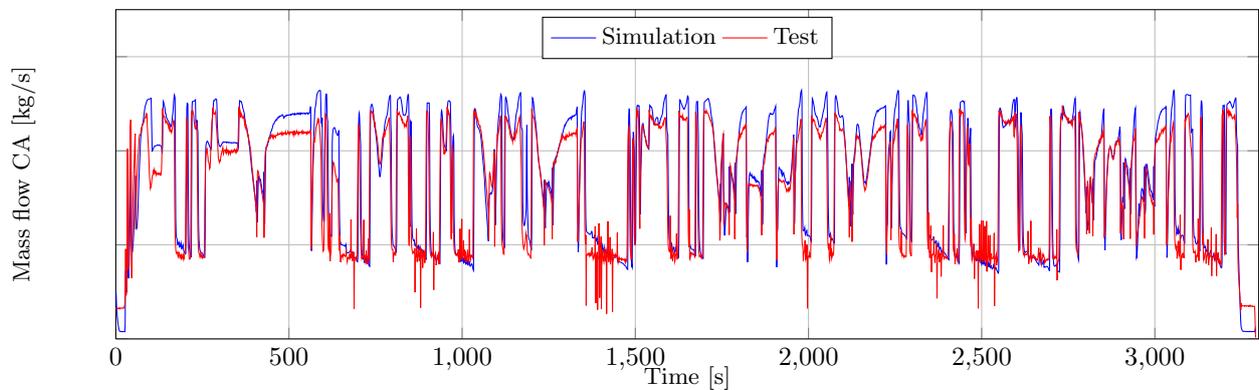


Figure 4.9: Plot: Mass flow CAC

4.1.4 Runtime

The fastest achieved runtime for this model is 4542 s for a part of Hamburg-Kassel drive cycle, which is 3291 s long. In other words the simulation runs 1.38 times slower than Real Time ($1.38 \times RT$).

4.1.5 Summary

The model implementing single-directional communication between the engine and the cooling subsystems outputs results consistent to test. For most of the observed parameters R^2 satisfies the required value of 0.8.

4.2 Model with double-directional coupling between engine and cooling subsystems

4.2.1 Description

The second model produced in this work implements double-directional connectivity between the engine and the cooling subsystems as shown on Figure 3.3. It reflects the interaction between the two subsystems, which is vital for purposes related to reduction of the total energy consumption in the truck. The main change in this model involves feeding CAC charge air outlet temperature and auxiliary torque from the cooling system simulation back to the engine model, Figure 4.10. These two signals are multiplexed and sent to the engine model through the "Go to" Simulink object "Engine_in".

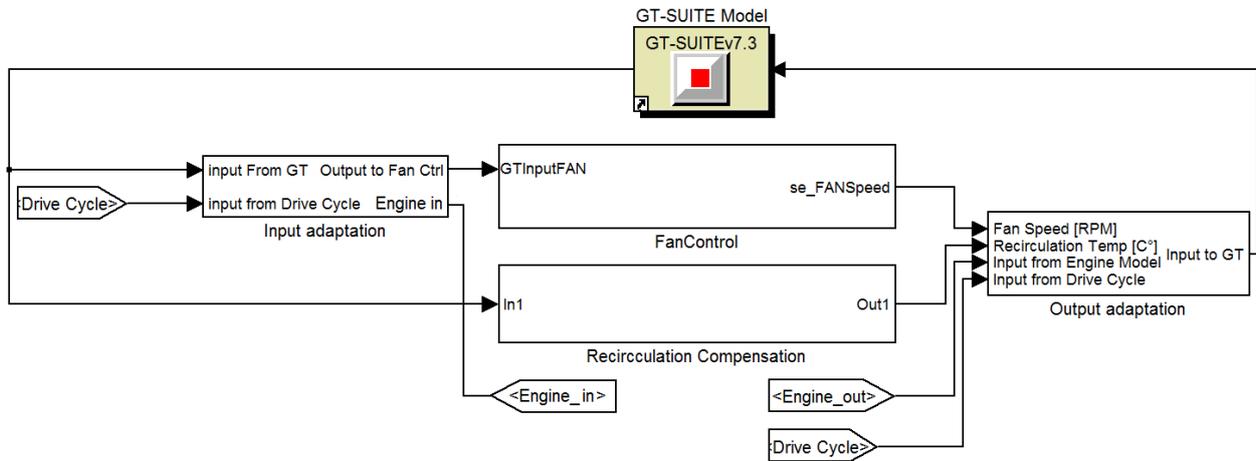


Figure 4.10: Block diagram of model with one way coupling between engine and cooling subsystems

4.2.2 Characteristics of the model

The model with double-directional connectivity has all characteristics of the one with single-connectivity.

4.2.3 Results and comparisons with measured test data

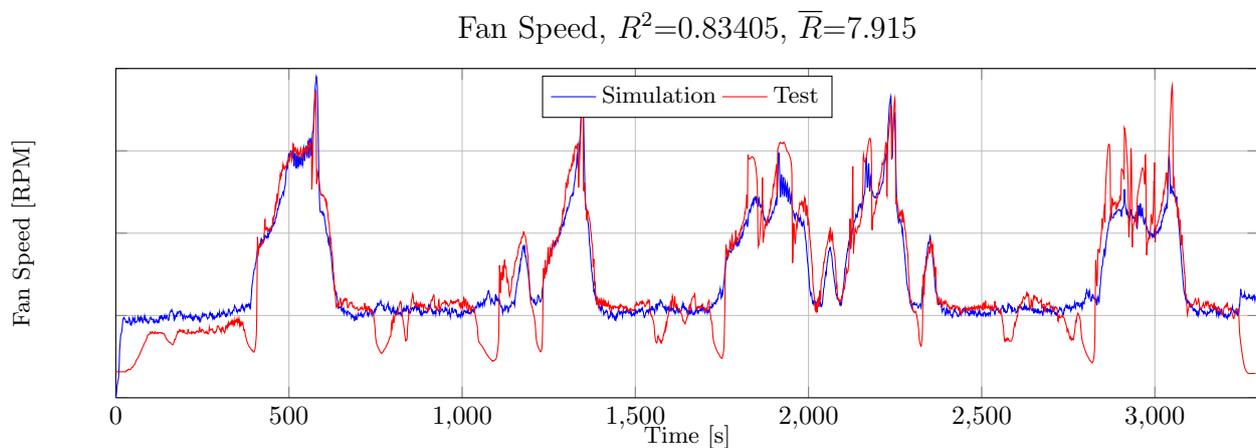


Figure 4.11: Plot: Fan speed, full-duplex

Radiator inlet temperature, $R^2=0.94002$, $\bar{R}=-0.16048$

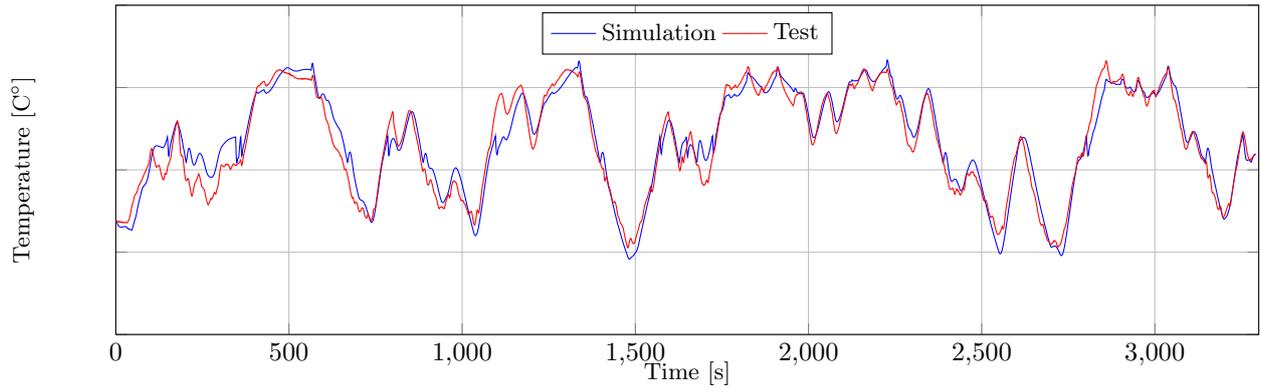


Figure 4.12: Plot: Radiator inlet temperature, full-duplex

Radiator outlet temperature, $R^2=0.8939$, $\bar{R}=0.58415$

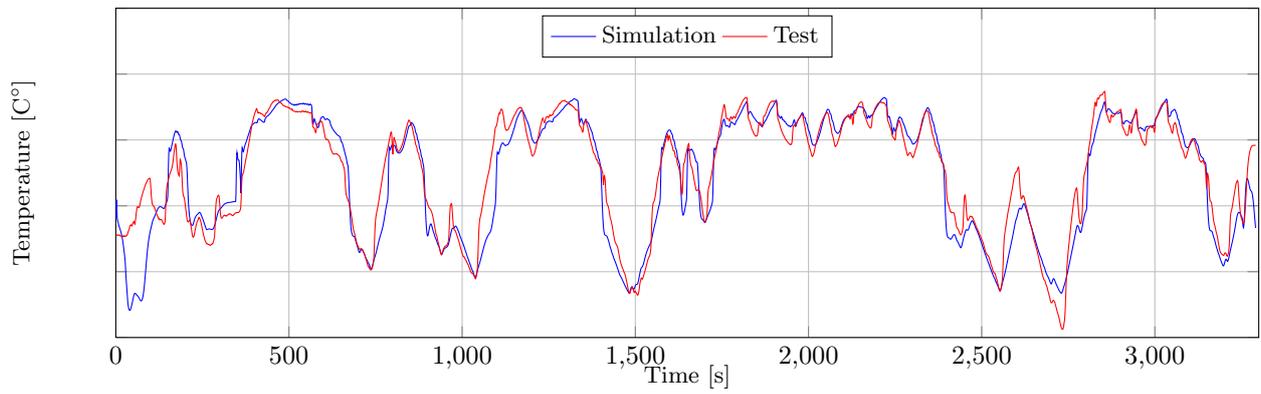


Figure 4.13: Plot: Radiator outlet temperature, full-duplex

Coolant flow through radiator, $R^2=0.75525$, $\bar{R}=0.17378$

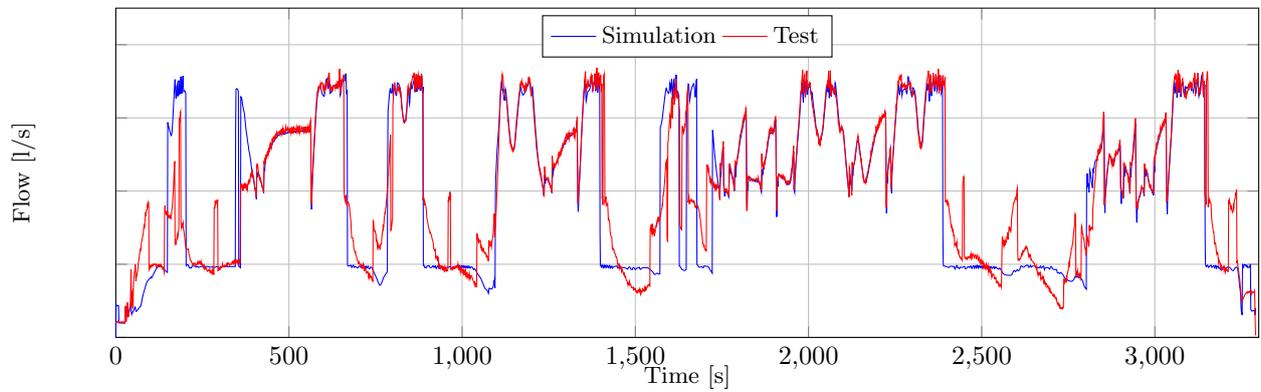


Figure 4.14: Plot: Coolant flow through radiator, full-duplex

Rejected heat through radiator, $R^2=0.43901$, $\bar{R}=1456.4733$

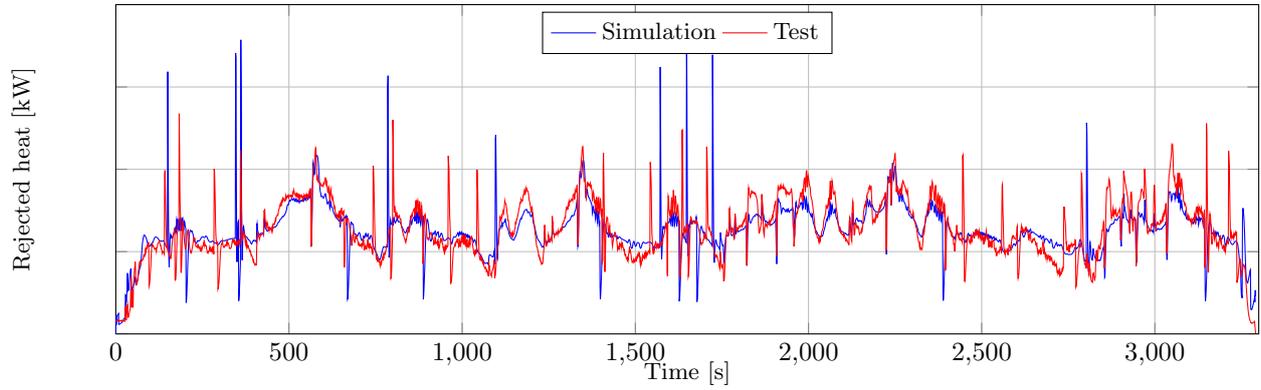


Figure 4.15: Plot: Rejected heat through radiator, full-duplex

Inlet charge air temperature CAC, $R^2=0.8182$, $\bar{R}=-2.09363$

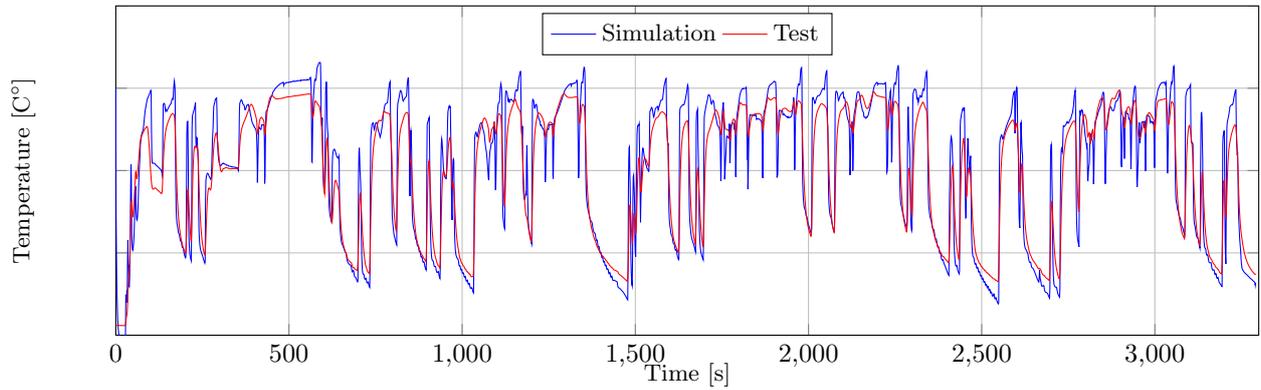


Figure 4.16: Plot: CAC inlet temperature, full-duplex

Outlet charge air temperature CAC, $R^2=0.72551$, $\bar{R}=0.25015$

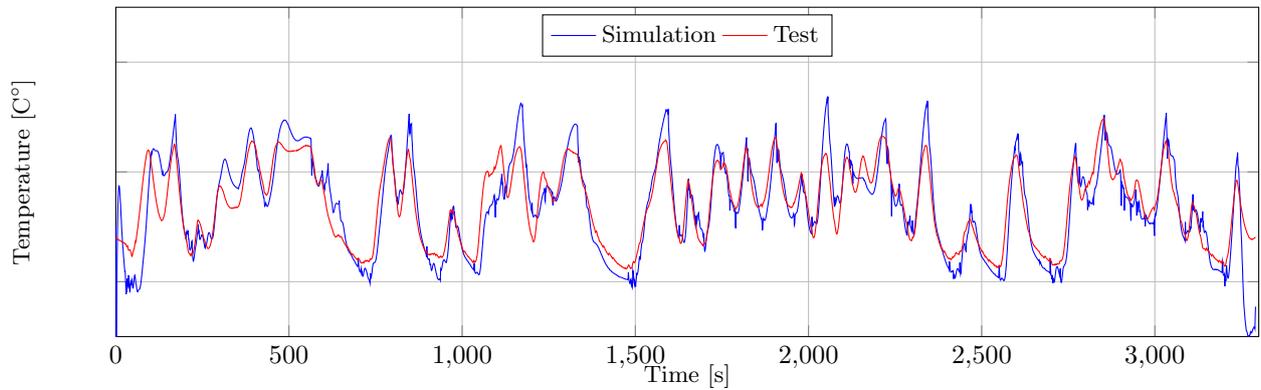


Figure 4.17: Plot: CAC outlet temperature, full-duplex

Charge air mass flow through CAC, $R^2=0.8254$, $\bar{R}=-0.01714$

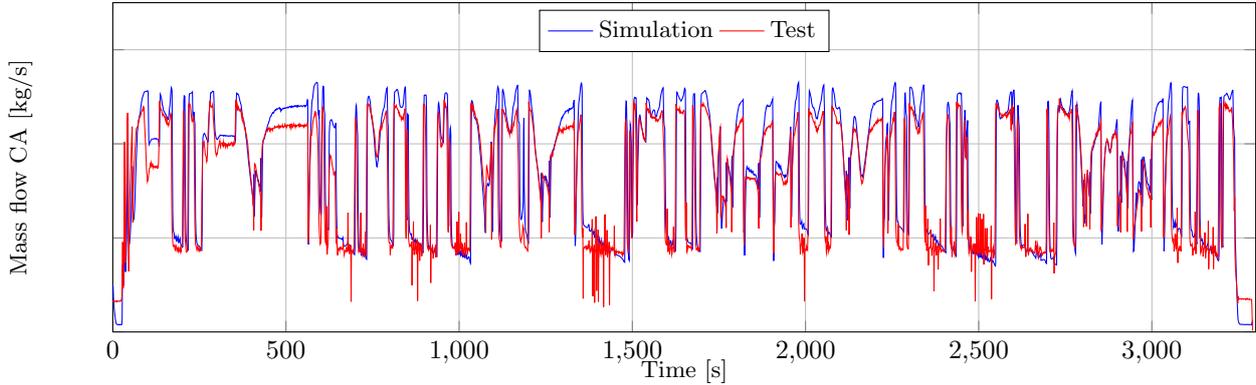


Figure 4.18: *Plot: Mass flow CAC, full-duplex*

4.2.4 Runtime

There is no significant change in simulation runtime in comparison with the previous model.

4.2.5 Summary

The accuracy of the model with two-directional connectivity between the engine and cooling subsystems is similar to the one achieved by the model with single connectivity. The benefits from implementing double connectivity are expressed in the possibility for monitoring the effect of auxiliaries and their control settings on the fuel consumption of the vehicle. Since it represents the interaction between the subsystems more realistically without any penalties in terms of runtime it is strongly recommended to use the model with double connectivity for all purposes.

Simulated temperatures after CAC closely follow the behavior of measured values reaching R^2 of ≈ 0.7 , which is seen to be adequate considering the accuracy of the inlet CAC temperature and mass flow (used as inputs in calculating the outlet CAC temperature) of ≈ 0.8 .

4.3 Investigation of fuel consumption

The models presented above are used to perform an investigation of fuel consumption for the Hamburg-Kassel drive cycle.

4.3.1 Method

Four simulations are performed for Hamburg-Kassel drive cycle. The first applies the model implementing single-directional connectivity between engine and cooling subsystems and therefore not accounting for the energy consumed by the auxiliaries (fan and coolant pump).

The second applies the model implementing double-directional connectivity between engine and cooling subsystems and therefore accounting for the energy consumed by the fan and coolant pump.

The third simulation is made using the latter model, but pump control is overridden to impose constant full-speed mode of operation.

The fourth simulation is performed using the double-directional model with both pump and fan permanently coupled to the crankshaft. Though this scenario is rather unrealistic, it is carried out for the purpose of providing a large fan torque requirement, which should be clearly visible in the results for consumed fuel. Fuel consumption is simulated, logged and compared for each of these scenarios.

4.3.2 Results

Figure 4.19 presents the results from the simulations.

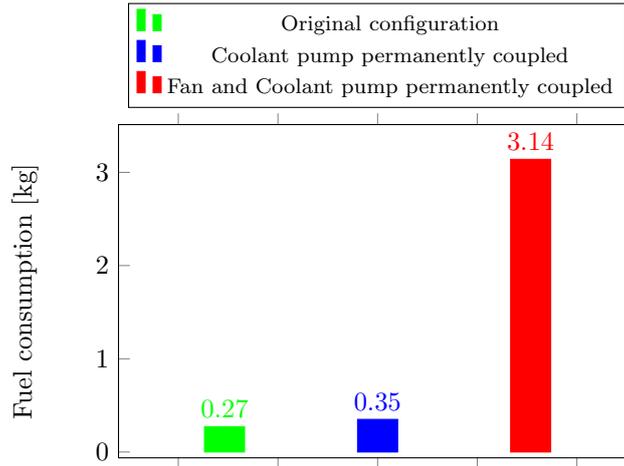


Figure 4.19: *Fuel consumption due to fan and coolant pump*

With coolant pump constantly engaged to the engine crankshaft fuel consumption increases with $\approx 0.2\%$, which is within the expected range of 0.1 to 0.8%.[11] The increase in fuel consumption as a result of having the fan and pump constantly connected to the engine crankshaft in comparison to the original configuration is 7.06%.

4.4 Summary of results

Hamburg-Kassel drive cycle simulation is performed by means of the currently used software for transient simulation. See detailed plots of results in Appendix D. Table 4.1 presents a side-by-side comparison of accuracy and runtime between the performed simulations. Differences between the two GT simulations are insignificant. On the other hand, there are major differences between the performance of GT and the other tool for transient analysis. Due to the imperfections of the models employed by the other tool, the dynamics of the thermal-fluid phenomena, which occur in the coolant circuit are not authentically represented. This has a direct impact on the accuracy of other readings, e.g. the radiator outlet temperature, whose accuracy in the other tool is much lower than the one achieved by GT.

The advantages of employing a detailed engine model are clear from the readings for R^2 on the CAC inlet temperature and CA mass flow. The consistency of these parameters to test measurements is high for the models implementing a predictive engine model, while the ones delivered by the other tool for transient analysis do not even give meaningful values for R^2 . Naturally, the CAC outlet temperature (i.e. boost temperature, which has direct influence on engine performance) is much more accurately accounted for by the GT-model.

Generally, the models done in GT achieve much higher accuracy than the other tool for transient analysis.

Furthermore, it has been demonstrated that the GT-model with dual connectivity can be used for energy-optimization analysis, i.e. investigations of fuel consumption, which may be useful for testing new control strategies for auxiliary devices, etc.

When it comes to runtime, GT-models are 12 to 13 times slower than the models done in the other tool for transient analysis.

Table 4.1: Summary of results

Property of comparison with test	GT model, single connectivity	GT model, double connectivity	Another tool
$\overline{R^2}$	0.846	0.855	0.762
Fan speed, R^2	0.834	0.834	0.650
Coolant temperature, R^2	0.941	0.940	0.920
Radiator outlet temperature, R^2	0.909	0.894	0.676
Coolant flow through radiator, R^2	0.702	0.755	0.804
Rejected heat through radiator, R^2	0.406	0.439	-0.050
CAC inlet temperature (CA), R^2	0.817	0.818	-0.877
CAC outlet temperature (CA), R^2	0.723	0.725	0.536
Mass flow through CAC (CA), R^2	0.828	0.825	0.745
Runtime [\times RT]	1.380	1.430	0.110

5 Discussion

5.1 Acceptance criteria

A question is asked: *Does the model satisfy the acceptance criteria?*

All areas stated in Section 3.1.2 have been examined and satisfied in the simulations, results from which are shown in the report. The independent functionality of the engine model is verified by the model with one-directional connectivity. Figure 4.6. Even though R^2 is lower than the required value, satisfactory consistency between test and simulated results is shown graphically. The reason for the low coefficient of determination is the rapid switching of coolant pump mode of operation as previously explained.

The independent functionality of the cooling system has been verified as a part of calibrating and verifying the underhood model as explained in Section 3.4.2 and [13]. The proper functionality of the cooling system has furthermore been confirmed by the model with one-directional connectivity. Figure 4.3 and 4.4. The achieved coefficients of determination satisfy the acceptance criteria.

The independent functionality of the fan and fan control is verified in Section 3.4.4. Satisfactory values for R^2 are reached.

The accuracy of the complete engine and cooling system installation model is verified by the double-directional model and results are clearly presented in Table 4.1. A detailed analysis of the results from simulation based on measurable quantities accounts for the fact that the model satisfies the acceptance requirements on all levels.

5.2 Simulation time step and runtime

A question is asked: *How does selection of time step influence accuracy and runtime?*

A short investigation was performed in order to answer this question. The Hamburg-Kassel drive cycle was run with different time steps and simulation runtime and accuracy were plotted. See Figure 5.1.

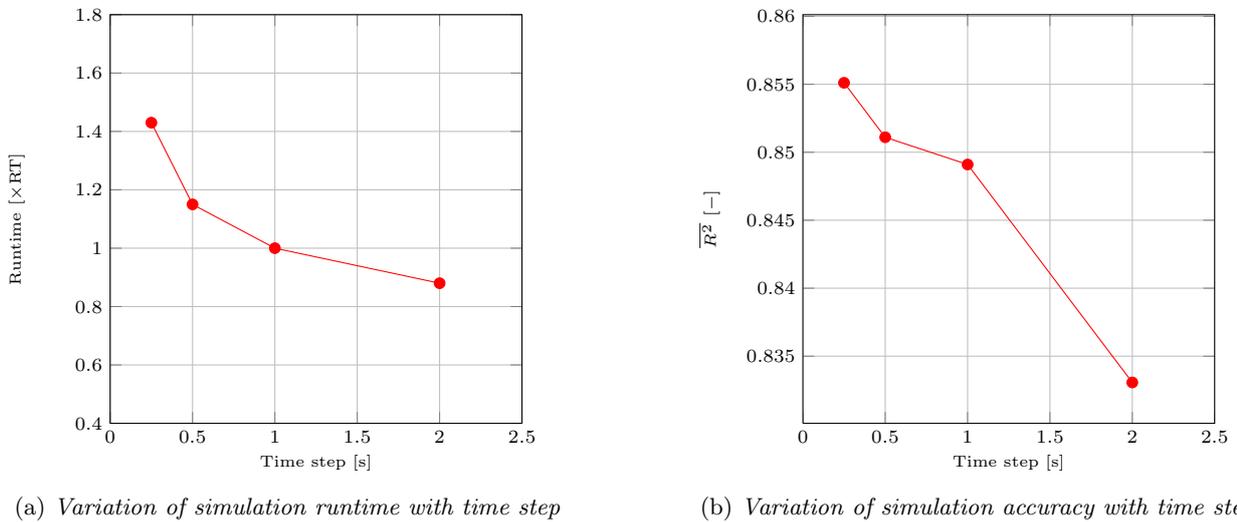


Figure 5.1

Figure 5.1a shows that simulation runtime drops by approx. 40% as a result from increasing the time step 8 times. Naturally, the accuracy of the simulation decreases accordingly with approx. 3%. A big reduction in runtime (60% of the total reduction) occurs between 0.25 and 1 s, Figure 5.1a. In this region of the time step, accuracy drops only approx. 20% of its total reduction. For this reason it is recommended to use time step of 1 s as it provides a good compromise between simulation run time and accuracy.

In relation to this point and on the topic of runtime in general, it must be pointed out, that runtime strongly depends on the computational resources and technical parameters of the machine(s) used to execute the solution. System characteristics of the machine used are available in Appendix G.

5.3 Recommendations for choice of software for transient analysis

A question is asked: *Which tool is better to use for transient analysis: the one developed in this thesis work, or the currently existing tool?*

The developed software provides obvious advantages in terms of accuracy, system integrity and authentic interaction of subsystems combined with effective 1D representation of fluid dynamics in the coolant circuit. Furthermore, the detailed engine model extends the application boundaries of the simulation providing possibility for investigations of fuel consumption. However, this comes at the price of increased simulation runtime. The currently existing tool, on the other hand provides fair accuracy, no proper physical description of the engine or coolant circuit, but its runtime is a factor of the one needed for the developed model. There is no universally best model. The choice which model to use should be made depending on the specific application.

For cooling systems-related problems, where interaction between engine and cooling system is of prime interest, and where the focus is on cooling performance, coolant circuit, etc. the detailed model developed in this thesis is recommended.

6 Conclusion

An integrated model for transient simulation including interactive, predictive models of cooling system and engine system with drive cycle input has been produced, calibrated, tested and results have been correlated with test measurements. The model provides:

- A flexible, modular architecture designed to facilitate easy exchange and integration of models between the respective departments in the organization
- Satisfactory consistency with test measurements on all examined system parameters with values for R^2 exceeding ≈ 0.8 (satisfying the acceptance criteria)
- Runtime effectively improved to approx. $1 \times RT$
- Potential in analyzing fuel consumption

All needs of this project in the context of cooling system simulations were satisfied by the functionality provided by GT-SUITE.

7 Future developments

The following areas have been identified for future development:

- Model verification with more drive cycles and correlation with test
- Improve the accuracy of each of the simulation sub-blocks in order to improve overall simulation accuracy
- Supply model of engine block with accurate data for internal heat transfer area and convective heat transfer coefficient dependent on flow
- Test and verify the transient operation of the coolant circuit at lower system temperatures, where the effect of the thermostat valve lift on the flow would be more noticeable. Supply more accurate data for discharge coefficients of thermostat as a function of valve lift. (Currently data from CFD is used. Actual test data is recommended.)
- Improve coolant pump behavior immediately after mode switching
- Add additional heat exchangers of interest to the system: e.g. oil cooler, compressor cooler, cab, heater, transmission cooler, etc. Care must be taken when adding extra volumes as runtime increases rapidly with additional flow volumes

A Appendix - System architecture

Due to secrecy reasons the contents of this table cannot be displayed in the public edition.

Table A.1: Method for acceptance

Area	Purpose	Method	Acceptance criteria
Engine	N/A	N/A	N/A
Cooling system	N/A	N/A	N/A
Complete engine and cooling installation	N/A	N/A	N/A

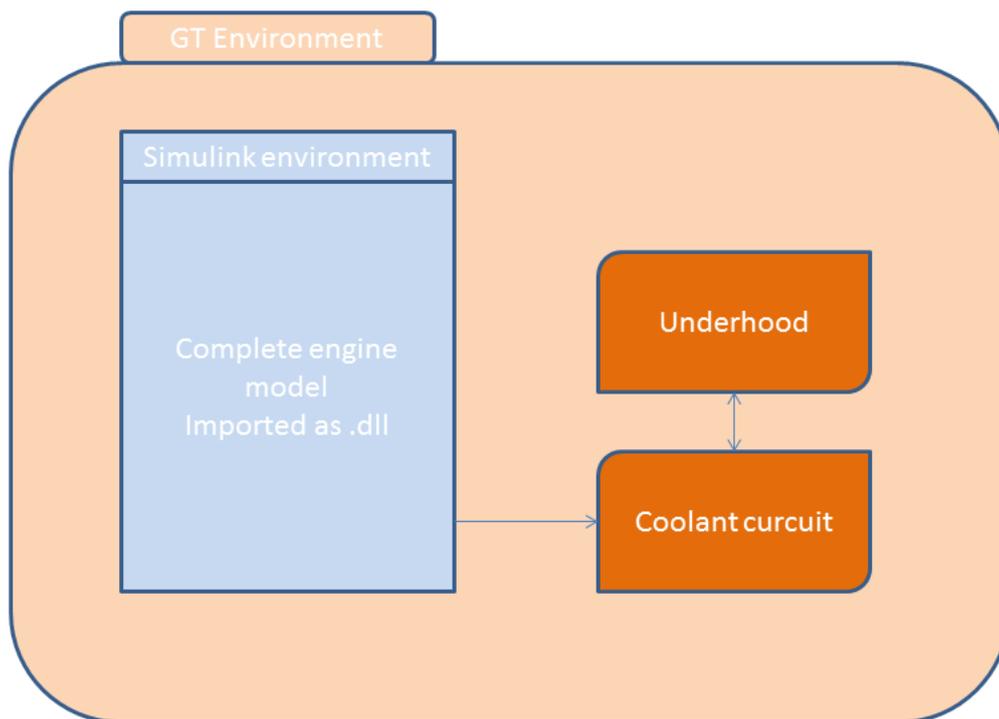


Figure A.1: Block diagram: GT - master, entire engine model is imported

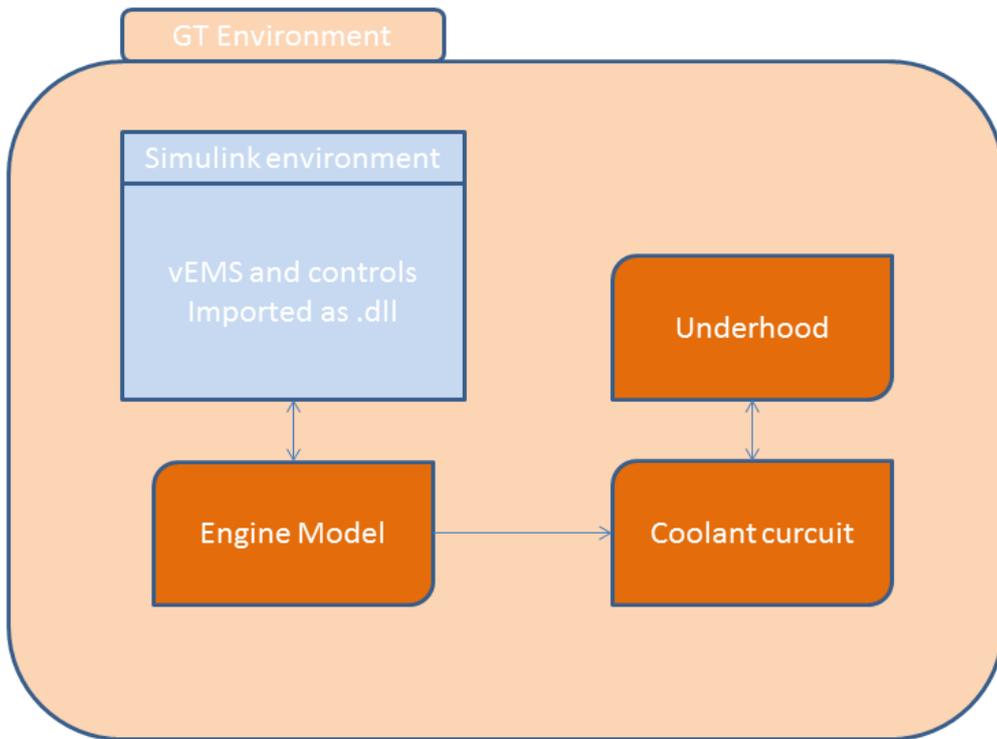


Figure A.2: Block diagram: GT - master, only vEMS is imported

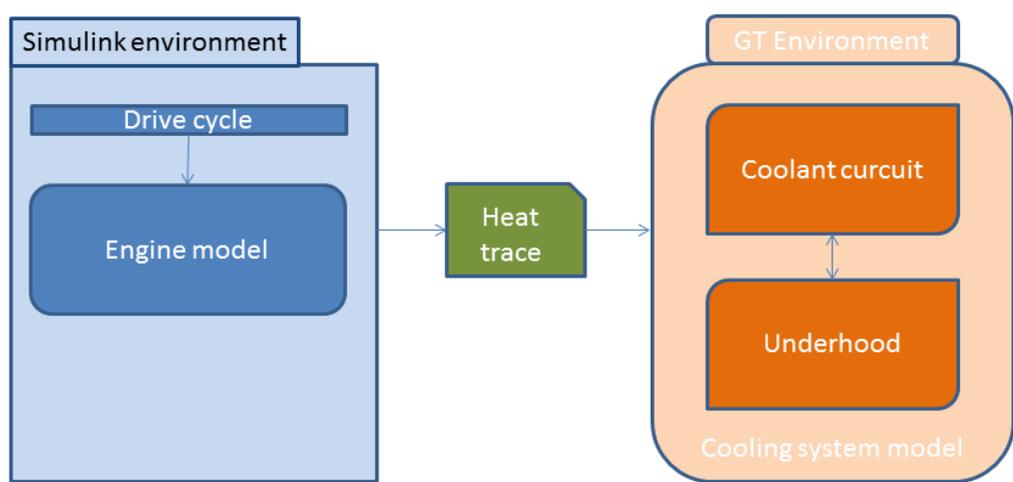


Figure A.3: Block diagram: Engine model and cooling system separate

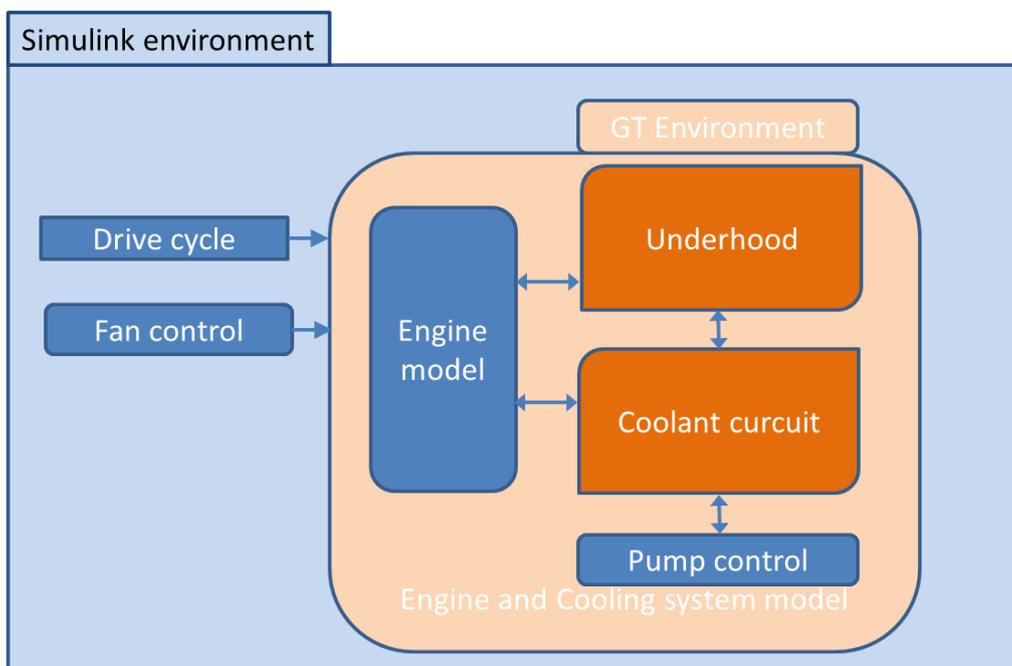


Figure A.4: Block diagram: Engine model and cooling system in a single GT file

B Appendix - Block diagrams

For secrecy reasons the input signals cannot be displayed in the public edition.

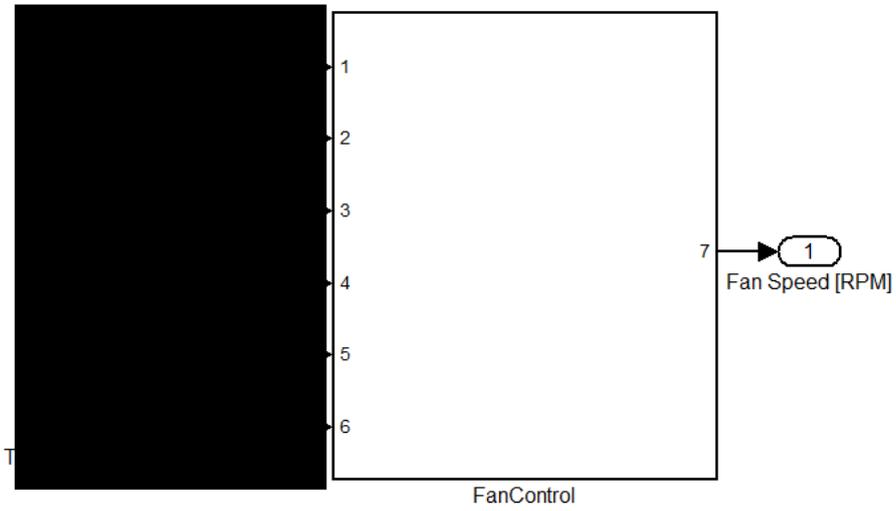


Figure B.1: *Block diagram: Fan control*

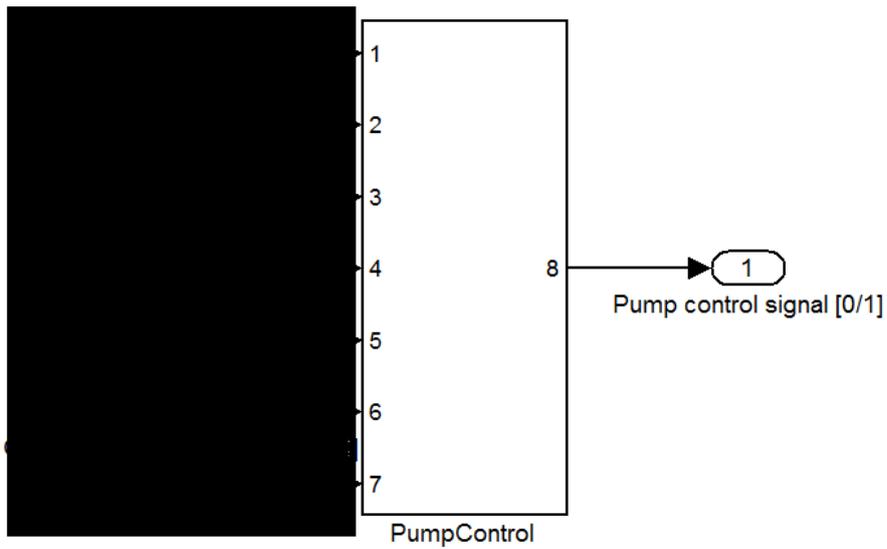


Figure B.2: *Block diagram: Pump control*

C Appendix - Implementation

For secrecy reasons this part of the report cannot be displayed in the public edition.

D Appendix - Results from Another tool for transient analysis

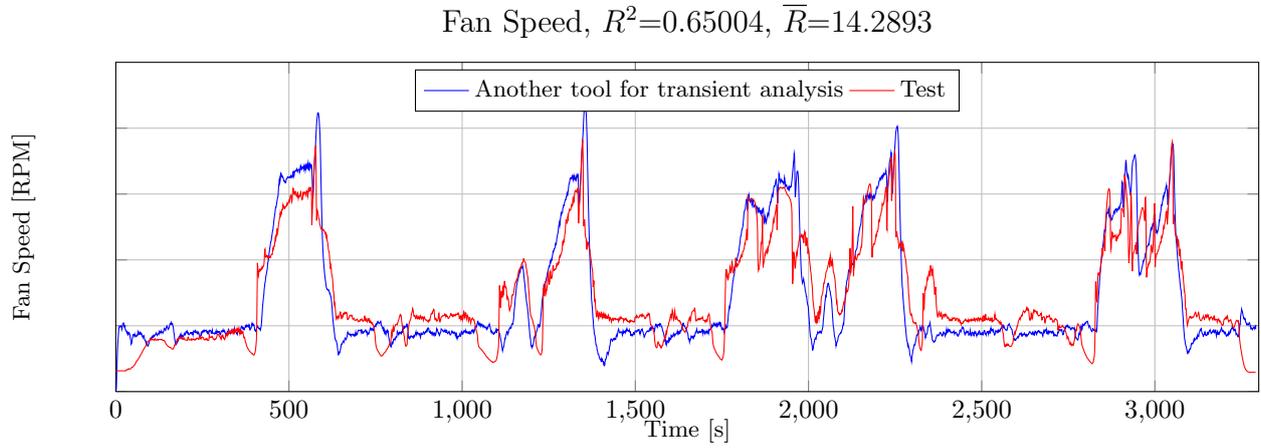


Figure D.1: *Plot: Fan speed, Another tool*

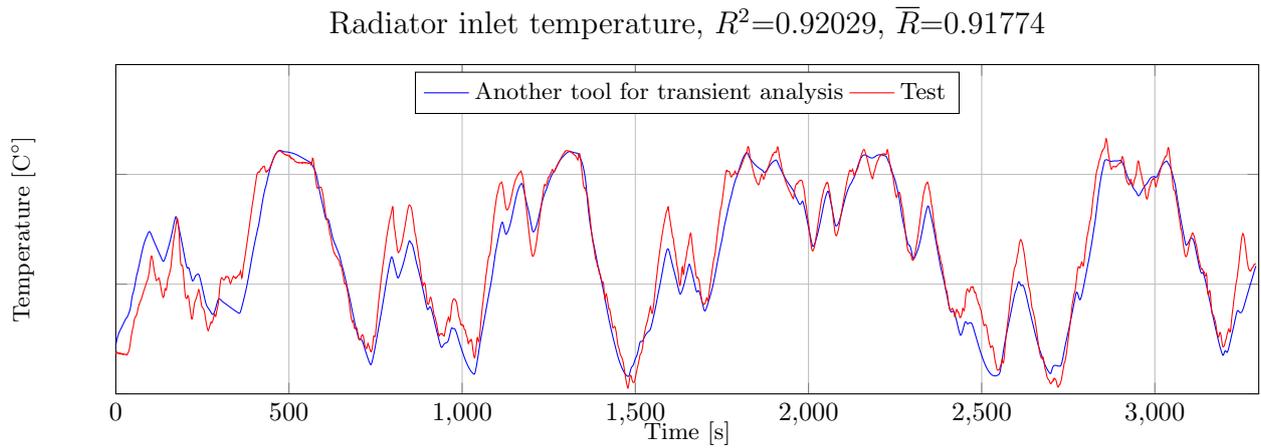


Figure D.2: *Plot: Radiator inlet temperature, Another tool for transient analysis*

Radiator outlet temperature, $R^2=0.67609$, $\bar{R}=3.06122$

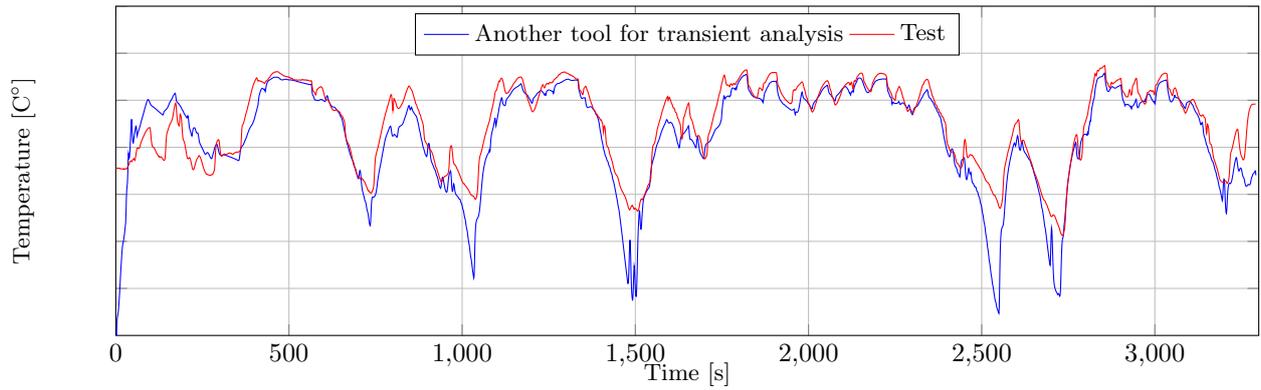


Figure D.3: Plot: Radiator outlet temperature, Another tool for transient analysis

Coolant flow through radiator, $R^2=0.80467$, $\bar{R}=0.2158$

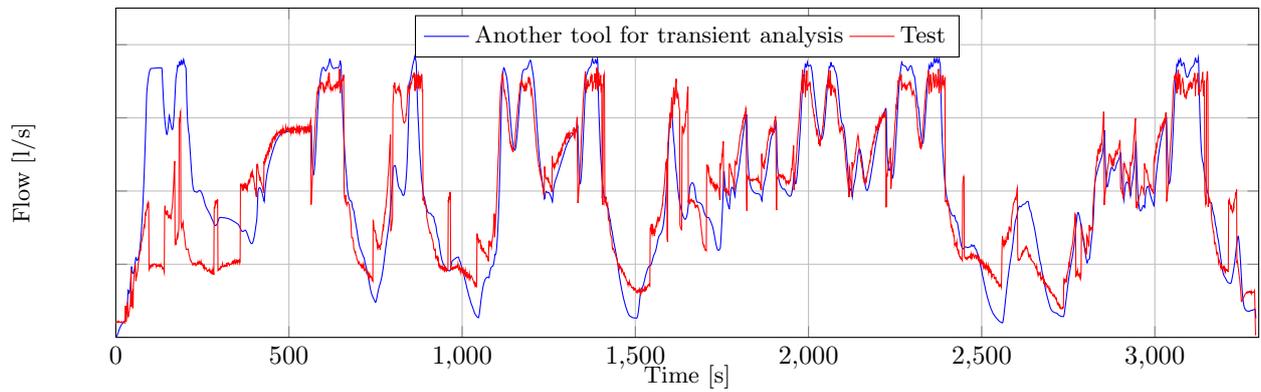


Figure D.4: Plot: Coolant flow through radiator, Another tool for transient analysis

Rejected heat through radiator, $R^2=-0.0507$, $\bar{R}=-7693.0849$

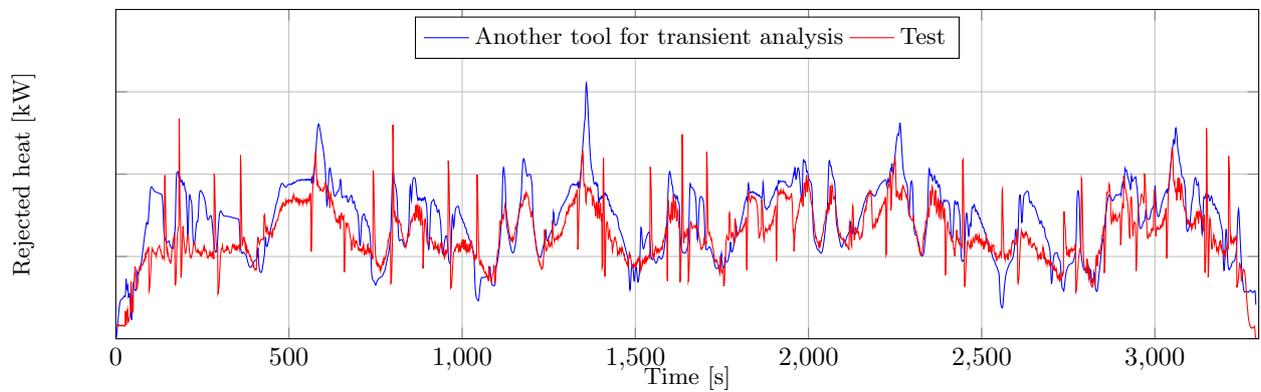


Figure D.5: Plot: Rejected heat through radiator, Another tool for transient analysis

Inlet charge air temperature CAC, $R^2=-0.87744$, $\bar{R}=40.29147$

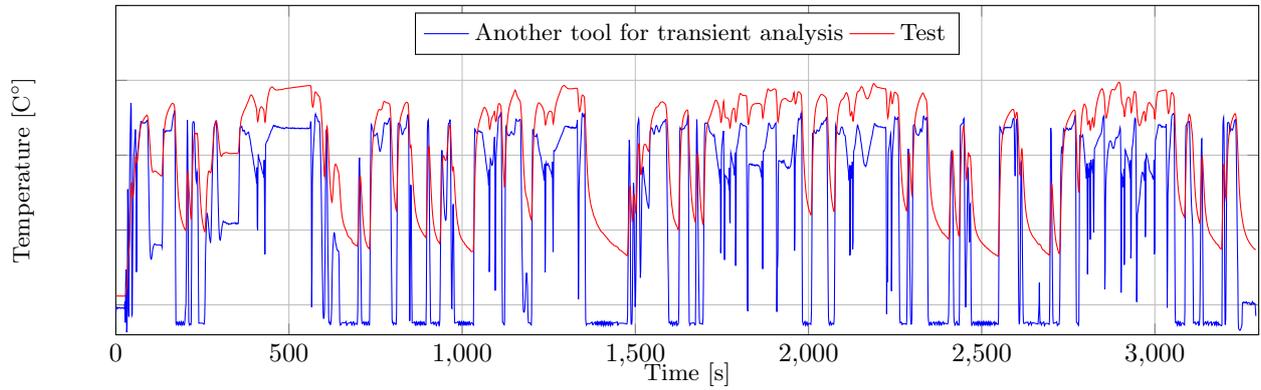


Figure D.6: Plot: CAC inlet temperature, Another tool for transient analysis

Outlet charge air temperature CAC, $R^2=0.53577$, $\bar{R}=2.16045$

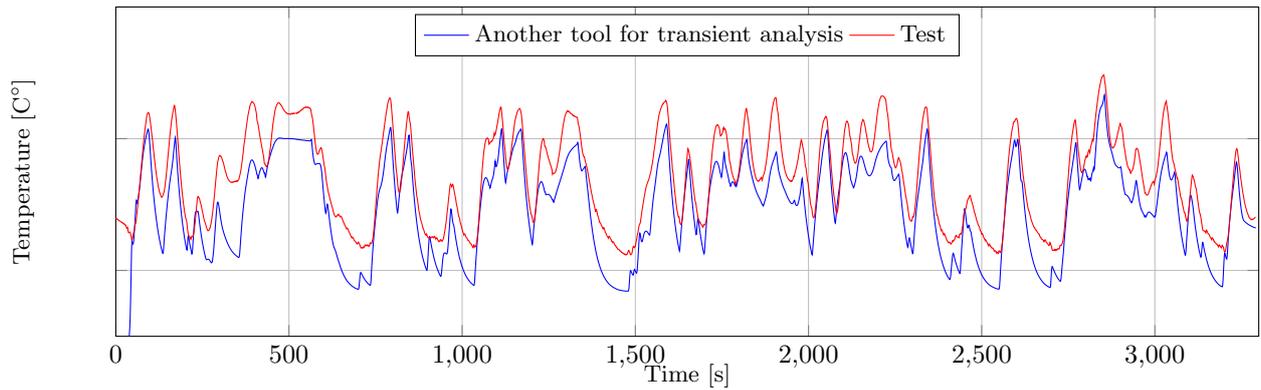


Figure D.7: Plot: CAC outlet temperature, Another tool for transient analysis

Charge air mass flow through CAC, $R^2=0.74559$, $\bar{R}=0.0245$

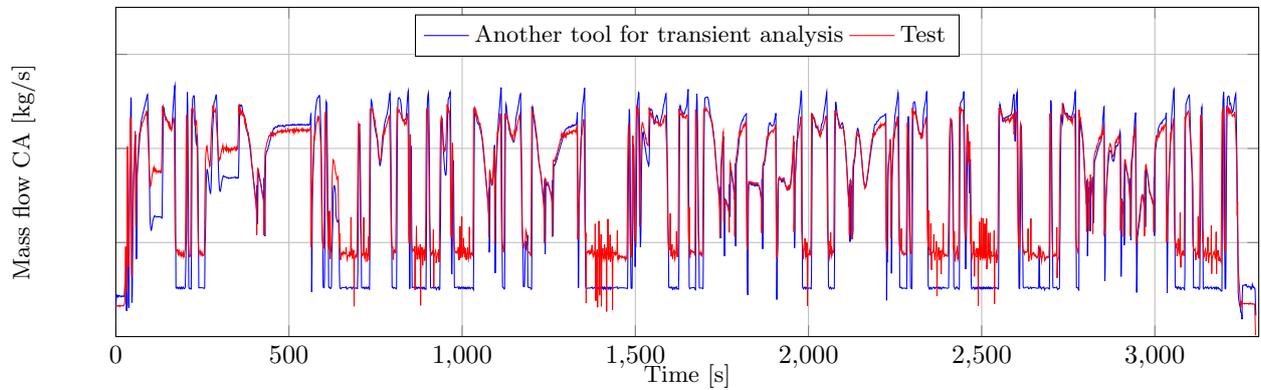
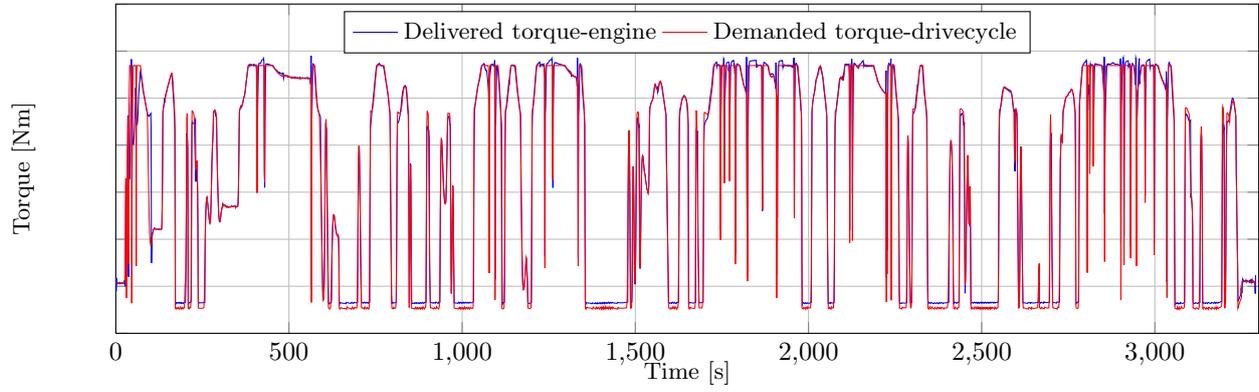


Figure D.8: Plot: Mass flow CAC, Another tool for transient analysis

E Appendix - Torque consistency

Comparison between demanded and delivered torque, $R^2=0.98503$, $\bar{R}=-11.68106$



Torque residuals (demanded - delivered torque)

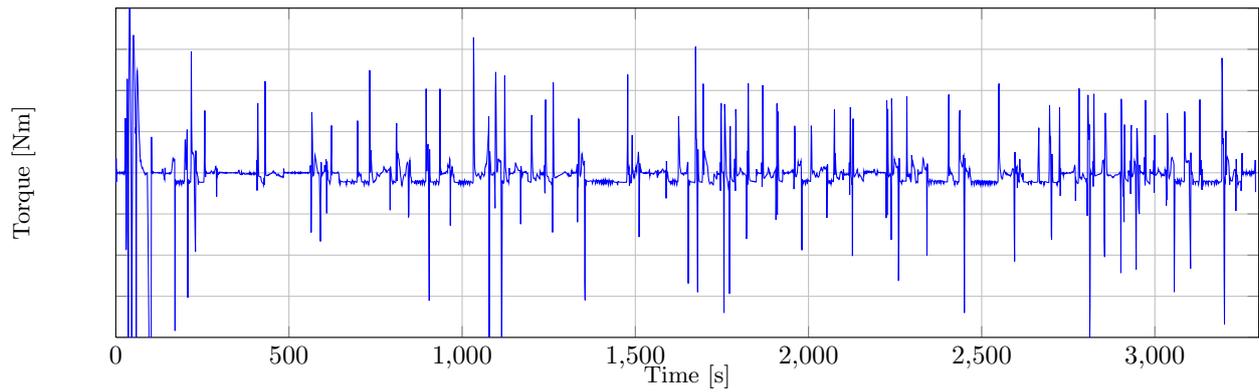


Figure E.1: *Plot: Torque consistency*

F Appendix - FlowPDropTableRef object

GT-SUITE manual [7] describes the object as follows: (cited from GT manual)

When FlowPDropTableRef object is used, a non-dimensional pressure loss coefficient and Reynolds number are calculated during simulation pre-processing for each point in the input data. During simulation the flow rate is imposed according to this non-dimensional relationship. By non-dimensionalizing the user defined pressure loss-to-flow relationship the pressure drop will respond adequately to changes in fluid temperature. For steady state, incompressible flow conditions, the pressure loss coefficient is calculated from the equation:

$$K = \frac{2\Delta P_{total}\rho A_{ref}^2}{\dot{m}^2} \quad (F.1)$$

where,

K = pressure loss coefficient

ρ = density of the fluid

A_{ref} = reference area

\dot{m} = mass flow rate

ΔP_{total} = difference in total pressure (total upstream pressure - total downstream pressure)

The Reynolds number is defined as:

$$Re = \frac{\dot{m}L_{ref}}{\mu A_{ref}} \quad (F.2)$$

where,

μ = dynamic viscosity

L_{ref} = Reference Length

G Appendix - Computer system characteristics

Processor: Intel(R) Core(TM) i7-2720QM CPU @ 2.20GHz

Installed memory (RAM) : 8.00GB

System type: 64-bit OS, Win7

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