Mean Value Modelling of the intake manifold temperature

Master's thesis
performed in **Vehicular Systems**

by
Anders Holmgren

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at **Linköpings universitet**

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Titel Medelvärdesmodellering av temperaturen i inloppsröret

Title Mean Value Modelling of the intake manifold temperature

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Sammanfattning
Abstract

The emission legislations and the new On Board Diagnostics (OBD) legislations are becoming more strict and making the demands on control and fault detection higher. One way to control and diagnose the engine is to use a control/diagnose strategy based on physical models and therefore better models are necessary. Also, to be competitive and meet the markets demand of higher power, longer durability and better fuel economy, the models needs to be improved continuously. In this thesis a mean value model of the intake system that predicts the charge air temperature has been developed. Three models of different complexity for the intercooler heat-exchanger have been investigated and validated with various results. The suggested intercooler heat-exchanger model is implemented in the mean value model of the intake system and the whole model is validated on three different data sets. The model predicts the intake manifold temperature with a maximum absolute error of $10.12K$.

Nyckelord
Keywords
mean value engine modelling, intercooler

Abstract

The emission legislations and the new On Board Diagnostics (OBD) legislations are becoming more strict and making the demands on control and fault detection higher. One way to control and diagnose the engine is to use a control/diagnose strategy based on physical models and therefore better models are necessary. Also, to be competitive and meet the markets demand of higher power, longer durability and better fuel economy, the models needs to be improved continuously. In this thesis a mean value model of the intake system that predicts the charge air temperature has been developed. Three models of different complexity for the intercooler heat-exchanger have been investigated and validated with various results. The suggested intercooler heat-exchanger model is implemented in the mean value model of the intake system and the whole model is validated on three different data sets. The model predicts the intake manifold temperature with a maximum absolute error of $10.12K$.

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Anders Holmgren
Södertälje, June 2005

Contents

Chapter 1

Introduction

This master's thesis was performed at Scania CV AB in Södertälje. The thesis describes an implementation of a temperature model for the intake system of a Scania DC16 V8 engine. The implementation is based on measurements made on board of an R-cab truck under different operating conditions.

1.1 Background

The emission legislations and the new On Board Diagnostics (OBD) legislations are becoming more strict and making the demands on control and fault detection higher. One way to control and diagnose the engine is to use a control/diagnose strategy based on physical models and therefore better models are necessary. To be competitive and meet the markets demand of higher power, better durability, better fuel economy and robustness against false alarms, it is important to have physical correct models, and these models needs to be improved.

1.2 Problem Formulation

The intake air temperature is the quantity that is to be modeled for diagnosis in this thesis. A MATLAB-SIMULINK Mean Value Model (MVM) of the intake system should be developed and validated. The test vehicle used for developing and validation of the model is a Scania R-cab truck with a DC16 V8 engine without Exhaust Gas Return (EGR) or Variable Geometry Turbo (VGT). The test vehicle is equipped with external sensors needed for the model development.

1.3 Objectives

The objectives are to construct an accurate and physically based model of the intake manifold temperature to be used for diagnosis of the boost temperature sensor. The model should be validated on a dataset different from the tuning data. The model objectives are:

- The model is to be developed in MatLab-Simulink
- The model should be based on physical relations as far as possible
- Inputs to the model should be signals available in the control unit

1.4 Target Group

The target group of this thesis is mainly people working at Scania CV, undergraduate and graduate science students. Knowledge in vehicular systems, fluid mechanics and thermodynamics increase the understanding.

Chapter 2

The Intake System

In this chapter the function of the intake system and its components will be described. The purpose is to give unfamiliar readers of this thesis a better understanding of the intake system and its components.

2.1 Components

The intake system includes the following components: Air-filter, compressor, intercooler and intake-manifold. Each of the components more or less contribute to effects such as: temperature changes, pressure changes and changes in flow. An illustrative sketch of the main components of the engine is shown In figure 2.1, where the intake system components are the non greyed. The sketch gives a picture of how the components interacts with each other. The components that have the greatest effect on the intake manifold temperature are the compressor, intercooler and the intake manifold.

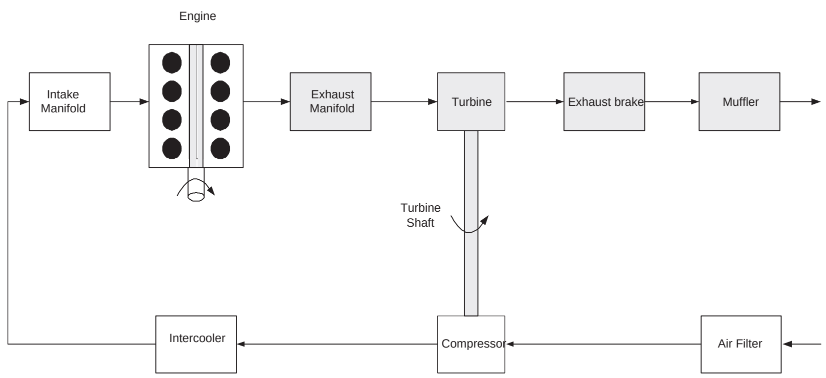


Figure 2.1: The main components of the conventional diesel engine.

2.1.1 Air Filter

A turbocharged engine consumes substantially more air than a normal conventional engine, therefore it is important to minimize intake restrictions. To ensure a smooth delivery of air to the compressor, the intake system includes a high-flow air filter with low-restriction tubing to deliver air from the atmosphere to the compressor. In order to keep the charge air temperatures after the compressor at a level as low as possible, it is also important that the air intake is placed in a position where the air has not yet been heated by the engine. On the test vehicle, the air-filter is mounted in front of the engine, right behind the air intake and prevents particles to pass and enter the compressor and other sensitive parts of the engine.

2.1.2 Compressor

The turbocharger used in the test vehicle is manufactured by Garret and is placed behind the engine centered between the two rows of cylinders. The compressor increases the density of the air, i.e. oxygen atoms which increases the power output and efficiency of the engine. A drawback is that it also increases the temperature of the air.

2.1.3 Intercooler

The purpose of having an intercooler after the compressor is to cool the very high temperature of the compressed air to ambient temperature and further increase the air density. Ideally, the temperature out of the intercooler is the same as the ambient temperature, but this is not the case in reality. The temperature out of the intercooler is a complex function of vehicle speed, fan speed, air mass-flows and the temperature in to the intercooler. Since this component directly affects the intake manifold temperature, three different models of predicting the temperature out of the intercooler have been investigated within this thesis. Scania uses an air cooled cross-flow intercooler with both fluids unmixed. It is placed at the front of the truck in a sandwich configuration with both the radiator and the air conditioning condenser.

2.1.4 Intake Manifold

The intake manifold is connected to the intercooler and is the last part of the intake system which feeds the cylinders with cold, high density fresh air. The intake manifold is made of aluminum and some heat transfer occur from the wall of the intake manifold to the intake air.

Chapter 3

Modelling

This chapter describes the model structure and how the modelling of the intake system has been implemented.

To be able to model the intake manifold temperature, the temperature after each component of the intake system needs to be modeled, since the only available temperature signal from the engine control unit is ”ambient temperature”. This means that a temperature model for each component of the intake system needs to be developed.

3.1 Model Structure

The model that has been developed and implemented in MatLab-Simulink is a component based mean-value model including sub-models for each component of the intake system. A mean value model describes the average behavior of the system, which means that no variations within one cycle are covered by the model. This implies that the model is only valid for time intervals far greater than one cycle [4]. The dynamic behavior has been implemented with two control volumes from a toolbox called MVEM-LIBRARY, see [5]. From the MVEM-LIBRARY, one incompressible restriction have also been implemented to model the restriction introducing a pressure drop over the intercooler. The the control volumes contain filling and emptying dynamics while the incompressible restriction determines a mass-flow with a temperature.
Assumptions have been made that the air-filter does not contribute to any effects on the either the temperature, pressure or the flow. This assumption have been verified by measurements with a good accuracy for the specific test vehicle. Figure 3.1 shows how the different components are connected to each other and where the model dynamics exists.

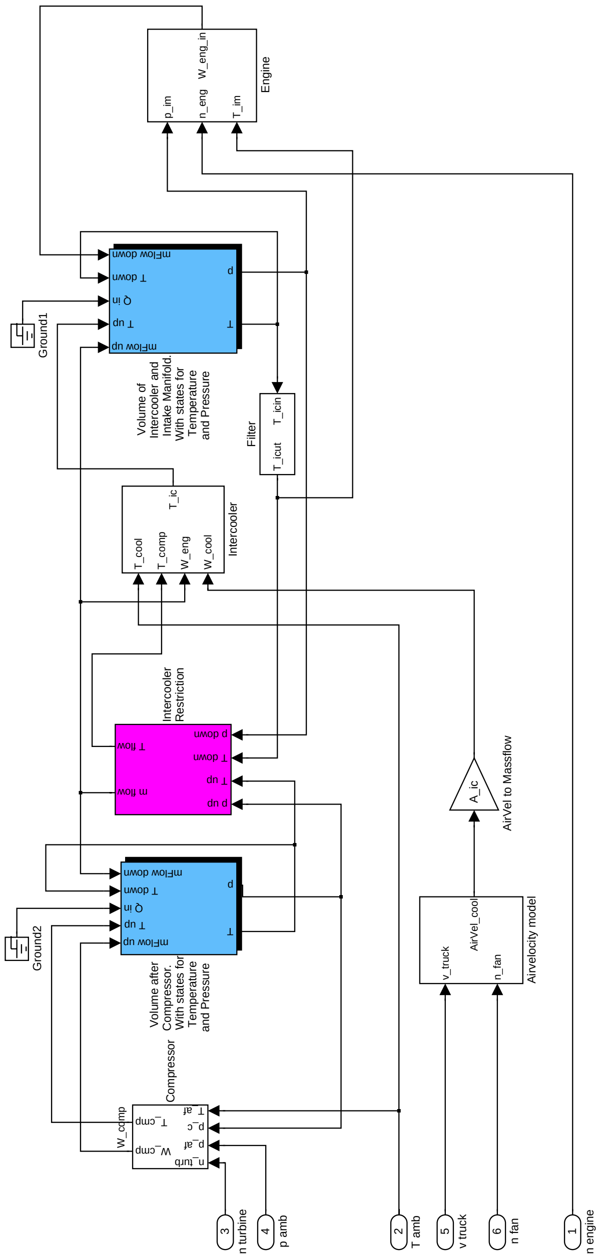


Figure 3.1: Intake System MVM structure.

3.1.1 MVEM restriction block

The incompressible restriction block from the MVEM-LIBRARY uses the downstream and upstream pressures as inputs and the mass-flow through the restriction as the output. The pressure loss over the restriction are described using only one parameter H_{res}, the restriction coefficient, see [9]. The following equations describes the MVEM restriction block.

$$\Delta p_{res} = p_{us} - p_{ds} = H_{res} \frac{T_{us} W_{res}^2}{p_{us}} \tag{3.1}$$

where Δp_{res} is the pressure loss over the restriction and p_{us} and p_{ds} are the upstream and downstream pressure, respectively.

Solving W_{res} from Eq. 3.1 gives:

$$W_{res} = \sqrt{p_{us} \frac{\Delta p_{res}}{H_{res} T_{us}}} \tag{3.2}$$

Since the derivative of Eq. 3.2 with respect to Δp_{res} approaches infinity as Δp_{res} approaches zero, the function is for $0 \leq \Delta p_{res} \leq p_{lin}$ linearized in the MVEM block to the following equation.

$$W_{res} = \sqrt{\frac{p_{us}}{H_{res} T_{ds}}} \frac{\Delta p_{ds}}{\sqrt{p_{lin}}} \tag{3.3}$$

For causality, the simplification that flows only runs in forward direction in the model has been made, W_{res} is set to 0 for $\Delta p_{res} \leq 0$.

3.1.2 MVEM control volume

The control volume from the MVEM-LIBRARY is a two state control volume with states for pressure p and temperature T. The control volume has a fixed volume V and the change of mass within the control volume is determined by the air mass-flows in and out of the control volume.

$$\frac{dm}{dt} = W_{in} - W_{out} \tag{3.4}$$

Within the control volume the energy is conserved and stored. But energy can be transferred to or from the control volume through the air mass-flows in and out or by heat transfer $\dot{Q}$. The first law of energy conservation gives the rate of change in internal energy $\frac{dU}{dt}$, see [9].

$$\frac{dU}{dt} = \dot{H}_{in} - \dot{H}_{out} + \dot{Q} \tag{3.5}$$

where, $\dot{H}_{in}$ and $\dot{H}_{out}$ are the enthalpy flows in and out from the control volume.

$$\dot{H}_{in} = W_{in} c_p T_{in} \tag{3.6}$$

$$\dot{H}_{out} = W_{out} c_p T_{out} \tag{3.7}$$

Due to difficulties in measuring quantities like energy and mass it is desirable to express the states for temperature and pressure as differential equations. To be able to do that, the following assumptions are made.

- the gas inside the control volume is ideal
- c_v and c_p are constant (i.e. $R = c_p - c_v$)
- $T_{out} = T$

With these assumptions the pressure can be determined by the ideal gas law

$$pV = mRT \tag{3.8}$$

and the temperature can be determined from the internal energy.

$$U = mu(T) = mc_v T \tag{3.9}$$

By differentiating the ideal gas law Eq. 3.8

$$V\frac{dp}{dt} = RT\frac{dm}{dt} + mR\frac{dT}{dt} \tag{3.10}$$

and solving $\frac{dp}{dt}$ gives the following expression of the state for pressure in terms of the state for temperature.

$$\frac{dp}{dt} = \frac{RT}{V}\frac{dm}{dt} + \frac{mR}{V}\frac{dT}{dt} \tag{3.11}$$

By using Eq. 3.4 and 3.8, Eq 3.11 can be rewritten as

$$\frac{dp}{dt} = \frac{RT}{V}(W_{in} - W_{out}) + \frac{p}{T}\frac{dT}{dt} \tag{3.12}$$

which is the state for pressure in terms of the state for temperature and other measurable quantities.

By differentiating Eq. 3.9

$$\frac{dU}{dt} = \frac{dm}{dt}c_v T + mc_v\frac{dT}{dt} \tag{3.13}$$

and using Eq. 3.4, 3.5, 3.6 and 3.7, this expression can be rewritten as.

$$W_{in}c_p T_{in} - W_{out}c_p T_{out} + \dot{Q} = W_{in}c_v T - W_{out}c_v T + mc_v\frac{dT}{dt} \tag{3.14}$$

Solving $\frac{dT}{dt}$ and substituting $c_p = R + c_v$ Eq. 3.14 becomes

$$\begin{aligned}\frac{dT}{dt} = \frac{1}{mc_v}(W_{in}c_v(T_{in} - T)) + \frac{1}{mc_v}(R(W_{in}T_{in} - W_{out}T_{out}))+ \\ \frac{1}{mc_v}(W_{out}c_v(T - T_{out}) + \dot{Q})\end{aligned} \tag{3.15}$$

For $T_{out} = T$, Eq. 3.15 can be rewritten as

$$\frac{dT}{dt} = \frac{1}{mc_v}\left(W_{in}c_v(T_{in} - T) + R(W_{in}T_{in} - W_{out}T) + \dot{Q}\right) \qquad (3.16)$$

To summarize, the states for temperature and pressure in terms of differential equations are,

$$\frac{dT}{dt} = \frac{1}{mc_v}\left(W_{in}c_v(T_{in} - T) + R(W_{in}T_{in} - W_{out}T) + \dot{Q}\right) \qquad (3.17)$$

$$\frac{dp}{dt} = \frac{RT}{V}(W_{in} - W_{out}) + \frac{p}{T}\frac{dT}{dt} \qquad (3.18)$$

The states, parameters, constants, inputs and outputs to the MVEM control volume are described in Table 3.1.

Table 3.1: MVEM control volume, signals and descriptions

Name	Type	Description
p	State	Pressure
T	State	Temperature
V	Parameter	Volume
R	Constant	Gas constant
c_v	Constant	Specific heat at constant volume
T_{us}	Input	Temperature in
W_{us}	Input	Mass-flow in
W_{ds}	Input	Mass-flow out
$\dot{Q}$	Input	Heat transfer
$\frac{dT}{dt}$	Output	Rate of temperature change
$\frac{dp}{dt}$	Output	Rate of pressure change

3.2 Compressor Model

The modelling of the compressor is based on static maps describing the efficiency and mass flow out of the compressor. Inputs to the maps are pressure ratio over the compressor and turbine speed.

$$W_{comp} = f_{W_{comp}}\left(\frac{p_{comp}}{p_{af}}, n_{turb}\right) \qquad (3.19)$$

$$\eta_{comp} = f_{\eta_{comp}}\left(\frac{p_{comp}}{p_{af}}, n_{turb}\right) \qquad (3.20)$$

The compressor maps are made by the manufacturer of the turbocharger and are sufficiently accurate to be used for modelling the temperature increase of the air passing through the compressor. If the inputs to the compressor maps at some point exceeds the coverage of the maps, the end values of the maps are used.
The compressor outlet temperature is modeled as Eq. 3.21, described in [8] and have been used in [1].

$$T_{comp} = T_{af} \left(1 + \frac{\prod_{comp}^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_{comp}} \right) \tag{3.21}$$

Where $\prod_{comp}$ is the pressure ratio over the compressor, γ_a the ratio of the specific heats c_p and c_v. The constant,inputs and output signals for the compressor outlet temperature model are described in Table 3.2.

Table 3.2: Compressor outlet temperature model, signals and descriptions

Name	Type	Description
γ_a	Constant	Ratio of specific heat $\frac{c_p}{c_v}$
η_{comp}	Input, map output	Compressor efficiency
$\prod_{comp}$	Input	Pressure ratio $\frac{p_{comp}}{p_{af}}$
n_{turb}	Input	Turbine speed
W_{comp}	Map output	Air mass-flow after compressor
T_{comp}	Output	Temperature after compressor

A control volume from the MVEM-LIBRARY is placed between the compressor and intercooler representing the volume of the pipe connecting the compressor with the intercooler. This control volume puts dynamics into the system, with states for pressure and temperature. An assumption have been made that there is no heat transfer to the volume and therefore $\dot{Q}$ is set to zero.

3.3 Intercooler Model

The Intercooler is a cross-flow intercooler with both of the fluids unmixed. It it modeled as an incompressible restriction and a heat exchanger. The intercooler also have a volume, but this volume is modeled together with the volume of the intake manifold.

3.3.1 Intercooler restriction model

The intercooler restriction model are as described earlier modeled as an incompressible restriction from MVEM-LIBRARY. The pressure after the compressor and the pressure after the intercooler are inputs to the restriction model and the mass-flow through the intercooler is the output. Equations for the intercooler restriction model follows:

$$\Delta p_{ic} = p_{comp} - p_{ic} = H_{ic} \frac{T_{comp} W_{ic}^2}{p_{comp}} \tag{3.22}$$

$$W_{ic} = \sqrt{p_{comp} \frac{\Delta p_{ic}}{H_{ic} T_{comp}}} \tag{3.23}$$

The function is linearized for $0 \leq \Delta p_{ic} \leq p_{lin}$, to:

$$W_{ic} = \sqrt{\frac{p_{comp}}{H_{ic} T_{comp}}} \frac{\Delta p_{ic}}{\sqrt{p_{lin}}} \tag{3.24}$$

For causality, W_{ic} is set to 0 for $\Delta p_{ic} \leq 0$. H_{ic} is determined by the method of least squares on measured data and p_{lin} is set to $100\ Pa$. The parameters, input and output signals for the intercooler restriction model are described in Table 3.3.

Table 3.3: Intercooler restriction model, signals and descriptions

Name	Type	Description
H_{ic}	Parameter	Restriction coefficient
p_{lin}	Parameter	Linearization pressure
T_{comp}	Input	Temperature after compressor
p_{comp}	Input	Pressure after compressor
T_{ic}	Input	Temperature after intercooler
p_{ic}	Input	Pressure after intercooler
W_{ic}	Output	Air mass-flow after intercooler

3.3.2 Intercooler heat exchanger model

The main problem of modelling of the intake system is to model the intercooler heat exchanger. Three different models of the heat exchanger have been investigated.

- NTU model
- Linear Regression model
- Use of map data from manufacturer

Both the NTU model and the linear regression model use the standard heat exchanger expression for the efficiency of the intercooler described in [9].

$$\eta_{ic} = \frac{T_{comp} - T_{ic}}{T_{comp} - T_{cool}} \tag{3.25}$$

Rearranging this expression gives the intercooler outlet temperature in terms of intercooler efficiency and temperature.

$$T_{ic} = T_{comp} + \eta_{ic}(T_{comp} - T_{cool}) \tag{3.26}$$

The difference between the NTU model and the linear regression model is the way of modelling the intercooler efficiency η_{ic}.
The intercooler heat exchanger acts as a first order low-pass filter with a time constant τ of $15s$. This means that when the operation conditions for the intercooler are changed with a step, the response have reached 63% of its end value after $15s$. This have been verified by measurements and a low-pass filter have been implemented after the control volume to model this behavior. For modelling purposes a start value for the intercooler outlet temperature have been set to $293K$.

NTU Model

The NTU model is a model that utilizes the NTU method based on the effectiveness of the heat exchanger in transferring a given amount of heat and are thoroughly described in [7]. The NTU model of a cross-flow heat exchanger with both fluids unmixed consists of the following equations described in [9].

$$T_{ic} = T_{comp} + \eta_{ic}(T_{comp} - T_{cool}) \tag{3.27}$$

$$\eta_{ic} = 1 - e^{\frac{e^{-CN^{0.78}}-1}{CN^{-0.22}}} \tag{3.28}$$

$$N = \frac{UA}{c_{p,air}W_{ic}} = \frac{K}{c_{p,air}} W_{ic}^{-0.2} u_i^{-0.5} \tag{3.29}$$

$$u_i = 2.3937 \cdot 10^{-7} \left(\frac{T_{comp} + T_{cool}}{2} \right)^{0.7617} \tag{3.30}$$

$$C = \frac{W_{ic}}{W_{cool}} \tag{3.31}$$

The variable N is called the number of transfer units (NTU). The parameter K is determined from a least square fit to measured data and should be a constant. The data used for determining K is a mapped set of measured data at static conditions. Since the data is measured onboard of a truck it was hard to find static conditions and the constant K did not remain constant. K varied between 0.007 and 0.020 and a mean value of 0.012 was used.

The parameter, inputs and output signals for the NTU heat exchanger model are described in Table 3.4.

Table 3.4: NTU model, signals and descriptions

Name	Type	Description
K	Parameter	-
T_{comp}	Input	Temperature after compressor
T_{cool}	Input	Temperature of cooling air
W_{ic}	Input	Air Mass-flow after intercooler
W_{cool}	Input	Cooling Air Mass-flow through the intercooler
T_{ic}	Output	Temperature after intercooler

Linear Regression Model

In the linear regression model the regressors are inspired by the NTU model. It has been shown in [9] that this model performs well for modelling the intercooler heat exchanger, and therefore this modelling approach has been investigated.

The following equations are used for modelling the intercooler outlet temperature with the linear regression model.

$$T_{ic} = T_{comp} + \eta_{ic}(T_{comp} - T_{cool}) \tag{3.32}$$

$$\eta_{ic} = a_0 + a_1 \left(\frac{T_{comp} + T_{cool}}{2} \right) + a_2 W_{ic} + a_3 \frac{W_{ic}}{W_{cool}} \tag{3.33}$$

The parameters a_0-a_3 is determined by a least square fit to a mapped set of measured data in static conditions. Like the NTU model, problems occur when applying the least square fit to on board measured data where the data is not from completely static conditions.

The parameters, inputs and output signals for the Linear regression heat exchanger model are described in Table 3.5.

Table 3.5: Lin.Reg model, signals and descriptions

Name	Type	Description
a_0-a_3	Parameters	-
T_{comp}	Input	Temperature after compressor
T_{cool}	Input	Temperature of cooling air
W_{ic}	Input	Air Mass-flow after intercooler
W_{cool}	Input	Cooling Air Mass-flow through the intercooler
T_{ic}	Output	Temperature after intercooler

Map Data Model

The map data model is based on mapped data from the manufacturer of the intercooler. This map originally consists of a 4 by 4 matrix presenting removed energy from the fluid with respect to the mass-flow of the fluid and the mass-flow of the cooling air at constant entrance temperatures. This map has been modified and extrapolated since the original map doesn't cover all operating air mass-flows. The modification is that instead of removed energy the map contains removed energy per ΔT, where ΔT is the difference between entrance temperatures T_{comp} and T_{cool}. The extrapolation of the original map have been done using a MATLAB script developed by an engineer at Scania. The script utilizes the NTU method and optimizes the parameters within this method to fit the original map. After the parameters have been determined, the script extrapolates this map to cover the desired operating air mass-flows. Equations for modelling the intercooler outlet temperature with the mapped data model are:

$$\dot{Q}[W/K] = f_{map,ic}(W_{ic}, W_{cool}) = W_{ic}c_{p,air}(T_{comp} - T_{ic})(T_{comp} - T_{cool}) \tag{3.34}$$

$$T_{ic} = T_{comp} - \frac{f_{map,ic}(W_{ic}, W_{cool})(T_{comp} - T_{cool})}{W_{ic}c_{p,air}} \tag{3.35}$$

The inputs and output signals for the Map Data heat exchanger model are described in Table 3.6.

Table 3.6: Map Data model, signals and descriptions

Name	Type	Description
T_{comp}	Input	Temperature after compressor
T_{cool}	Input	Temperature of cooling air
W_{ic}	Input	Air Mass-flow after intercooler
W_{cool}	Input	Cooling Air Mass-flow through the intercooler
T_{ic}	Output	Temperature after intercooler

3.4 Cooling air mass-flow model

Since all described heat exchanger models rely on the cooling air mass-flow to predict a correct outlet temperature from the intercooler, a model of the cooling air mass-flow becomes crucial. Signals from the control unit that can be used as input to the cooling air mass-flow model are the velocity of the truck and the speed of the cooling fan. Four different static and linear modelling approaches of the cooling air velocity have been investigated.

Cooling air velocity model 1:

$$v_{air,cool} = a_0 + a_1 n_{fan} + a_2 v_{truck} \tag{3.36}$$

Cooling air velocity model 2:

$$v_{air,cool} = a_0 + a_1 n_{fan} + a_2 v_{truck} + a_3 \frac{v_{truck}}{n_{fan}} \tag{3.37}$$

Cooling air velocity model 3:

$$v_{air,cool} = a_1 n_{fan} + a_2 v_{truck} \tag{3.38}$$

Cooling air velocity model 4:

$$v_{air,cool} = a_1 n_{fan} + a_2 v_{truck} + a_3 \frac{v_{truck}}{n_{fan}} \tag{3.39}$$

The parameters a_0 - a_3 has been determined using the method of least squares on a mapped data set in static conditions. Since the ratio $\frac{v_{truck}}{n_{fan}}$ will approach infinity when n_{fan} approaches zero, n_{fan} is set to 100 rpm for fan speeds lower than 100 rpm. In practice, the fan speed will not be lower than 150 rpm. Together with the intercooler surface area the cooling air mass-flow becomes:

$$W_{cool} = v_{air,cool} \cdot A_{ic} \cdot \rho_{air} \tag{3.40}$$

The parameters,inputs and output signals for the cooling air mass-flow model are described in table 3.7.

Table 3.7: Cooling air mass-flow model, signals and descriptions

Name	Type	Description
a_0-a_3	Parameters	-
A_{ic}	Parameter	Intercooler surface area
ρ_{air}	Parameter	Air density
v_{truck}	Input	Truck speed
n_{fan}	Input	Fan speed
W_{cool}	Output	Cooling air mass-flow through intercooler

3.5 Intake Manifold

The intake manifold is modeled with a second control volume from the MVEM-LIBRARY. This control volume is representing the combined volume for the intercooler and the intake manifold, introducing states for pressure and temperature. The parameter V in this control volume is the physical correct volume of the intercooler and intake manifold together. Assumptions have been made that there is no heat transfer from the intake manifold walls to the gas inside this volume and therefore $\dot{Q}$ is set to zero here as well. The upstream air mass-flow input to this model is the air mass-flow out from the intercooler W_{ic}, downstream air mass-flow is the engine cylinder air mass-flow $W_{eng,in}$ and the upstream temperature is the intercooler outlet temperature T_{ic}.

The engine cylinder air mass-flow depends on the engine speed, intake manifold pressure, intake manifold temperature and the volumetric efficiency η_{vol} of the engine. η_{vol} is the efficiency of the engine to induct air and is defined as the ratio between the actual volume of inducted air divided by the theoretical volume of inducted air into the engine [4]. The modelling is based on a map describing the volumetric efficiency of the engine. Inputs to the map are engine speed and the intake manifold temperature.

$$\eta_{vol} = f_{\eta_{vol}}(n_{eng}, T_{im}) \tag{3.41}$$

The following equations have been used for modelling of the air mass flow into the engine

$$W_{im} = W_{eng,in} \tag{3.42}$$

$$W_{eng,in} = \eta_{vol} \frac{V_d n_{eng} p_{im} N_{cyl}}{60 N_r R_{im} T_{im}} \tag{3.43}$$

and are also described in [4].

The parameters, inputs and output signals for the intake manifold model are described in Table 3.8.

Table 3.8: Intake manifold model, signals and descriptions

Name	Type	Description
N_r	Parameter	Number of revolutions
N_{cyl}	Parameter	Number of cylinders
V_d	Parameter	Displacement volume
R_{im}	Constant	Gas constant
n_{eng}	Input	Engine speed
T_{im}	Input	Intake manifold temperature
P_{im}	Input	Intake manifold pressure
η_{vol}	Input, map output	Volumetric efficiency
$W_{eng,in}$	Output	Engine cylinder air mass-flow

Chapter 4

Measurement Setup

Measurements have been carried out to get data both to tune and validate the models. The measurements have been done on board on the test vehicle under different on road operating conditions. One of the objectives of this thesis is to build a model with available input signals from the control unit. As there are few sensor available, some external sensors had to be installed in purpose to be modeled and/or to verify the model with. Both signals from the control unit and signals from the external sensors were sampled and recorded with the measuring system Vision from ATI. All measured signals are presented in table 4.1 and the measurement setup can be seen in Figure 4.1.

4.1 External Sensors

The following types of external sensors have been mounted: pressure sensors, temperature sensors, inductive angular velocity sensors and sensors for the air-velocity of the cooling air passing through the intercooler.

4.1.1 Pressure Sensors

The pressure sensors used for the external pressure signals are manufactured by Kistler. They are able to measure frequencies up to 30 kHz and can handle temperatures up to 140 $^{\circ}C$. Two different models of the Kistler pressure sensor have been used, one with the range 0-5 bar (0-5 V) and one with the range 0-10 bar (0-5 V).

4.1.2 Temperature Sensors

The temperature sensors that have been used for the external temperature signals are $0.5mm$ thermocouples of type K. They can handle a temperature range of 253 to $1423K$ with a accuracy of $1.3K$ up to $415K$ and $\pm 0.3\%$ over

Table 4.1: Measured Signals

Pressure	Description	Sensor
T_{amb}	Ambient temp [K]	From control unit
T_{boost}	Boost temp [K]	From control unit
T_{ai}	Temp at air-intake [K]	Thermocouple
T_{af}	Temp after air-filter [K]	Thermocouple
T_{comp}	Temp after compressor [K]	Thermocouple
T_{ic}	Temp after intercooler [K]	Thermocouple
$T_{atboost}$	Temp at prod. sensor for T_{boost} [K]	Thermocouple
$T_{cool_{1-9}}$	Temp of the cooling air [K]	Thermocouples
p_{amb}	Ambient pressure [bar]	From control unit
p_{boost}	Boost pressure [bar]	From control unit
p_{ai}	Pressure at air-intake [bar]	Kistler
p_{af}	Pressure after air-filter [bar]	Kistler
p_{comp}	Pressure after compressor [bar]	Kistler
p_{ic}	Pressure after intercooler [bar]	Kistler
n_{eng}	Engine speed [rpm]	From control unit
n_{turb}	Turbine speed [rpm]	Inductive sensor
n_{fan}	Speed of the cooling fan [rpm]	Inductive sensor
$v_{air,cool_{1-9}}$	Cooling air velocity [m/s]	Kurtz

$415K$ [3]. The thermocouples were mounted without encapsulation and with the tips at the best available spots. The time constant of $0.5mm$ thermocouples are according to [2] $1.4s$ and have been verified by measurements.

4.1.3 Angular Velocity Sensors

Two angular velocity sensors have been used to measure the angular velocity of the turbine/compressor and the cooling fan. The sensor for measuring the turbine angular velocity is a induction revolution sensor from Micro Epsilon, custom built for a Garret turbo-charger. When the turbine shaft rotates, the sensor induces pulses through a coaxial cable connected to a converter included by the manufacturer of the sensor. The converter generates a voltage representing the compressor angular velocity in rpm. The sensor used for the measurement of the cooling fan angular velocity was also an inductive sensor, installed close to the wings of the cooling fan. On one of the wings a metal plate were mounted so that the inductive sensor detects and generates one pulse through a coaxial cable for each revolution of the fan. The coaxial cable were then connected to an external frequency/Voltage converter and the voltage representing the cooling fan speed were measured.
The voltage signal from the compressor revolution sensor was good, but the voltage signal from the cooling fan converter was not. The reason to that were occasional sensor detection failures and this was taken care of by low-

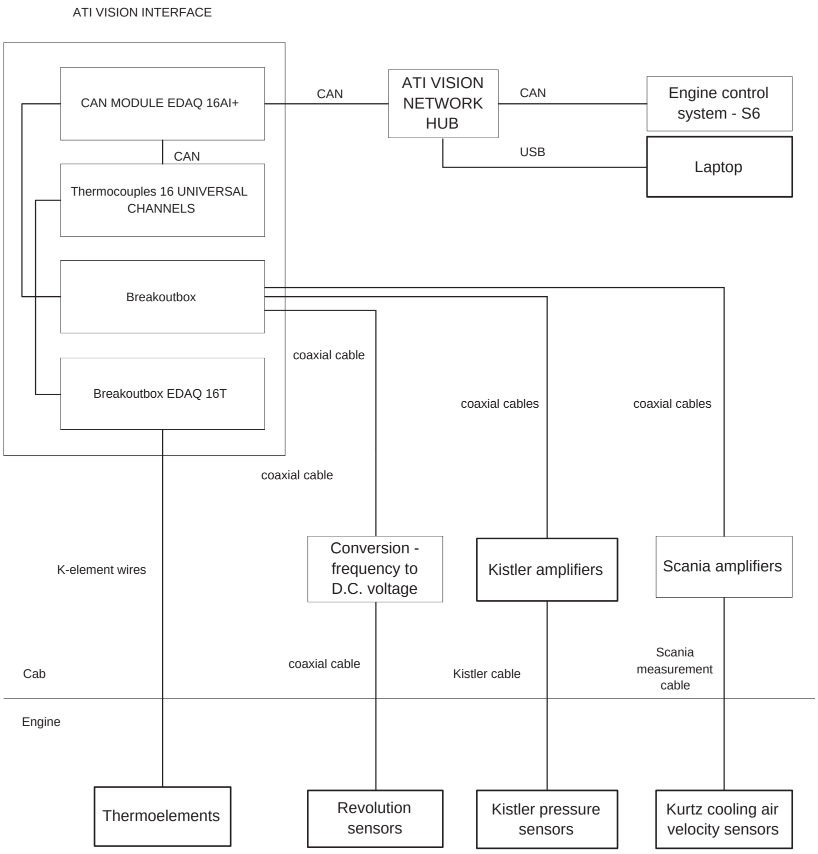


Figure 4.1: The measurement setup.

pass filtering the measured voltage.

4.1.4 Cooling Air-Velocity Sensors

The sensors for measuring the air velocity through the intercooler were mounted on a custom designed construction in front of the intercooler. The sensors are manufactured by Kurtz and have previously been used in a master thesis [6]. The sensors are calibrated for air velocities of $0 - 15m/s$, corresponding an output of $0 - 5V$. To achieve the actual air-velocity, the measured voltages has to be compensated for the present air density as equation 4.1 shows.

$$v_{air,actual} = v_{air,meas} \frac{\rho_{air,cal}}{\rho_{air,actual}} \tag{4.1}$$

where, $\rho_{air,cal}$ is the air density present at the time for the sensor calibration.

A mean value of the sensor signals were taken as the correct velocity of the cooling air. With knowing the cooling air velocity, temperature, pressure, density and the surface area of the intercooler, one can determine the cooling air mass-flow.

Chapter 5

Validation

In this chapter, the validation method is described and results from both the validation of each component and the validation of the total mean value model of the intake system are presented. All models are validated using measured data from different operating conditions. Plots and calculated errors are presented in terms of the mean relative error and the maximal relative error. Possible sources to the errors are also discussed in this chapter.

Types of considered errors:

$$\text{mean relative error} = \frac{1}{n}\sum_{i=1}^{n}\frac{|\hat{x}(t_i)-x(t_i)|}{|x(t_i)|} \tag{5.1}$$

$$\text{mean absolute error} = \frac{1}{n}\sum_{i=1}^{n}|\hat{x}(t_i)-x(t_i)| \tag{5.2}$$

$$\text{maximum relative error} = \max_{1\leq i\leq n}\frac{|\hat{x}(t_i)-x(t_i)|}{|x(t_i)|} \tag{5.3}$$

$$\text{maximum absolute error} = \max_{1\leq i\leq n}|\hat{x}(t_i)-x(t_i)| \tag{5.4}$$

where n is the number of samples, $x(t_i)$ and $\hat{x}(t_i)$ are measured data and modeled data, respectively.
Each model is validated on three different data sets. The data sets are onboard measured dynamic data under different driving conditions.
Data sets used for validation:

- Mixed driving 1
- Mixed driving 2
- Heavy driving
- Fan speed test

Except for the ”fan speed test” validation data set, all data sets for the validation are separated from the data sets used for determining parameters in the models for each component. Four static points of data from the ”fan speed test” validation data set are used in the mapped data set for determining the parameters in the cooling air flow model. The two mixed driving data sets are measured in areas around Södertälje under normal driving conditions with various speeds on the truck and cooling fan. The weather condition was perfect at the time with no wind in any direction. The third heavy driving data set was measured under heavy on road conditions uphill close to Norrköping. The weather condition was not as perfect with shifting wind. The fourth data set ”fan speed test” was meant for determining the parameters in the cooling air mass-flow models, but have also been used for validation. The ”fan speed data set was a measured when the truck where standing still but at different speeds of the cooling fan.

5.1 Component Validation

in this section, the models of each component are validated separately with measured input signals. The different models of the intercooler heat exchanger and the cooling air mass-flow are validated and evaluated to determine which models are to be used in the total mean value model of the intake system.

5.1.1 Compressor outlet temperature model

The model of the compressor outlet temperature have been validated on the following three validation data sets: mixed driving 1, mixed driving 2 and heavy driving. Since the compressor compress the air very fast, the compressor outlet temperature from the model increases faster than the thermoelement measuring this quantity. This is due to the time constant of the thermoelement earlier described in 4.1.2 causing big errors between measured and modeled data in transients. Therefore, the model has been implemented with two outputs for the compressor outlet temperature, one with sensor dynamics included and one without sensor dynamics. In Table 5.1 the errors for both outputs are presented.
It can be seen that the output with sensor dynamics included represents the measured signal much better than the other output. Validation plots of the compressor outlet temperature model with sensor dynamics included can be seen in Figure 5.1. Validation plots of the compressor outlet temperature without sensor dynamics can be seen in Appendix. What one can see in Figure 5.1 is that the error is much more noticeable for low boost pressures. A reason to this might be conductive heat transfer from the hot turbine part of turbocharger to the compressor housing and convection heat transfer from the compressor housing to the air passing through the compressor. This phenom-

Table 5.1: Compressor outlet temperature validation

Compressor outlet	No sensor dynamics		Sensor dynamics	
Rel. error (%)	mean	max	mean	max
Mixed driving 1	3.38	11.60	2.74	7.07
Mixed driving 2	2.59	11.31	2.19	6.76
Heavy driving	3.02	15.04	2.32	7.93
Abs. error (K)	mean	max	mean	max
Mixed driving 1	11.21	42.05	8.97	23.15
Mixed driving 2	8.51	40.14	7.11	21.75
Heavy driving	10.96	59.93	8.36	27.00

ena has been described in [8]. Further, the heat transfer theory is supported by looking at the convection heat transfer equation 5.5.

$$q = W_{comp} c_p (T_{comp,out} - T_{comp,in}) = hA(T_{comp,housing,avg} - T_{comp,avg}) \tag{5.5}$$

where q is the energy per unit time transferred to the air passing through the compressor, W_{comp} is the air mass-flow through the compressor, $T_{comp,out}$ is the compressor outlet temperature, $T_{comp,in}$ is the compressor inlet temperature, $T_{comp,housing,avg}$ is the compressor housing temperature, $T_{comp,avg}$ is the average air temperature inside the compressor, A is the area of the flow channel in contact with the air and h is the heat transfer coefficient.
At low compression, q remains almost constant, the flow W_{comp} decreases and $T_{comp,in}$ remains constant, $T_{comp,out}$ must increase.

According to [8], another explantation to this phenomena might be the fact that the compressor efficiency and flow maps are not that accurate at these operating conditions with low angular velocity on the compressor shaft and low pressure ratio over the compressor. A more thoroughly research in this matter has not been done. The validation results can for worst case scenario be summarized with a maximum relative error of 7.93% corresponding to $27K$ and a mean relative error of 2.73% corresponding to $8.97K$. These results are not perfect, but they are good enough for this application since the efficiency of the intercooler is very high.

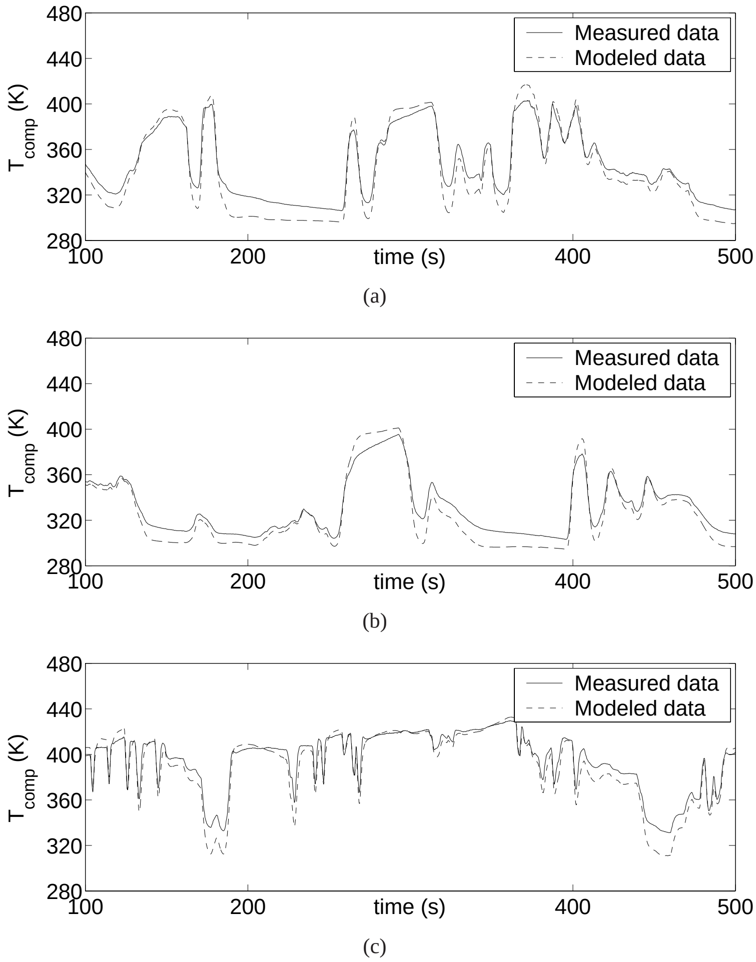


Figure 5.1: Plots of the compressor outlet temperature at the different driving conditions with sensor dynamics included, where 5.1(a) represents mixed driving 1, 5.1(b) represents mixed driving 2 and 5.1(c) represents heavy driving

5.1.2 Intercooler heat exchanger model

The three different models of the intercooler heat-exchanger have been validated on three different data sets. The validation data sets used are: mixed driving 1, mixed driving 2, heavy driving and the errors for the different heat exchanger models are presented in Table 5.2.

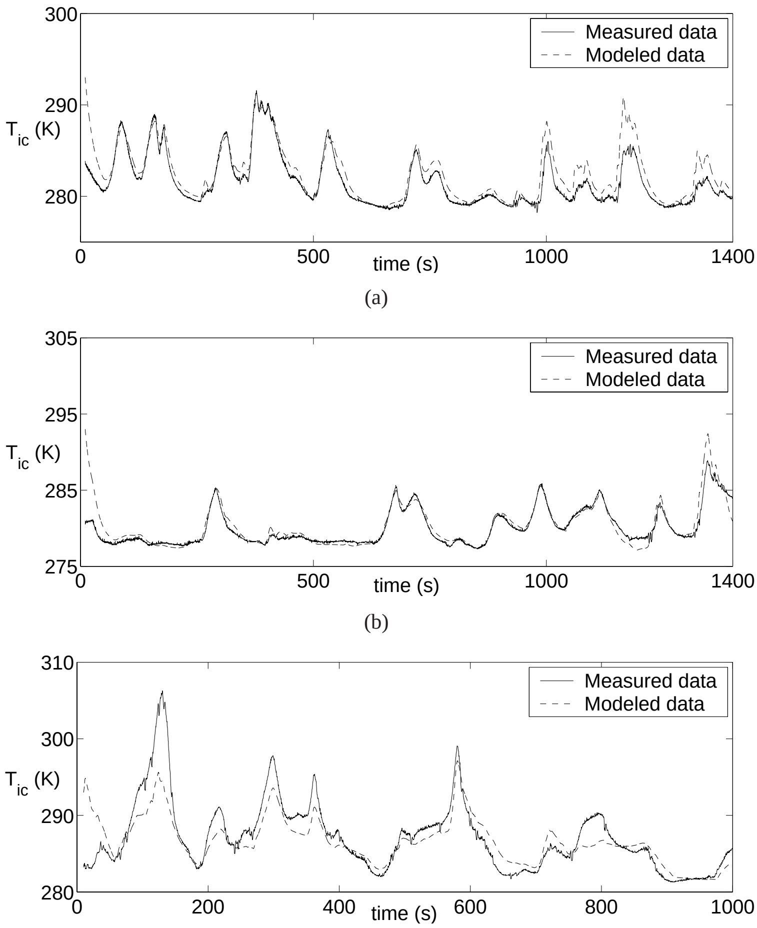
Table 5.2: Intercooler Model Validation

Intercooler outlet	NTU		Lin.Reg.		Map Data	
Rel. error (%)	mean	max	mean	max	mean	max
Mixed driving 1	0.23	1.46	0.34	2.15	0.51	4.79
Mixed driving 2	0.33	1.88	0.18	1.55	0.44	4.69
Heavy driving	0.65	5.26	0.56	3.94	0.71	2.69
Abs. error (K)	mean	max	mean	max	mean	max
Mixed driving 1	0.66	4.26	0.96	6.12	1.46	13.66
Mixed driving 2	0.94	5.35	0.52	4.33	1.25	13.55
Heavy driving	1.85	16.11	1.59	12.01	2.04	8.20

In Table 5.2 one can see that for mixed driving conditions, the NTU model and the linear regression model performs well, while the mapped data model is not as good. For heavy driving conditions the mapped data model has a smaller maximum error than the other two models, but the mean error is still better for the NTU and linear regression model. Overall, the linear regression model performs slightly better than the NTU model and a lot better than the map data model and therefore will be used in the MVM model of the intake system. Figure 5.2 illustrate validation plots of the modeled and measured data for the suggested linear regression model in the different driving conditions. Validation plots of the NTU model and the map data model can be seen in Appendix. In these validation plots the settling time of the modeled signal introduced by the low-pass filter described in 3.3.2 can be seen. The error calculation is performed after this settling time, between 100s and 1400s.

What one can see in Figure 5.2 is that for heavy driving, when the compressor outlet temperature is constantly high for a long time, the intercooler efficiency seems to be lower than any of the heat exchanger models predicts. An explanation might be that for a constant high intercooler inlet temperature, the tubes inside the intercooler are heated from the inside by the warm air from the compressor and that the cooling air can not cool it of in the same rate as it gets heated. The result of this phenomena would be that the temperature of the tubes are a lot higher than the ideal air temperature.
Another explanation to the lowered intercooler efficiency is the so called ”recirculation phenomena”, which means that the cooling air that has been heated up when passing through the intercooler recirculates and passes through the intercooler again but with a higher temperature than the ambient tempera-

ture. The validation results for the linear regression model can be summarized for worst case scenario with a maximum relative error of 3.94% corresponding to $12.01K$ and a mean relative error of 0.56% corresponding to $1.59K$.



(a)

(b)

(c)

Figure 5.2: Plots of the suggested linear regression model for the intercooler heat exchanger at the different driving conditions, where 5.2(a) represents mixed driving 1, 5.2(b) represents mixed driving 2 and 5.2(c) represents heavy driving. The settling time of the modeled signal is introduced by the first order low-pass filter of the intercooler outlet temperature

5.1.3 Cooling Air Mass-Flow Model

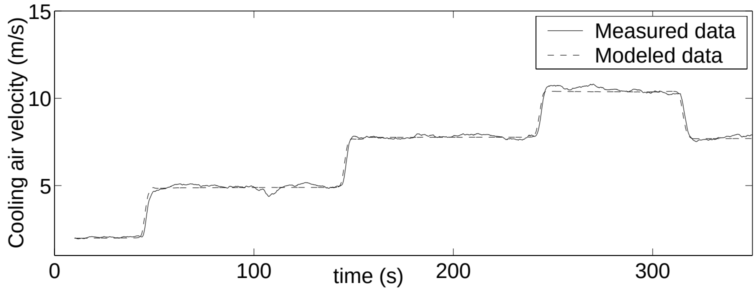
The four different models of the cooling air velocity have been validated on the following three validation data sets: fan-speed test, mixed driving 2 and heavy driving. The results are presented in Tables 5.3 and 5.4, where the cooling air velocity models 3 and 4 corresponds to the cooling air velocity models 1 and 2 but without parameter a_0. What on can see in these Tables by looking at the absolute error is that all models perform quite similar. The suggested model for the MVM model of the intake system is the cooling air velocity model 2, with parameter a_0 included. Validation plots of the suggested model can be seen in Figure 5.3, while the validation plots of the other cooling air velocity models can be seen in Appendix. What one can see in Figure 5.3 is that the suggested model performs great for the fan-speed test and mixed driving 2 validation data sets, while it for the heavy driving validation data set is far from good. A reasonable explanation to the big error for the heavy driving validation data set is that the weather condition at the time was not the best and the air velocity sensors was most likely disturbed by the shifting wind. A more thoroughly research of what reason causing this error would be preferable. The large relative maximum errors occur at low air velocities were small absolute errors corresponds to large relative errors and should be neglected.

Table 5.3: Cooling Air Mass-Flow Model Validation

Modeltype	Model 1		Model 2	
Rel. error (%)	mean	max	mean	max
Fan Speed Test	6.331	43.74	2.18	28.81
Mixed driving 2	15.71	106.08	8.04	35.29
Heavy driving	30.88	162.3	28.98	94.57
Abs. error (m/s)	mean	max	mean	max
Fan Speed Test	0.37	1.06	0.13	0.76
Mixed driving 2	0.54	1.56	0.32	1.40
Heavy driving	2.26	7.73	2.27	7.49

Table 5.4: Cooling Air Mass-Flow Model Validation

Modeltype	Model 3		Model 4	
Rel. error (%)	mean	max	mean	max
Fan Speed Test	6.52	47.86	6.47	51.07
Mixed driving 2	16.02	119.01	10.84	125.57
Heavy driving	30.74	186.37	31.21	249.05
Abs. error (m/s)	mean	max	mean	max
Fan Speed Test	0.34	1.15	0.29	1.24
Mixed driving 2	0.55	1.55	0.35	1.34
Heavy driving	2.23	7.70	2.30	7.46



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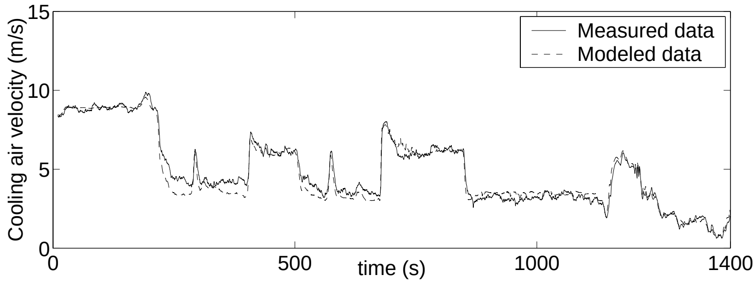

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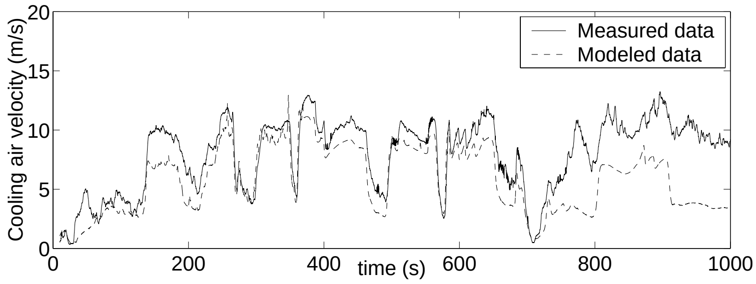

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Figure 5.3: Plots of the suggested cooling air velocity model 2 at the different driving conditions, where 5.3(a) represents fan speed test, 5.3(b) represents mixed driving 2 and 5.3(c) represents heavy driving

5.2 Total Mean Value Model Validation

In this section, the total mean value model of the intake system seen in Figure 3.1 is validated on the same following validation data sets: mixed driving 1, mixed driving 2 and heavy driving. Inputs to the mean value model of the intake system are: ambient temperature, ambient pressure, truck velocity, speed of the cooling fan, turbine speed and engine speed. Validated quantities are the states of temperature and pressure of the mean value model of the intake system. Next follows the validation results for validated quantities.

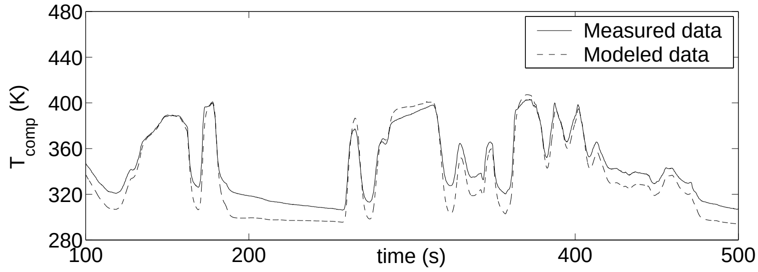
5.2.1 Compressor

Both, the modeled temperature with sensor dynamics included and the pressure after the compressor have been validated. In Table 5.5 the errors for the compressor outlet temperature and pressure after the compressor are presented. Validation plots of the compressor outlet temperature and the pressure after the compressor can be seen Figure 5.4 and 5.5 respectively.

By comparing the errors of the modeled compressor outlet temperature from the total mean value model validation of the intake system with the modeled compressor outlet temperature for the component validation, one can see that the error is slightly bigger for the total mean value model. The reason to this is that for the component validation, measured values of the pressure after the compressor where used and for the total mean value model, modeled pressures after the compressor where used.
The errors of the modeled pressure after the compressor is very small, with a worst mean relative error of 0.67% for the mixed driving 2 validation data set and a worst maximum relative error of 6.97% for the heavy driving validation data set. These errors corresponds to $4.51kPa$ and $11.27kPa$ respectively.

Table 5.5: MVM Compressor Validation

MVEM Compressor	Temperature		Pressure	
Relative error	mean(%)	max(%)	mean(%)	max(%)
Mixed driving 1	3.04	7.46	0.51	4.19
Mixed driving 2	3.34	8.00	0.67	4.51
Heavy driving	2.69	8.51	0.60	6.97
Absolute error	mean(K)	max(K)	mean(kpa)	max(kpa)
Mixed driving 1	9.83	25.58	0.87	9.67
Mixed driving 2	10.66	24.71	1.11	11.40
Heavy driving	9.80	29.26	1.44	11.27



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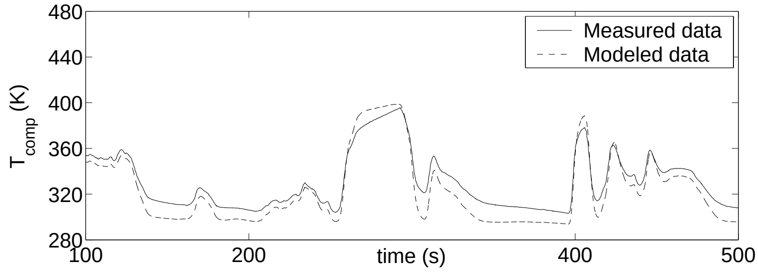

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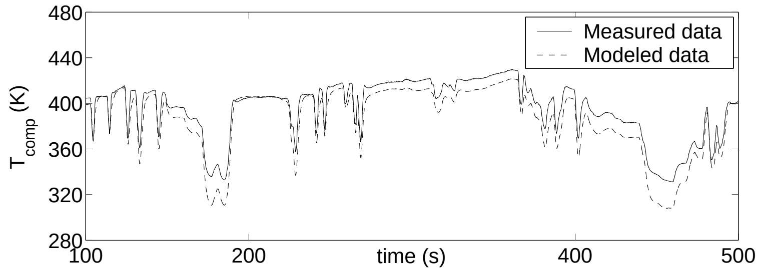

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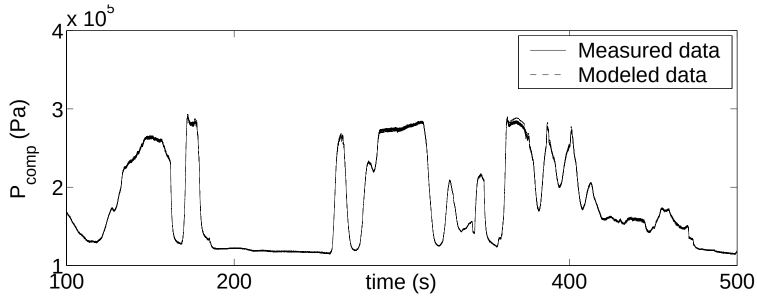
Figure 5.4: Plots of the compressor outlet temperature with sensor dynamics included at the different driving conditions, where 5.4(a) represents mixed driving 1, 5.4(b) represents mixed driving 2 and 5.4(c) represents heavy driving



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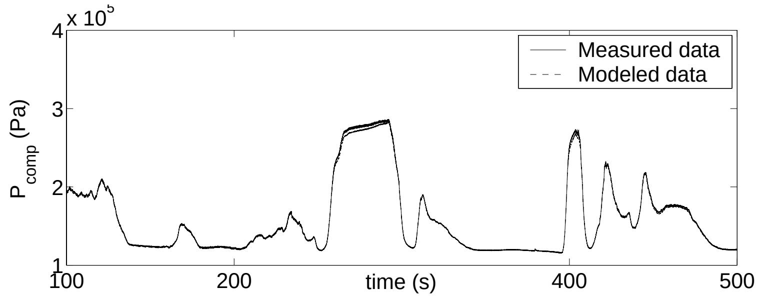

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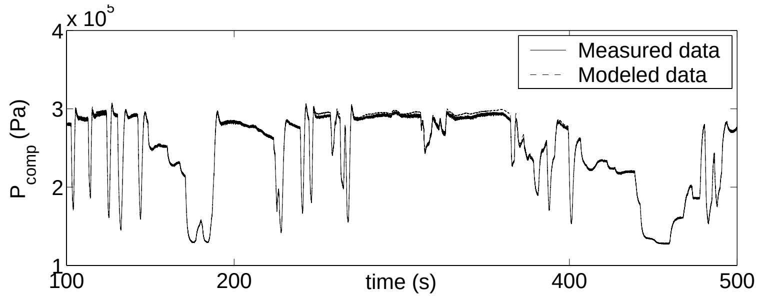

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Figure 5.5: Plots of the pressure after the compressor at the different driving conditions, where 5.5(a) represents mixed driving 1, 5.5(b) represents mixed driving 2 and 5.5(c) represents heavy driving

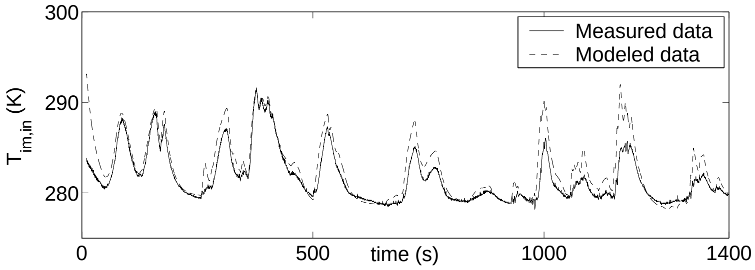
5.2.2 Intercooler

In this section, the modeled intercooler outlet temperature and the modeled pressure after the intercooler have been validated. The errors for the intercooler outlet temperature and pressure after the intercooler are presented In Table 5.6 and validation plots can be seen Figure 5.6 and 5.7 respectively. The settling time of the modeled signal of the intercooler outlet temperature can be seen here as well. The error calculation is performed after this settling time, between $100s$ and $1400s$.

Table 5.6: MVM Intercooler Validation

MVM Intercooler	Temperature		Pressure	
Relative error	mean(%)	max(%)	mean(%)	max(%)
Mixed driving 1	0.37	2.57	1.07	8.57
Mixed driving 2	0.40	2.38	1.10	7.46
Heavy driving	0.68	3.03	1.11	13.52
Absolute error	mean(K)	max(K)	mean(kpa)	max(kpa)
Mixed driving 1	1.04	7.35	1.65	18.35
Mixed driving 2	1.14	6.82	1.64	17.68
Heavy driving	1.97	9.22	2.31	17.14

By comparing the errors of the modeled intercooler outlet temperature from the total mean value model of the intake system with the modeled intercooler outlet temperature for the component validation, one can see that the errors, both relative and absolute errors are slightly larger for the total mean value model, except for the absolute maximum error at heavy driving, which is slightly smaller. The reason to these differences is that for the component validation, the measured temperature after the compressor was used and for the total mean value model, the modeled temperature after the compressor was used.



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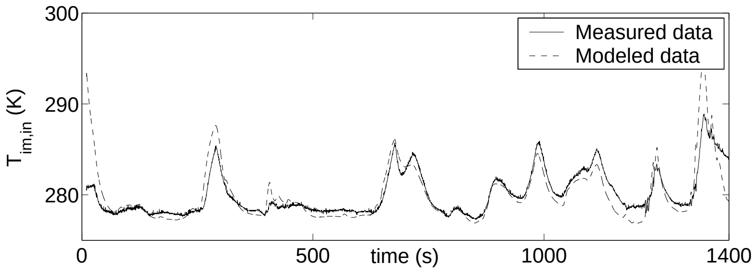

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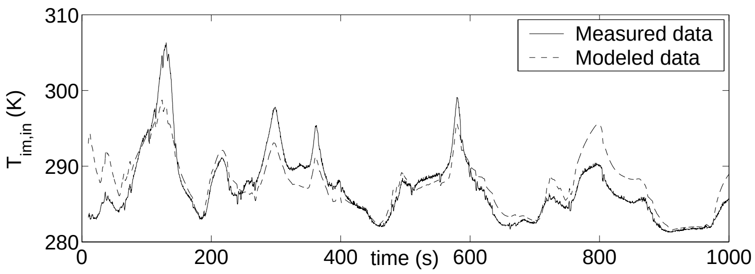

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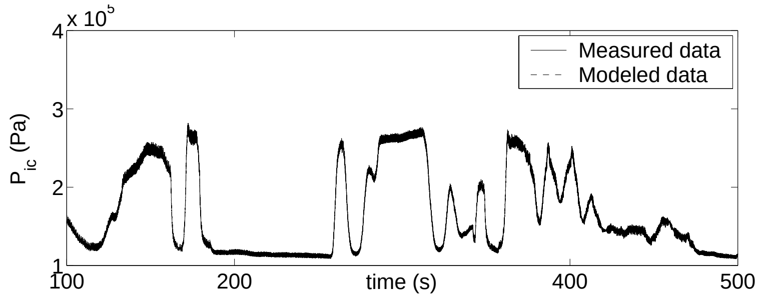
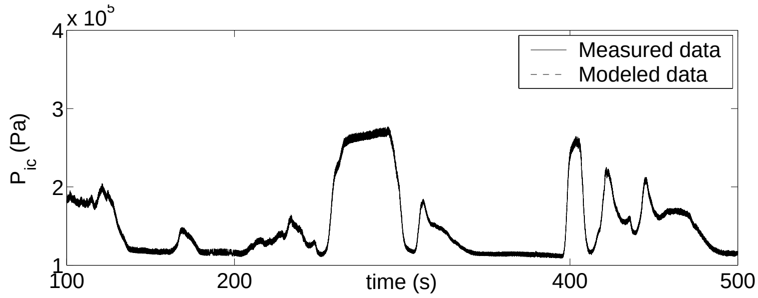
Figure 5.6: Plots of the intake manifold inlet temperature at the different driving conditions, where 5.6(a) represents mixed driving 1, 5.6(b) represents mixed driving 2 and 5.6(c) represents heavy driving. The settling time of the modeled signal is introduced by the first order low-pass filter of the intercooler outlet temperature



(a)



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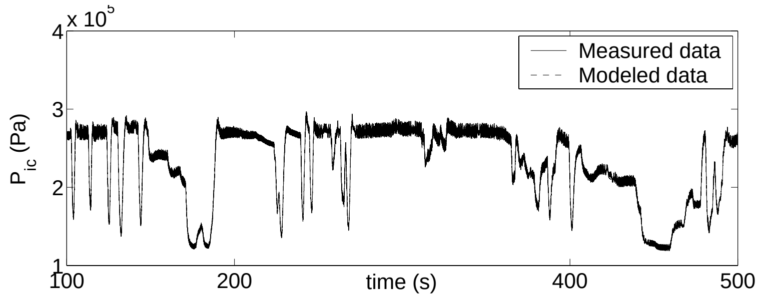

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Figure 5.7: Plots of the Intake manifold pressure at the different driving conditions, where 5.7(a) represents mixed driving 1, 5.7(b) represents mixed driving 2 and 5.7(c) represents heavy driving

5.2.3 Intake Manifold

In this section, the modeled intercooler outlet temperature have been validated on the measured temperature inside the intake manifold. The thermocouple is placed at the exact same position as the production sensor measuring the intake air temperature is located. The errors are presented in Table 5.7 and validation plots can be seen Figure 5.8, were the settling time of the modeled signal of the intercooler outlet temperature can be seen here as well. The error calculation is performed after this settling time, between 100s and 1400s.

Table 5.7: MVM Intake manifold temperature validation

MVM Intercooler	Intake manifold temperature	
Relative error	mean(%)	max(%)
Mixed driving 1	0.49.	1.23
Mixed driving 2	0.52	1.15
Heavy driving	0.86	3.31
Absolute error	mean(K)	max(K)
Mixed driving 1	1.40	3.48
Mixed driving 2	1.46	3.25
Heavy driving	2.51	10.12

By comparing the results with the results for the intercooler outlet temperature validation, one can see that the errors are slightly larger for the intake manifold temperature validation. And by looking at the validation plots, one can easily see that there is a constant error between the measured and modeled signals. Another phenomenon that differs from the intercooler outlet temperature validation plots, is that the measured signal inside the intake manifold fluctuates a lot more.

Reasons to these observations are most likely non modeled heat transfer effects from the solid intake manifold causing the constant error and so called pumping effects from the engine causing the fluctuating signal. Pumping effects are a phenomena that occur inside the intake manifold and means that the engine is pumping hot gases from the cylinders back and fourth. These pumping effects might also contribute to the constant error observed.

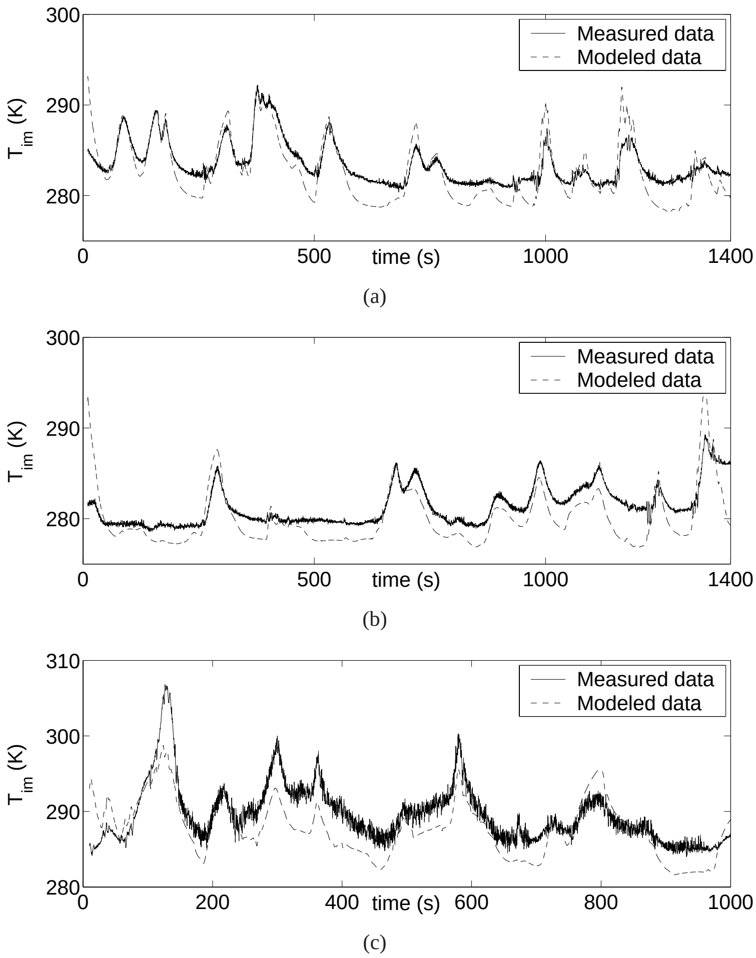

Figure 5.8: Plots of the intake manifold temperature validation at the different driving conditions, where 5.8(a) represents mixed driving 1, 5.8(b) represents mixed driving 2 and 5.8(c) represents heavy driving. The settling time of the modeled signal is introduced by the first order low-pass filter of the intercooler outlet temperature

Chapter 6

Results and Future Work

6.1 Results

A mean value model of the intake system have been developed consisting of 4 states for temperature and pressure after compressor and intercooler. Inputs to the model are: ambient temperature, ambient pressure, truck velocity, engine speed, cooling fan speed and turbine speed. The model predicts the intake manifold temperature with a maximum absolute error of $10.12K$ and a maximum mean error of $2.51K$. The results are only valid for the specific test vehicle and configuration.

With a suitable placement of the production boost temperature sensor and a model of its sensor dynamics, it should be possible this model for diagnoses. Today, desktop computers have to be used for simulations, but with minor changes it should be possible to run the model in the Engine Control Unit (ECU). If this model is to be used in a diagnose application, conditions for when the diagnose test is allowed to be performed should be introduced. The reason for these conditions is to make sure that the diagnose test is performed under suitable operating conditions when the model performance is robust and accurate. An example of these rules is that the truck velocity should be higher than 40 kph. At low truck velocities, the cooling air mass-flow model becomes more sensitive to uncontrollable objects in front of the truck, such as after market headlamps, leafs and snow.

6.2 Future Work

Since the mean value model of the intake system is only validated on the specific test vehicle, measurements on a similar vehicle with the exact same configuration needs to be carried out to be able to validate the model properly. To be able to use this model in the future it needs to be continuously improved to be valid when additional components are added to the engine. In the future, vehicles are most likely equipped with both Exhaust Gas Recirculation (EGR) and Variable Geometry Turbo (VGT), which the model needs to be updated with. Further, a method for building a library of configuration specific parameters used in the model needs to be developed. Examples of configuration specific parameters are: restriction coefficients, Volumes, Intercooler heat exchanger parameters and Cooling air velocity parameters. This library will make it possible to use the model for different vehicle configurations.

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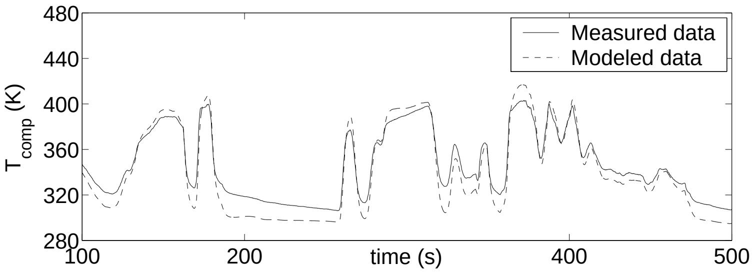
Appendix A

Component validation plots



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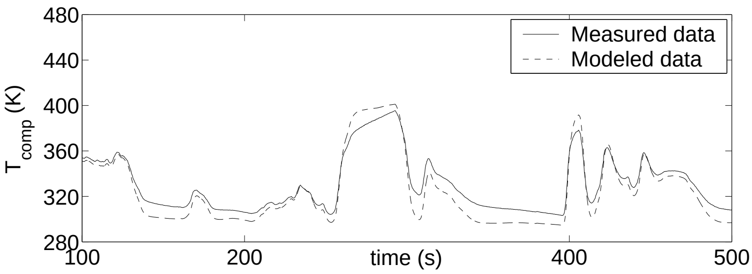

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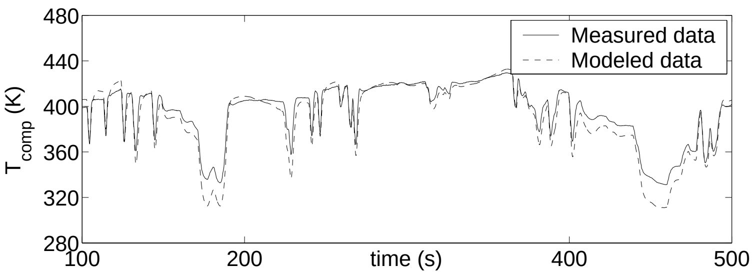
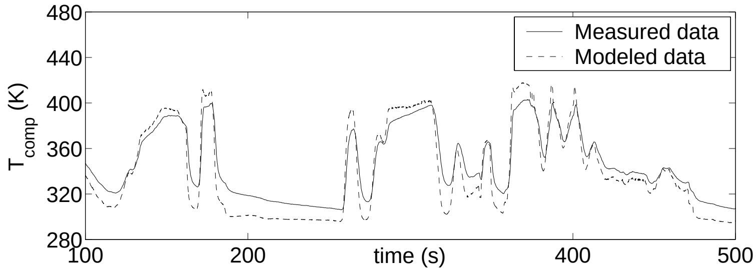


(c)

Figure A.1: Plots of the compressor outlet temperature when sensor dynamics are included at the different driving conditions, where A.1(a) represents mixed driving 1, A.1(b) represents mixed driving 2 and A.1(c) represents heavy driving



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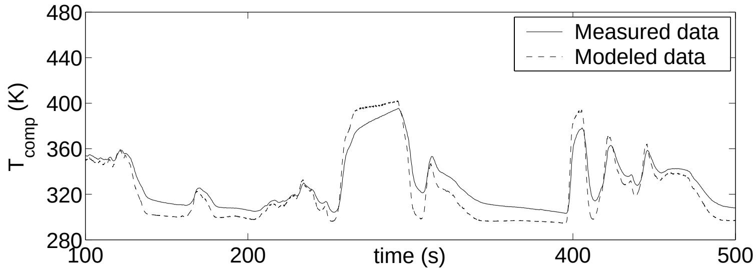

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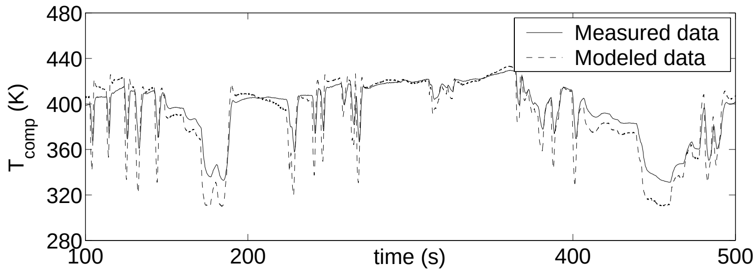
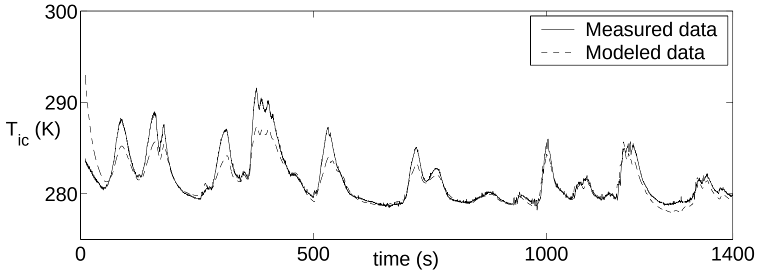
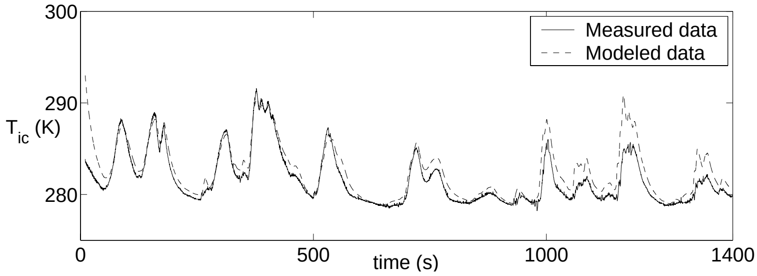


(c)

Figure A.2: Plots of the compressor outlet temperature when sensor dynamics are excluded at the different driving conditions, where A.2(a) represents mixed driving 1, A.2(b) represents mixed driving 2 and A.2(c) represents heavy driving



(a)



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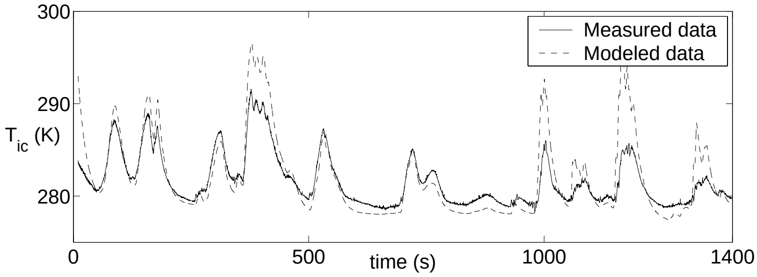


(c)

Figure A.3: Plots of the intercooler outlet temperature for the different heat exchanger models at mixed driving 1, where A.3(a) represents the NTU model, A.3(b) represents the linear regression model and A.3(c) represents heavy driving. The settling time of the modeled signal is introduced by the first order low-pass filter of the intercooler outlet temperature

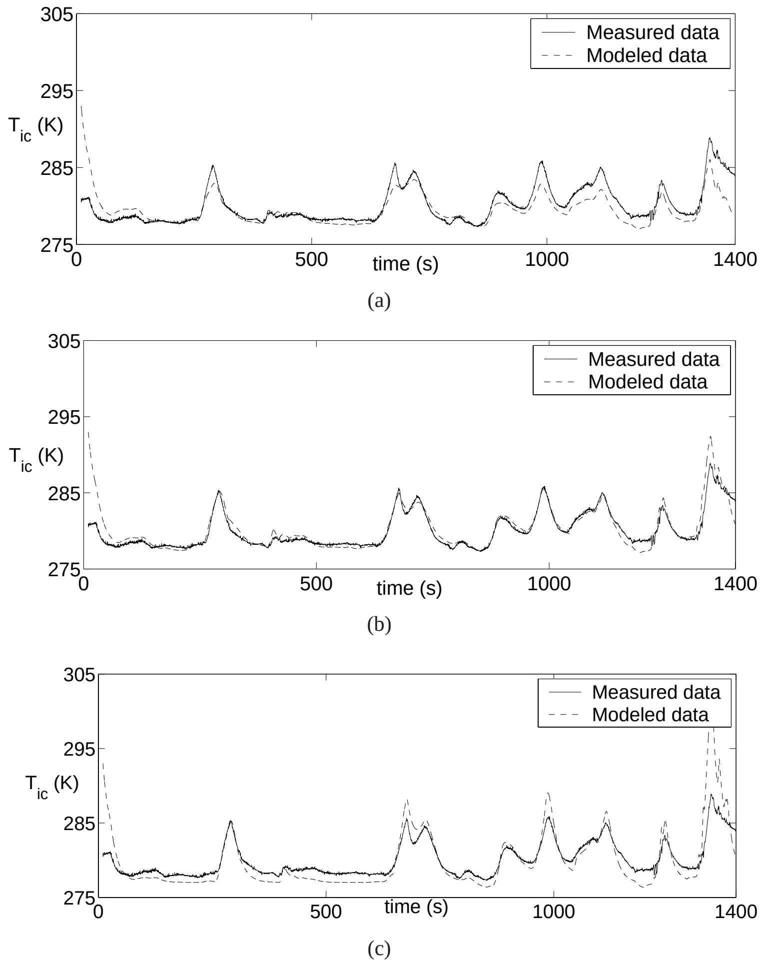
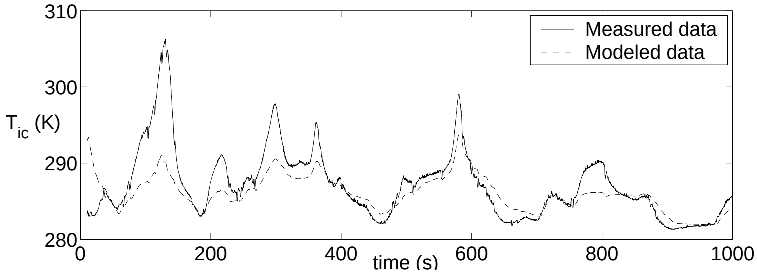


Figure A.4: Plots of the intercooler outlet temperature for the different heat exchanger models at mixed driving 2, where A.4(a) represents the NTU model, A.4(b) represents the linear regression model and A.4(c) represents heavy driving. The settling time of the modeled signal is introduced by the first order low-pass filter of the intercooler outlet temperature



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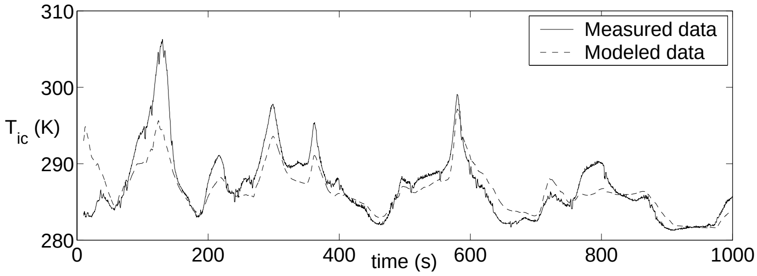

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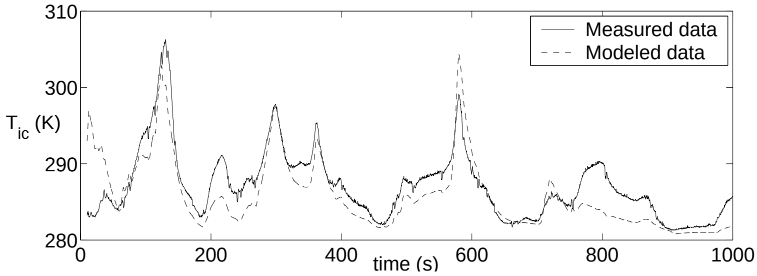
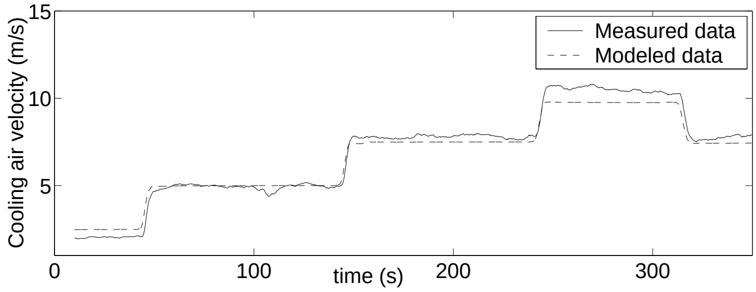


(c)

Figure A.5: Plots of the intercooler outlet temperature for the different heat exchanger models at heavy driving, where A.5(a) represents the NTU model, A.5(b) represents the linear regression model and A.5(c) represents heavy driving. The settling time of the modeled signal is introduced by the first order low-pass filter of the intercooler outlet temperature



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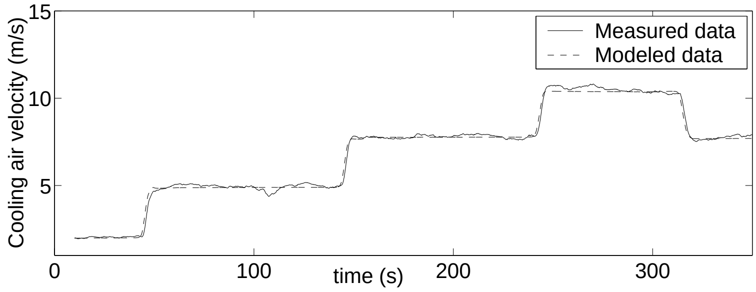

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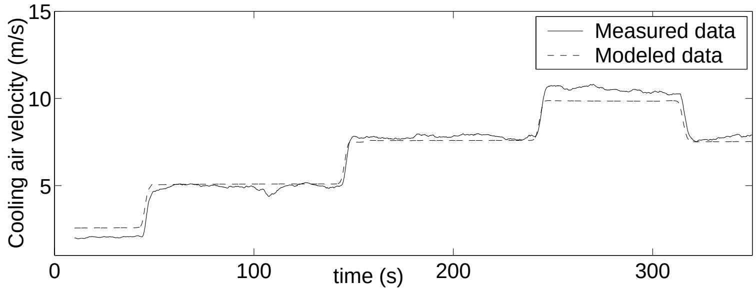

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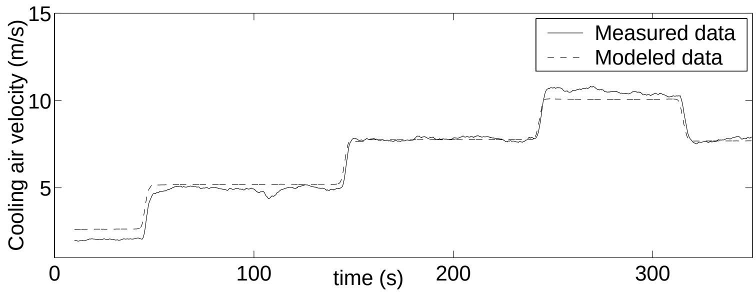
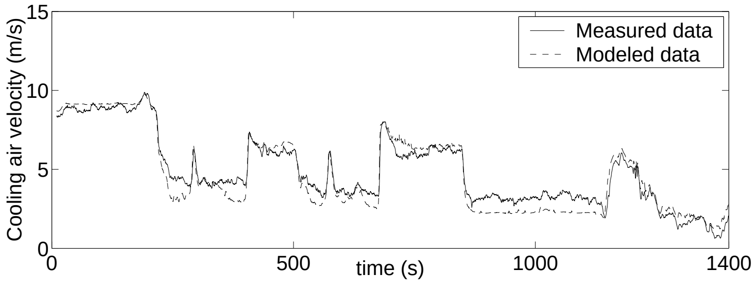


(d)

Figure A.6: Plots of the different cooling air velocity models for the fan speed test , where A.6(a) represents the AirVel model 1, A.6(b) represents the AirVel model 2, A.6(c) represents AirVel model 3 and A.6(d) represents AirVel model 4



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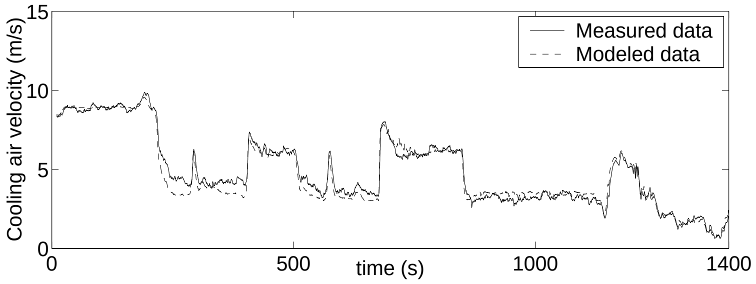

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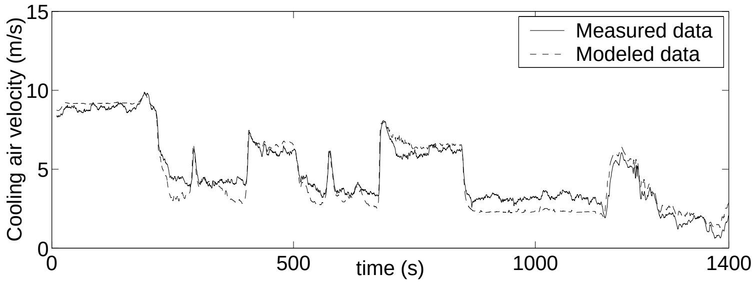

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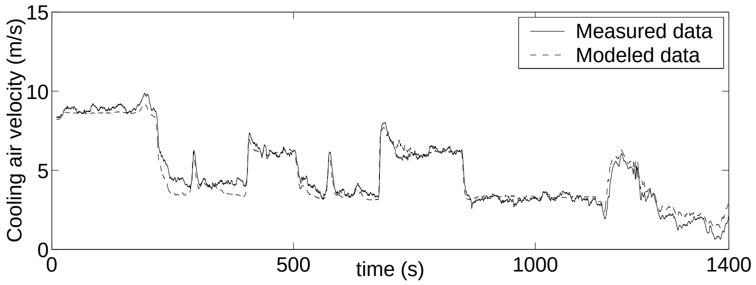
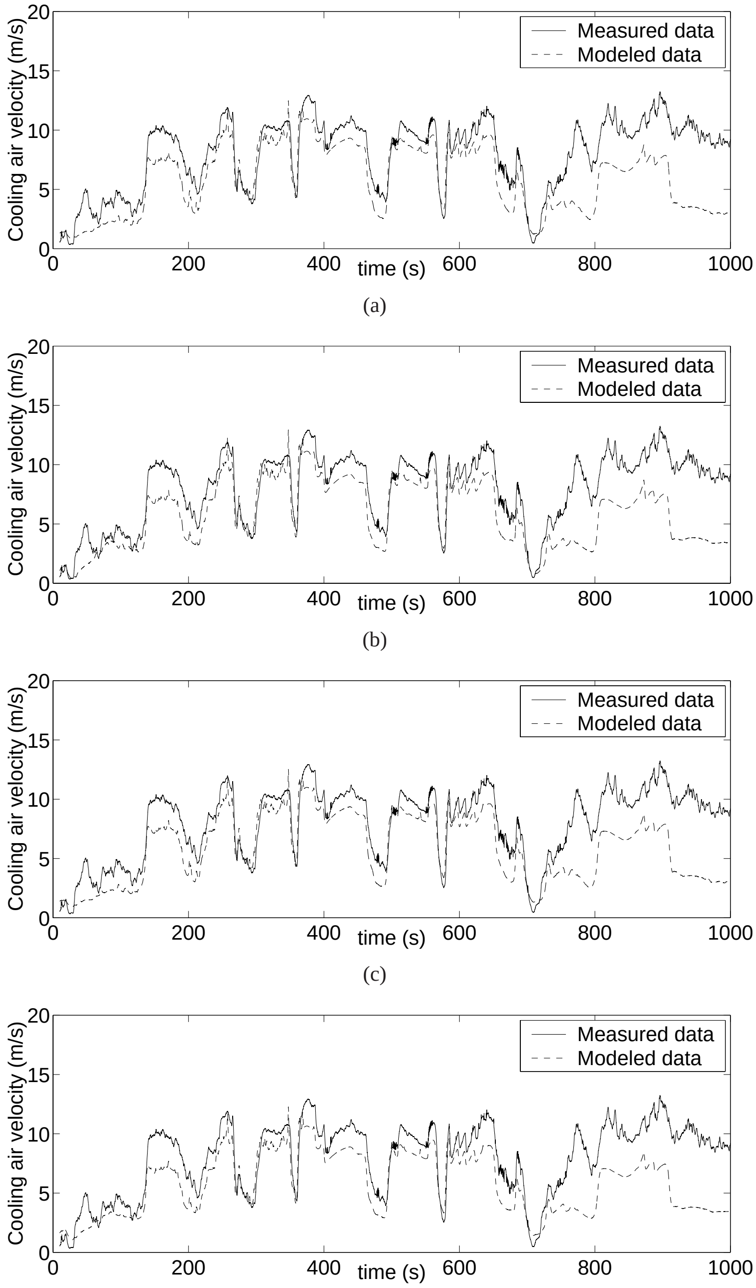


(d)

Figure A.7: Plots of the different cooling air velocity models at mixed driving 2, where A.7(a) represents the AirVel model 1, A.7(b) represents the AirVel model 2, A.7(c) represents AirVel model 3 and A.7(d) represents AirVel model 4



(a)

(b)

(c)

(d)

Figure A.8: Plots of the different cooling air velocity models at heavy driving, where A.8(a) represents the AirVel model 1, A.8(b) represents the AirVel model 2, A.8(c) represents AirVel model 3 and A.8(d) represents AirVel model 4