The art of injecting the correct amount of fuel

Modelling of a gaseous sequential injection System

Ylva Nilsson

Reg nr: LiTH-ISY-EX-2034

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Examensarbetet utfört inom Fordonssystem
vid Tekniska Högskolan i Linköping
av

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Abstract

As my master thesis, I have studied a gaseous sequential injection system (GSI) consisting of a fuel tank, safety shut-off valves, a vaporiser/pressure regulator (V/P), injectors and an electronic control module. The aim of this study was to make a mathematical description of the behaviour of the GSI. The main question has been how to inject the correct amount of fuel. It was important that the developed models could be transformed into real-time running algorithms.

As the GSI injectors were still under development, no study of the injectors has been done. The work has instead been focused on determining the parameters that affect the actual injected amount of fuel, and how to derive them from the parameters measured by the available sensors.

Steady state models describing the pressures and outlet temperature of the vaporiser/pressure regulator, the temperature change of the fuel between the V/P and the injectors, outdoor temperature and tank temperature, have been made. The pressure models have been made by identification, while the models for the outlet fuel temperature of the V/P and the change in temperature before the injectors are based on physical knowledge. The models are all describing the measured data well.

Transient models have been made for the outlet temperature of the V/P and the change in fuel temperature between the V/P and the injectors. When making these models, emphasis was put on making the algorithms easy to calculate in the electronic control module. The transient behaviour has thus been approximated with first order filters and offsets, which gave rather good results.

The developed models can be transformed into real-time running algorithms.

There are some things left to be described, but it seems to be possible to make a real-time running software which calculates how to inject the correct amount of fuel from the information given by the available sensors.

Contents

1 Introduction

1.1 Background

In some countries liquefied petroleum gas (LPG) is readily available and is a relatively cheap fuel, which creates a wish to have cars running on LPG. Since it can be very hard to find LPG in other places and since the engine does not run properly on LPG in some cases, people like their cars to run on petrol too. The department of combustion engines at TNO has developed a LPG injection system, which together with an ordinary petrol engine makes a petrol/LPG dual fuel engine system. This system has been in commercial use for many years already.

With higher environmental requirements and more demanding customers, there has emerged a need for a completely new injection system. One of the planned improvements is to use models to calculate how to inject the correct amount of fuel, i.e. the amount of fuel that gives a stoichiometric mixture (λ=1).

1.2 The thesis

This thesis is a study of the new injection system.

The main goal of this master thesis was to know how to inject a desired amount of fuel. The injection is controlled by shutting on and off a current through the injectors (*Figure 1*). Besides the injection time, the actual injected amount is depending on a lot of different parameters, for example pressure, temperature and engine speed.

As the injectors are still under development, no study of the injectors has been included in this thesis. Instead, the work has been concentrated on determining what parameters affect the actual injected amount of fuel, and how to derive them from the parameters measured by the available sensors. Implementation aspects have been considered, because the developed algorithm is meant to be used in a normal car. As a consequence, the calculations involved in the algorithm should be kept simple.

The study has been limited to include only normal operation behaviour of the system. That means that the system is assumed to be fully functional and that the fuel becomes completely vaporised inside the vaporiser/pressure regulator.

The aim of this study is to work as basis for further development of the system algorithms.

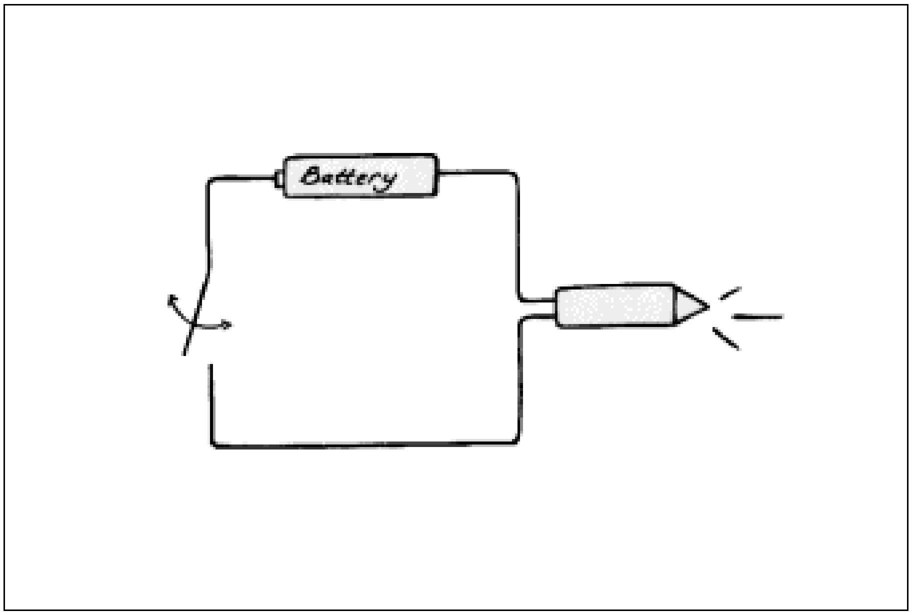

Figure 1: *The injectors are controlled by an electrical current*

1.3 The report

The purpose of this report is to describe and discuss the work I have done. This is done in the following chapters:

Chapter 2: Description of what the studied system looks like and how it works.

Chapter 3: Description of the system from a modelling point of view

Chapter 4-6: Modelling discussions concentrated on particular parts of the system; the connection between the vaporiser/pressure regulator and the injectors, the vaporiser/pressure regulator and the tank. The main effort has been put on temperature models.

Chapter 7: Implementation issues

Chapter 8: An account of things that not have been considered when developing the models but nevertheless might influence the system.

Chapter 9: Conclusions and recommendations

Appendix A includes explanations for some symbols and abbreviations used in this thesis.

2 The system

The studied system consists of a gaseous sequential injection system (GSI) and an engine. There is always at least one extra fuel supply system connected to the engine; a petrol injection system. There might also be some other fuel supply systems. These fuel supply systems do not belong to the studied system.

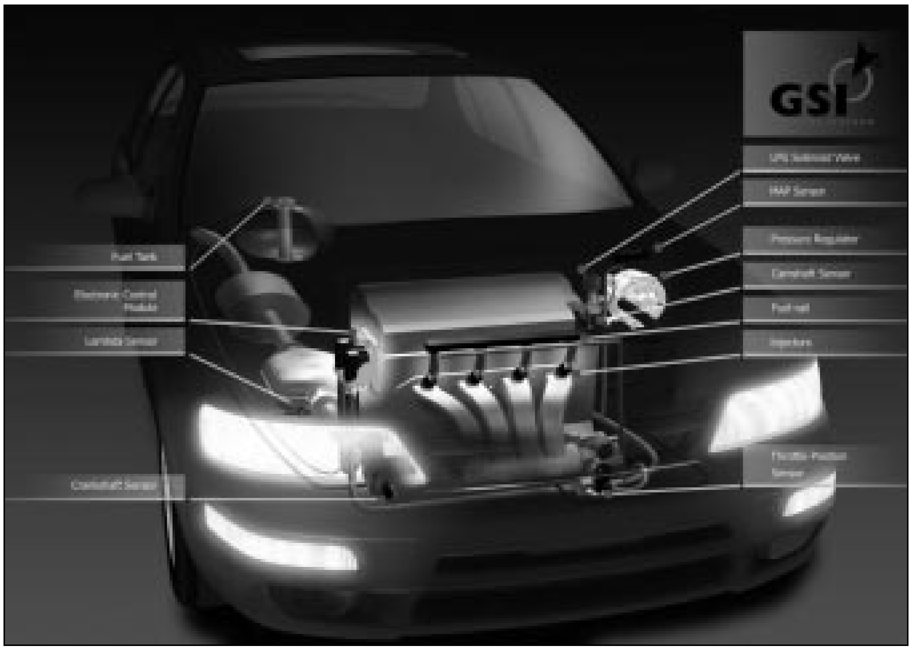

Figure 2: The gaseous sequential injection system

The gaseous injection system (*Figure 2*) consists of the following parts: A fuel tank, some safety shut-off valves (at the tank and vaporiser/pressure regulator), a vaporiser/pressure regulator, injectors and an electrical control module (ECM). The fuel is fed from the tank into the vaporiser/pressure regulator, where it is vaporised and heated. When the fuel is flowing from the vaporiser/pressure regulator to the injectors, the pressure is set by the vaporiser/pressure regulator to a specific value relative to a reference pressure, which most of the time is the same as the inlet manifold pressure (p_{im}). Later, it is injected into the inlet manifold by the injectors. If LPG is not in use, the shut-off valves close to prevent a flow of

LPG into the engine. This is done of safety reasons. The whole injection process is controlled by the ECM.

For this study, the interesting engine parts are the throttle, the inlet manifold, the combustion chambers (cylinders), the exhaust manifold, the catalyst and the cooling system (*Figure 3*). These parts are made for the use of petrol, but they don't have to be modified to make it possible to run the engine on LPG. This report will not include any detailed description or discussion of the engine. This can be found elsewhere. [3]

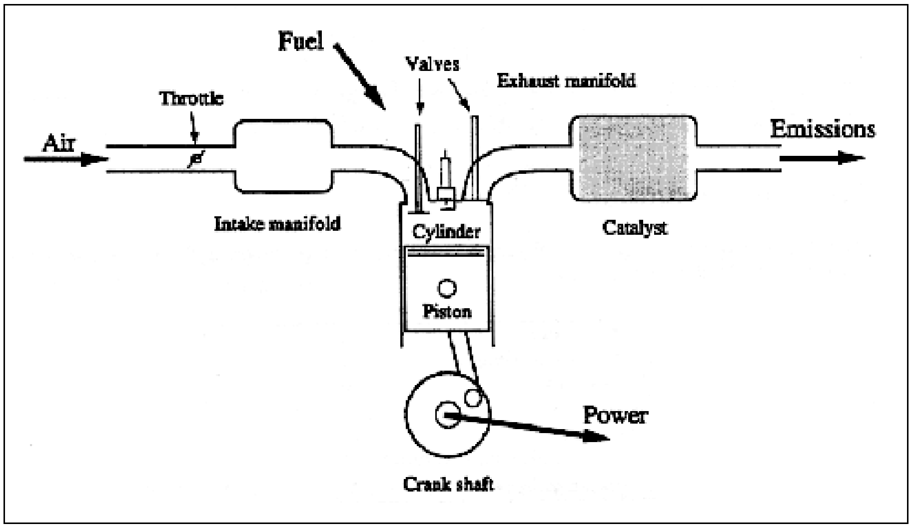

Figure 3: The engine

To make it possible to control the system, different sensors are included in the system. These are e.g. sensors for measuring coolant temperature, battery voltage, manifold air pressure (p_{im}), throttle angle, engine speed and the air/fuel ratio.

2.1 Fuel tank

At room temperature combined with normal sea level pressure, LPG is in a gaseous state. Therefore, a special LPG fuel tank is used, which can stand the vapour pressure at ambient temperature.

When the tank is refilled, the LPG will start to vaporise. Due to this change of state, the pressure will at the same time start to rise. The vaporisation of fuel will continue until the pressure is as high as the boiling pressure. This means that there will always be a layer of gas on top of the liquefied fuel. The fuel taken from the tank is in a liquid state due to the construction of the outlet valve.

2.2 Vaporiser/pressure regulator

It is important for the injector performance that the fuel is fully vaporised when it is injected into the engine cylinders. Therefore, a vaporiser is included in the system. At the moment the fuel is taken from the tank, the fuel is in a liquid state, but on the edge of boiling. When entering the vaporiser/pressure regulator, the pressure decrease causes the fuel to vaporise. To make sure that all fuel will vaporise, and to prevent the vaporiser/pressure regulator to freeze up, the fuel is heated. This is done by letting the coolant flow through the vaporiser/pressure regulator and thereby heat up the metal surrounding the fuel.

To make it possible to inject the fuel into the inlet manifold easily, the pressure in the inlet manifold must be lower than the pressure in the connection between the V/P and the injectors. For the control of the injection process, the difference between these two pressures must be of a certain known value. That is why a pressure regulator is needed. In this system the desired pressure difference is 0.96 bar. Unfortunately, the target is not reached for all combinations of fuel flows and temperatures.

2.2.1 Principal behaviour of the vaporiser/pressure regulator

The vaporiser/pressure regulator consists of two stages. The purpose of the first stage is to vaporise the fuel and to control the pressure, while the aim of the second stage is to regulate the pressure. Schematic pictures of the vaporiser/pressure regulator are shown in *Figure 4* and *Figure 5*.

When the gas pressure in the first stage (a) of the vaporiser/pressure regulator is not significantly higher than the reference pressure (b) (the same as the inlet manifold pressure in steady state conditions), the fuel can flow from the tank into the first stage. This is because the first stage spring (1) is pressing down the siphon (2) that otherwise would cover the first stage inlet valve (3). Due to the fact that the tank pressure is higher than the pressure in the first stage, fuel will flow into the vaporiser/pressure regulator and the pressure will rise. When a certain difference between the first stage pressure and the reference pressure is reached (1.40 bar at nominal conditions) the diaphragm (4) will, together with the spring (5) connected to the siphon, press back the spring on top of the diaphragm and close the inlet valve with the siphon. Inside the first stage chamber, the fuel is heated with energy transferred from the coolant (d) via the walls enclosing the chamber.

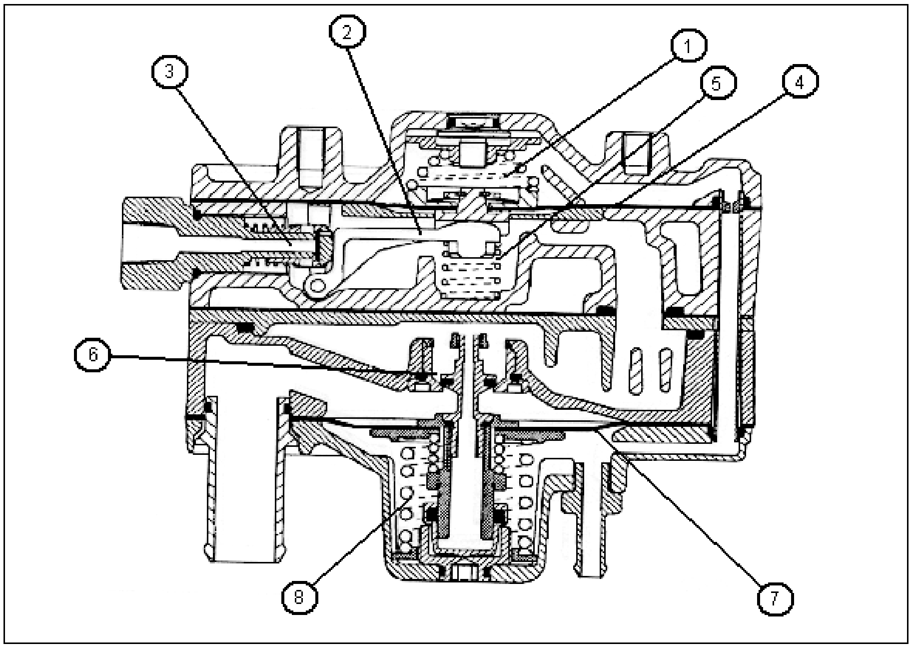

Figure 4: *The vaporiser/pressure regulator*
1. First stage spring
2. Siphon
3. First stage inlet valve
4. First stage diaphragm
5. Spring connected to the siphon
6. Second stage inlet valve
7. Second stage diaphragm
8. Second stage spring

After having passed through the first stage chamber, the fuel reaches the second stage chamber (c) by the open second stage inlet valve (6). The purpose of the second stage is to stabilise the pressure at 0.96 bar overpressure (relative to the reference pressure). When the pressure has reached a level of 0.96 bar overpressure, the force on the second stage diaphragm

(7) will be greater and opposite to the force of the second stage spring (8), thus pressing down on the spring and closing the valve. [4]

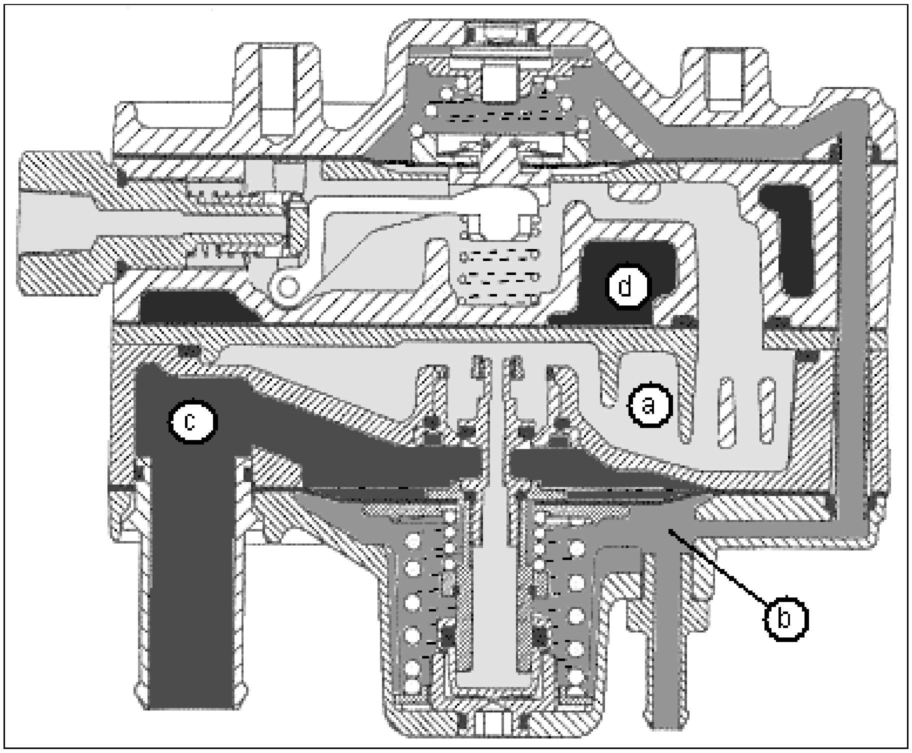

Figure 5: *The vaporiser/pressure regulator*

a: First stage pressure
b: Reference pressure
c: Second stage pressure
d: Coolant

2.3 The connection between the V/P and the injectors

The connection between the vaporiser/pressure regulator and the injectors consists of hoses and a rail. There is a hose going from the regulator to the rail and a hose for each injector going from the rail to the injectors (*Figure 6*). The hoses are made of rubber and their lengths depend on which type of car the gaseous injection system is used in. The rail can be of several different shapes and is made of metal, rubber or plastic.

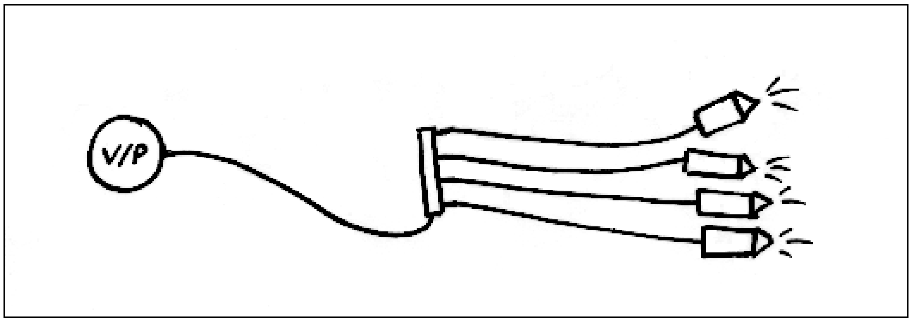

Figure 6: *The connection*

2.4 Injectors

The last part the fuel will pass before entering the manifold is one of the injectors (*Figure 7*). Its task is to inject the fuel into the inlet manifold. It is important to inject the correct amount, not too little and not too much.

Figure 7: *Injectors*

The injector is controlled by a an electrical circuit. When a current flows through the injector, an electromagnetic force is created inside the injector. This will cause the injector to open, and the fuel will flow into the inlet manifold due to the lower pressure there.

2.5 ECM

An electronic control module (ECM) controls the gaseous injection system. It determines what to do and checks how the engine is performing with the help of various sensors. These are sensors for measuring coolant temperature, battery voltage, inlet manifold pressure, throttle angle, engine speed and air/fuel ratio.

The calculation time available each cycle is very limited (about 8ms). Also the storage capacity is limited, but that usually causes minor problems compared to the short time available. Therefore, look-up tables are often used instead of complicated calculations.

2.6 LPG

LPG stands for liquefied petroleum gas. It consists mainly of propane and butane. The ratio between propane and butane changes between different countries and also with the time of year. In the Netherlands, the ratio is often 60/40, and the ratio between i-butane and n-butane is roughly 1/3. About half the LPG is produced in association with production of natural gas and the other half is produced in association with crude oil refining.

The advantages of using LPG are, except for being relatively inexpensive in some countries, that it is better for the environment in some aspects and that it mixes very well with air. Hardly any fuel will condense on the walls of the inlet manifold when LPG is used, because the fuel wants to stay in gaseous form. The disadvantages are that LPG is more space consuming due to the lower energy concentration and the availability is scarce. [5]

Appendix B contains some useful LPG data.

3 The system - Modelling

From a modelling point of view, the system looks as follows (*Figure 8*):

A driver controls the airflow into the engine via a throttle. The ECM gets information about the magnitude of the airflow. It calculates the amount of fuel to inject and determines the injection time (t_{inj}). After that, it creates a current through the injectors during t_{inj} seconds. The injector opens and fuel flows into the engine. Air and fuel mixes and is set on fire. On the way out from the engine the air/fuel ratio of the burned mix is measured. This ratio gives feedback to the ECM, so it knows if the algorithm works properly or if corrections have to be done.

The models below and in the following chapters are all mean value models. This means that the signals, parameters and variables considered are averaged over one or several engine cycles.

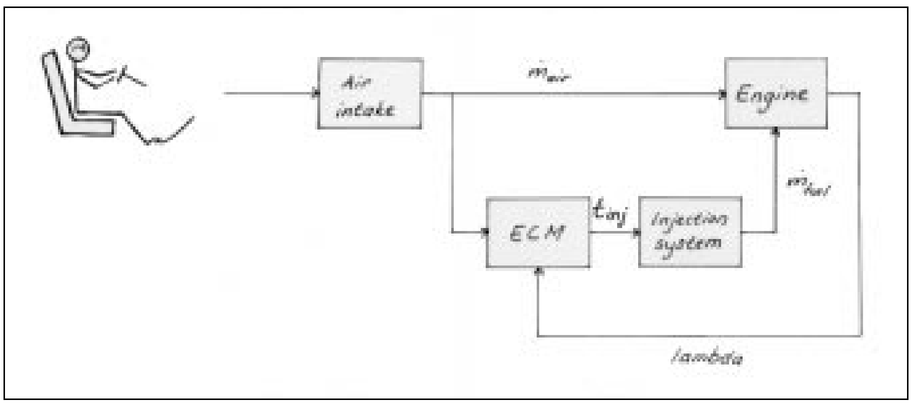

Figure 8: The system

3.1 The air intake

According to [3], the air mass flow into the engine is

$$m_{air}(N, p_{im}, T_{ch}) = \eta_{vol}(N, p_{im})\frac{V_d n_{cyl} p_{im} N}{60 n_r R T_{ch}} \qquad (1)$$

V_d is the displaced volume, n_{cyl} is the number of cylinders, n_r is the number of revolutions per engine cycle ($n_r = 2$ for a four stroke engine), R the gas constant and T_{ch} the charge temperature. η_{vol} is called the volumetric efficiency and has to be measured for each type of engine.

T_{ch} causes problems, since there is no sensor for measuring the charge temperature. One possibility is to relate T_{ch} to the airflow, coolant temperature and ambient temperature. The background for the choice of parameters is that before the air enters the engine, it has ambient temperature. Inside the intake it is heated, because the engine has a temperature about the same as the coolant. How much the air is heated depends on how much time is spent inside the intake manifold.

Table 1: f as function of airflow

Scaling factor for the air flow (linear)	*F*
0	0.47
8	0.47
16	0.47
32	0.566
48	0.664
64	0.715
80	0.754
96	0.8
112	0.84
128	0.86
144	0.88
160	0.9
176	0.918
192	0.938
208	0.945
224	0.957
240	0.977

This is a model for the charge temperature used at TNO:

$$T_{ch} = T_{cool} + f(m_{air}) \cdot (T_{amb} - T_{cool}) \tag{2}$$

$f(m_{air})$ is given by a look-up table. *Table 1* contains the values for an engine with an aluminium manifold and a long intake duct (in the cylinder head).

There is no sensor for measuring the temperature enclosed in the system. The ambient temperature must hence be determined in some other way. One way is to use the information given from the coolant temperature sensor at the moment the engine is turned on. If the car has not been in use for a long time, the coolant temperature is the same as the ambient temperature. The first measured value of the coolant temperature is thus the same as the ambient temperature. Obviously, this is not the case if the engine has been turned of for just a few minutes. The temperature of the coolant changes slowly and will not have reached the ambient temperature.

A model describing the ambient temperature that works well in most cases is

$$T_{amb} = \begin{cases} T_{cool}(0) & \text{if } T_{cool}(0) \le T_{average} \\ T_{average} & \text{if } T_{average} < T_{cool}(0) \le T_{\max} \text{ and } t_{key-off} \le t_{\max} \\ T_{cool}(0) & \text{if } T_{average} < T_{cool}(0) \le T_{\max} \text{ and } t_{key-off} > t_{\max} \\ T_{average} & \text{if } T_{cool} > T_{\max} \end{cases} \tag{3}$$

$T_{cool}(0)$ is the temperature of the coolant at the time the engine is turned on, and $t_{key-off}$ is the amount of time the engine has been turned off. Suitable values for $T_{average}$, T_{max} and t_{max} are 10°C, 35°C and 3-4 hours respectively.

3.2 ECM

The amount of fuel to inject depends on how much air is flowing into the engine. This can be described by

$$m_{fuel} = \frac{m_{air}}{\lambda \cdot (m_{air} / m_{fuel})_S} \tag{4}$$

$(m_{air} / m_{fuel})_S$ is the stoichiometric air/fuel ratio. λ should be equal to 1.

The other models used by the ECM are described in the following chapters.

3.3 Injection system

The purpose of the injectors is to inject fuel into the intake manifold. The actual amount injected depends on a lot of different things: Density, supply voltage, engine speed and pulse width of the injector current.

The density is given by the ideal gas law:

$$p_{inj}V = m_{fuel}RT_{inj} \tag{5}$$

which means that

$$\rho_{inj} = \frac{p_{inj}}{RT_{inj}} \tag{6}$$

R is the gas constant and should not be mixed up with the molar gas constant.

The temperature as well as the pressure will affect the behaviour of some parts inside the injectors. This will have influence on the amount injected.

The injectors open faster when the voltage supply is high. This is because the magnetic force created in the solenoid is stronger.

It takes some time for the injectors to open. If it has to open and close very frequently, i.e. if the engine speed is high, it is not completely open during a great part of the electrical pulse. As a result, the fuel flow is affected.

Figure 9 shows a flow chart for the injection algorithm. Engine speed and supply voltages are measured directly by the GSI sensors. T_{inj} and p_{inj} have to be measured indirectly. The models creating the necessary links between the GSI sensors and T_{inj} and p_{inj} will be discussed in the following chapters.

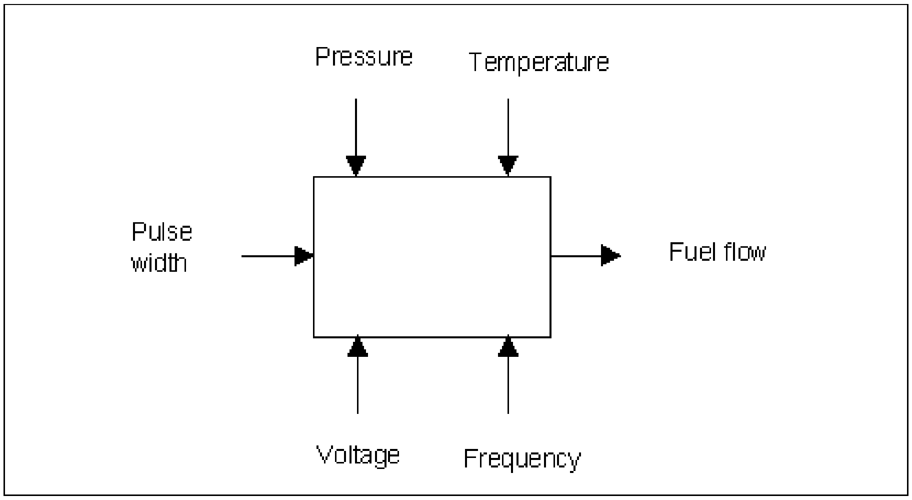

Figure 9: *Injection model*

4 Modelling the tank temperature

Most likely, the temperature of the tank will be close to the ambient temperature. Unfortunately, there is not any sensor that measures the temperature of the tank or the ambient temperature. Therefore, the temperature must be estimated with the help of other sensors. This is done in the same way as with the ambient temperature in *3.1*.

An approximation of the tank temperature that works well in most cases is

$$T_{Tank} = \begin{cases} T_{cool}(0) & if\ T_{cool}(0) \le T_{average} \\ T_{average} & if\ T_{average} < T_{cool}(0) \le T_{\max}\ and\ t_{key-off} \le t_{\max} \\ T_{cool}(0) & if\ T_{average} < T_{cool}(0) \le T_{\max}\ and\ t_{key-off} > t_{\max} \\ T_{average} & if\ T_{cool}(0) > T_{\max} \end{cases} \quad (7)$$

Suitable values for $T_{average}$, T_{max} and t_{max} are 10^{o}C, 35^{o}C and 3-4 hours respectively.

5 Modelling the vaporiser/pressure regulator

The interesting properties in the vaporiser/pressure regulator (*Figure 10*) are the second stage pressure and the outlet fuel temperature. This is due to the fact that the fuel temperature at the injectors depend on the fuel temperature at the vaporiser/pressure regulator (V/P), and the pressure at the injectors depends on the second stage pressure.

The first stage pressure is needed for the calculation of the outlet temperature. One of the reasons is that the boiling temperature in the first stage chamber, which is included in the expression for the outlet temperature, depends on the first stage pressure. The other reason is that the pressure drop between the first and second stage chamber will affect the temperature.

Below, the modelling of the first stage pressure (p_{stage1}), the second stage pressure ($p_{V/P}$) and the outlet fuel temperature ($T_{V/P}$) are described.

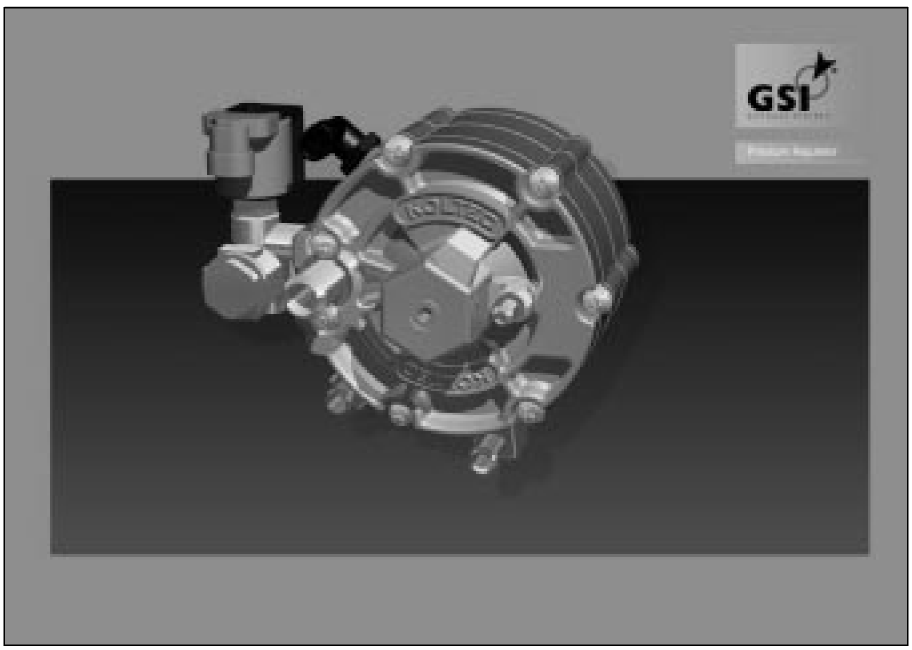

Figure 10: The vaporiser/pressure regulator

5.1 Pressures - Steady state

The principal idea behind the vaporiser/pressure regulator is a pressure working on a diaphragm, which in its turn presses down a spring. The first and second stage pressures caused by this action should be 1.4 and 0.96bar respectively above the reference pressure. Experience shows that this is not always the case. The pressure varies due to that the behaviour of the diaphragm and springs depends on temperature and flow.

The V/P encloses a lot of parts that almost all affect the pressure in some way. Add that the quality of the pieces fluctuate. Then it is not difficult to realise that it is hard, if not to say almost impossible, to get a high accuracy of the physical model in the whole working range. At the department of engine management systems (TNO), a model describing the pressures in the V/P has been developed, but it is not working sufficiently. Instead of trying to improve this model, which is a very time consuming job and most likely results in an exceedingly complicated algorithm when implemented, identification has been used.

5.1.1 Measurements

For the identification of how the pressures are depending on coolant temperature and mass flow, the first and second stage pressures were measured. This was done for nine coolant temperatures (10, 20,..., 90^{o}C), six fuel mass flows (1, 3, 5, 6, 8 and 10Nm^3/h[1]) and one inlet manifold pressure of 100kPa.

The results of the measurements are shown in *Appendix G*.

The discussion of causes of measurement errors in *5.2.2* is valid also in this case, since the steady state measurements for the pressures as well as the temperature were done at the same point in time.

5.1.2 First stage pressure

When the relationship between the heat transfer factor and hose lengths was determined (in *6.1.3.1* and *6.1.3.3*), the general idea of the models was clear from the beginning. Only the model parameters had to be identified. In this case however, there was no obvious form of the model. Different types had to be studied and compared with each other.

[1] Normal cubic meter per hour

The following attempts were made (units for m_{fuel} and T_{cool} are kg/s and Kelvin)

1. $p_{stage1} = a + bm_{fuel} + cT_{cool}$

2. $p_{stage1} = a + bm_{fuel}$

3. $p_{stage1} = a + bm_{fuel} + cm_{fuel}^2 + dT_{cool}$

4. $p_{stage1} = a + bm_{fuel} + cT_{cool} + dT_{cool}^2$

The parameters were estimated by regression analysis. After that, a comparison was made between the standard deviations between the values predicted by the model and the measured ones (*Table 2*).

Table 2: The average difference between the measured and predicted values

Model	std
1	1.81
2	1.81
3	1.82
4	1.83

According to the standard deviations, the models are almost equally good. Since model 2 is the simplest one, it is the best choice. The look of model 2 is (p_{im} = 100kPa)

$$p_{stage1}[kPa] = 242 - 3.3 \cdot 10^3 \cdot m_{fuel} \tag{8}$$

The first stage pressure depends of the inlet manifold pressure. This is an approximation that works at inlet manifold pressures close to 100kPa[2]:

$$p_{stage1} = p_{im} + 142 - 3.3 \cdot 10^3 \cdot m_{fuel} \qquad (9)$$

5.1.3 Second stage pressure

The following models were studied

1. $p_{V/P} = a + bm_{fuel} + cT_{cool}$
2. $p_{V/P} = a + bm_{fuel}$
3. $p_{V/P} = a + bm_{fuel} + cm_{fuel}^2 + dT_{cool}$
4. $p_{V/P} = a + bm_{fuel} + cT_{cool} + dT_{cool}^2$
5. $p_{V/P} = a + bm_{fuel} + cm_{fuel}^2$

The standard deviations between the values predicted by the models and the measured ones are shown in *Table 3*.

As can be seen, model number 3 gives the best result. However, the model is more complicated than model number 1. Although model number 1 is not really as good as number 3, it is still describing the measured data very well (the standard deviation is about 0.5% of the pressure value) and would be good enough.

Model number 1 has the following appearance (p_{im} = 100kPa)

[2] The measurements were done for only one inlet manifold pressure. To get an expression that works on a wider range, new measurements have to be made.

$$p_{V/P}[kPa] = 221 - 1.0 \cdot 10^3 \cdot m_{fuel} - 0.068 \cdot T_{cool} \tag{10}$$

Table 3: The average difference between the measured and predicted values

Model	std
1	1.03
2	1.7
3	0.84
4	1.04
5	1.59

An approximation that works when the inlet manifold pressure is about 100kPa is

$$p_{V/P}[kPa] = 121 - 1.0 \cdot 10^3 \cdot m_{fuel} - 0.068 \cdot T_{cool} \tag{11}$$

5.2 Temperature - Steady state

Most of the formulas used below can be found in [1].

The pressure in the vaporiser/pressure regulator is lower than the pressure in the tank. A lower pressure also means a lower boiling temperature. The liquid fuel vaporises due to the pressure decrease. The energy needed for the vaporisation process is taken from the internal energy of the fuel. If there is no external energy supply, the temperature will decrease until it reaches boiling temperature or the fuel is fully vaporised. In this case however, energy is transferred from the coolant. When the fuel is completely vaporised, the temperature will rise on condition that it is still lower than the coolant temperature. To summarise the paragraph above, energy is coming from

– The coolant by heat transfer (Q_{tr})

– The heat released by the pressure decrease (Q_{rel})

while energy is consumed by

– Vaporisation (H)

– Heating of fuel (Q_{abs})

or, written in another way,

$$Q_{tr} + Q_{rel} = H + Q_{abs} \tag{12}$$

The heat change caused by a temperature increase of the fuel can be written as

$$Q = c_p m \Delta T \tag{13}$$

where c_p is the heat capacity, $\dot{m}$ is the mass flow and ΔT is the change in temperature of the fuel. c_p is fluctuating somewhat with the temperature, but the most dramatic difference is when the LPG changes from liquid to gas. As a consequence, c_p is in this work approximated with two fixed constants; $c_{p,liq}$ and $c_{p,gas}$. Q_{rel} and Q_{abs} can now be written in the following way

$$Q_{rel} = c_{p,liq} m_{fuel} \left(T_{Tank} - T_{boil}\right) \tag{14}$$

$$Q_{abs} = c_{p,gas} m_{fuel} \left(T_{stage1} - T_{boil}\right) \tag{15}$$

T_{Tank} represents the fuel temperature in the tank and T_{boil} the boiling temperature corresponding to the pressure in the first stage chamber. The relationship between boiling temperature and pressure is shown in *Figure 11*.

Energy consumed by the vaporisation process is

$$H = r m_{fuel} \tag{16}$$

where r is the heat of vaporisation.

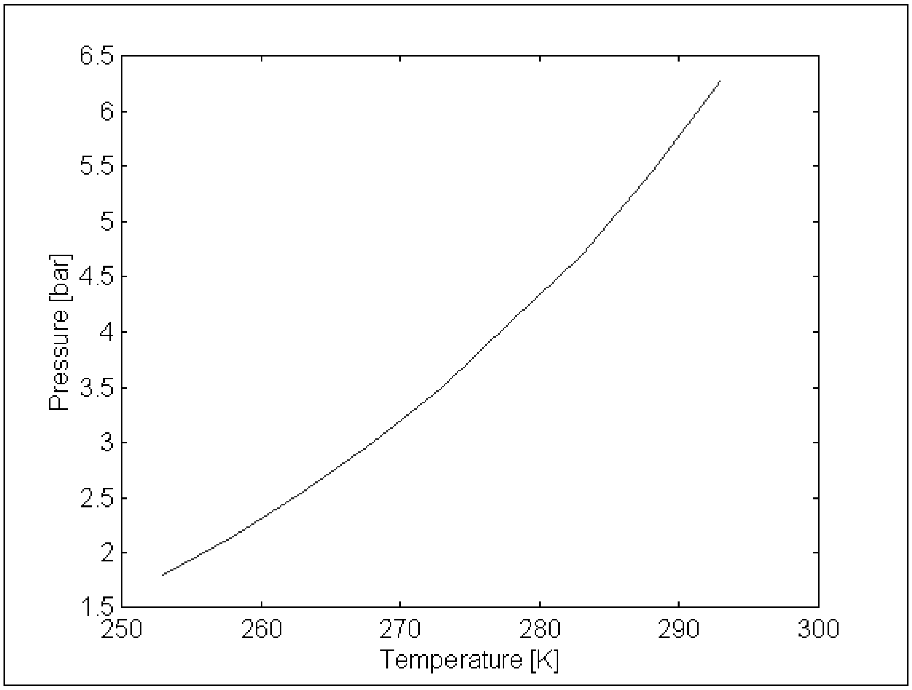

Figure 11: Absolute pressure as function of boiling temperature (60% propane and 40% butane).

The heat flow from the coolant to the fuel is

$$Q_{tr} = UA\Delta T \tag{17}$$

where U is a heat transfer constant and A is the area the heat is transferred through. It is difficult to determine the area in this case, hence UA will be treated as a single constant, instead of two separate constants. ΔT is the mean temperature difference between the coolant and the fuel, which can be written as (if assumed that the coolant temperature will stay the same between the inlet and outlet)

$$\Delta T = \frac{T_{stage1} - T_{boil}}{\ln\left[\left(T_{cool} - T_{boil}\right) / \left(T_{cool} - T_{stage1}\right)\right]} \tag{18}$$

An explanation of the expression above is given in *Appendix D*.

(12) together with (14) - (18) gives

$$T_{stage1} = T_{cool} - \left(T_{cool} - T_{boil}\right) \cdot e^{-\frac{UA}{\dot{m}_{fuel} c_{p,gas}} \left(\frac{f(T_{boil})}{T_{stage1} - T_{boil} + f(T_{boil})} \right)} \tag{19}$$

where

$$f(T_{boil}) = \frac{r - c_{p,liq}\left(T_{Tank} - T_{boil}\right)}{c_{p,gas}} \tag{20}$$

(19) can not be solved for T_{stage1}. To be able to calculate its value for a certain fuel mass flow, one can calculate it recursively, i.e. use the value calculated for a somewhat lower flow. The initial value to this recursive loop is $T_{stage1}(0) = T_{cool}$.

When entering the second stage chamber, the pressure drops. This results in a temperature decrease. Due to that [2]

$$Tp^{-\frac{\kappa - 1}{\kappa}} = const \tag{21}$$

$T_{V/P}$ may be written

$$T_{V/P} = T_{stage1}\left(\frac{p_{stage1}}{p_{V/P}}\right)^{-\frac{\kappa-1}{\kappa}} \qquad (22)$$

5.2.1 Measurements

For the calculation of *UA* and to check whether the model describes the behaviour of the vaporiser sufficiently, the fuel temperature after the regulator was measured. This was done for nine coolant temperatures (10, 20, ..., 90^{o}C) and six fuel mass flows (1, 3, 5, 6, 8 and 10Nm^3/h).

5.2.2 Results of the measurements

The measurement results are presented in *Figure 12* and *Appendix E*. The results for T_{cool} = 10^{o}C have been removed, because these measurements were done at another point in time than the other ones were. The adjustments of the vaporiser/pressure regulator were different at the two occasions, which affected the outcome of the experiments. This is no big loss, because the only time the fuel did not come out as liquid was at m_{fuel} = 1Nm^3/h.

Things that might have reduced the accuracy of the results were a fluctuating coolant temperature, and that we sometimes might have been too impatient when we measured. As will be described in *5.3*, the fuel temperature changes slowly for a step in flow, and in some cases we might have thought that it was steady state when it was not. m_{fuel} was not always at the desired value. The inaccuracy could be as high as 0.5Nm^3/h at high flows or 20% off at low flows. Finally, the sensor's ability to measure the temperature correct decreases with decreasing flow.[3]

[3] It can be noticed from the experiments that part of the fuel came out of the vaporiser/pressure regulator in liquid form at temperatures much higher than the boiling temperature (about minus 13^{o}C when p_{im} = 100kPa). The model assumes that the fuel becomes fully vaporised before it leaves the regulator, which means that it is not valid when part of the fuel comes out as liquid even if $T_{V/P}$ is far above boiling temperature. Probably, the lowest temperature at which the fuel is still fully vaporised depends on p_{im}.

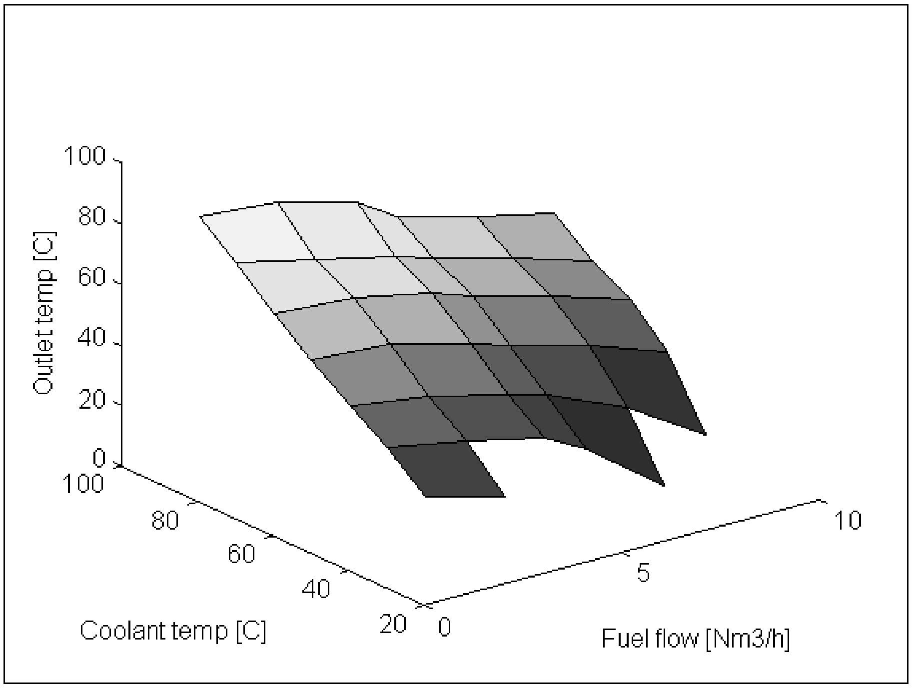

Figure 12: Measured outlet temperature

5.2.3 Determining UA

By inverting and combining (19), (20) and (22), one gets the following expressions

$$T_{stage1} = T_{V/P}\left(\frac{p_{V/P}}{p_{stage1}}\right)^{\frac{\kappa-1}{\kappa}} \tag{23}$$

$$UA = \frac{m_{fuel}\left(r + c_{p,gas}\left(T_{stage1} - T_{boil}\right) - c_{p,liq}\left(T_{Tank} - T_{boil}\right)\right)}{\Delta T} \tag{24}$$

The calculated *UA* values can be found in *Appendix F*. When calculating *UA* for a flow of 1Nm^3/h the value becomes complex, since $T_{V/P}$ is higher than T_{cool} minus the temperature drop caused by the decrease in pressure. This means that according to the model, T_{stage1} is higher than the coolant temperature. This is of cause not possible. An explanation of this strange calculation outcome is that the fuel is heated up a bit in the second stage chamber and not only in the first stage chamber.

After removing the *UA* values for 1Nm^3/h, the average value is 50W/K.

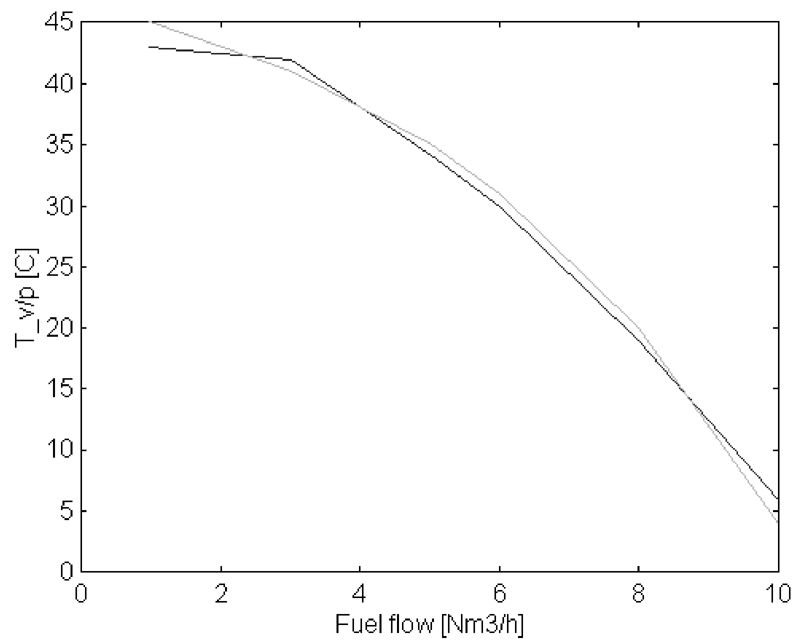

Figure 13: $T_{V/P}$ *as function of* $\dot{m}_{fuel}$ *at* $T_{cool} = 50^oC$*. The darker line shows the calculated values while the lighter line shows the measured values.*

5.2.4 The model's ability to describe the temperature

A comparison between the measured temperatures after the regulator and the ones predicted by the model at T_{cool} = 50^{o}C with UA = 50W/K, gives that the model describes the reality very well (*Figure 13*).

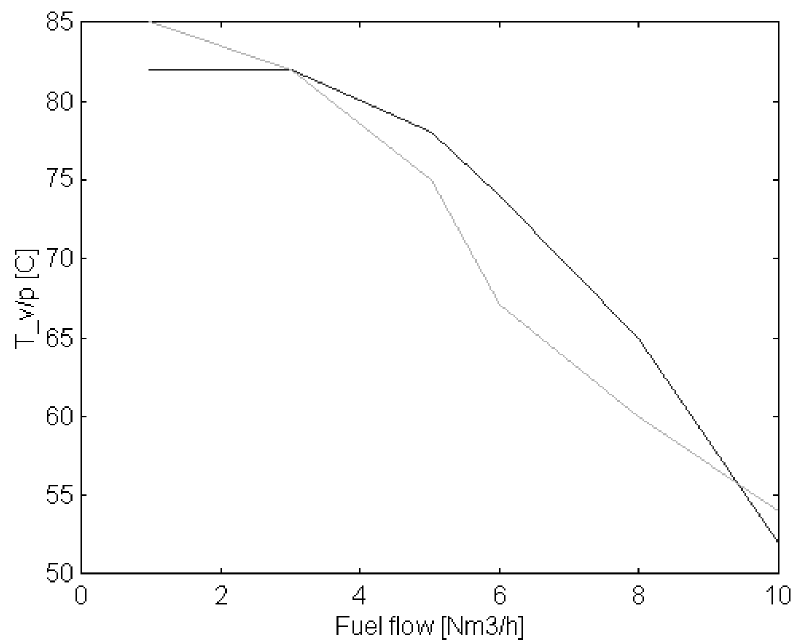

Figure 14: $T_{V/P}$ *as function of* $\dot{m}_{fuel}$ *at* T_{cool} *= 90^{o}C. The darker line shows the calculated values, while the lighter line shows the measured values.*

The model performance at T_{cool} = 50^{o}C is interesting, because for lower temperatures the fuel came out in liquid form when the flow was high. At higher temperatures, the measurement error is bigger, because it was difficult to keep the temperature of the coolant stable. The greatest deviation between the model and the reality is at T_{cool} = 90^{o}C (*Figure 14*). At this temperature, we had to heat the coolant up after having measured for some of the flows, because it was far too low (almost 5^{o}C off). This explains the strange shape of the curve. Despite the big measuring error, the difference between the values predicted by the model and the measured values is not alarmingly big.

5.3 Temperature - Transient behaviour

The temperature of the fuel after the vaporiser/pressure regulator depends mainly on three parameters, according to (19) and (22). These are the fuel mass flow, the coolant temperature and the pressure ratio ($p_{V/P}/p_{stage1}$). The temperature of the coolant changes so slowly, that the transient behaviour of the fuel temperature caused by a change in coolant temperature is not worth mentioning, while the transient behaviour caused by a change in fuel mass flow and pressure ratio has to be described. Experiments, done at the same time as the steady state measurements, show that the fuel temperature changes significantly slower than the fuel flow does. The flow changes almost immediately, while the change in temperature takes a couple of minutes (*Figure 15*).

The slow change of the temperature of the fuel coming out of the vaporiser/pressure regulator, is probably due to that the wall separating the coolant from the fuel is affected not only by the temperature of the coolant, but also of how hot the fuel is. If the fuel is cold, the part of the wall that is closest to the fuel will also be cold. If the mass flow is decreased, more heat is transferred to the fuel because it stays longer inside the regulator. That means that the fuel temperature will increase, and so will the temperature of the part of the wall closest to the fuel. Some of the heat released by the coolant will thus be used to heat the walls instead of the fuel.

It is natural that the bigger the difference is between the steady state temperature and the actual temperature, the quicker the temperature of the fuel will change. A simple model describing the transient behaviour is thus

$$Q \sim [T_{V/P}]_{SS} - T_{V/P} \tag{25}$$

$$\frac{dT_{V/P}}{dt} \sim Q \tag{26}$$

where $[T_{V/P}]_{SS}$ is the value given by (19).

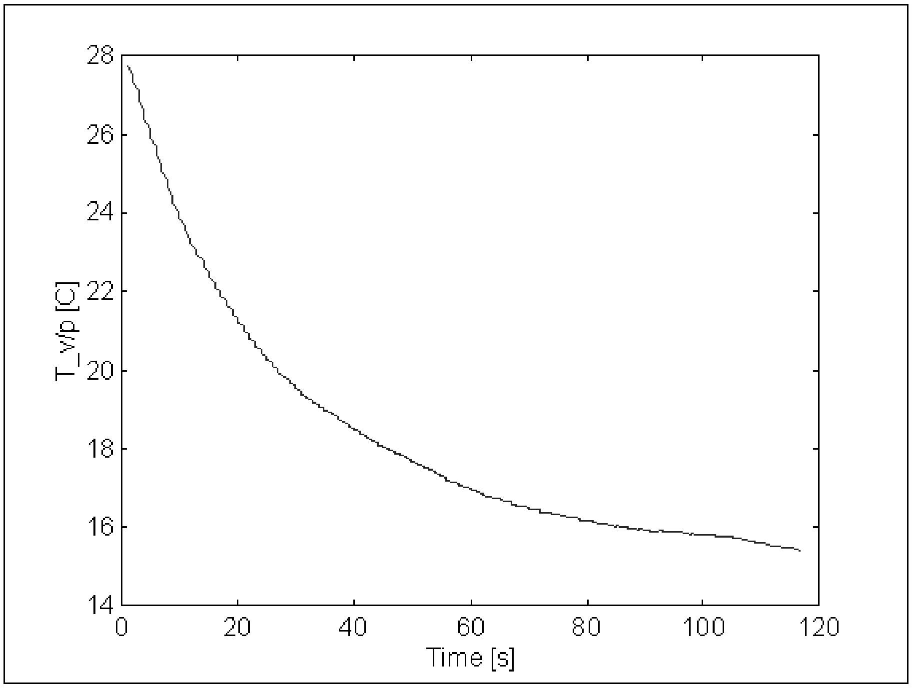

Figure 15: *Response of a step in fuel mass flow (1-4Nm^3/h) when the coolant temperature is 30^{o}C.*

(25) and (26) gives

$$\frac{dT_{V/P}}{dt} = \frac{1}{\tau} \cdot \left([T_{V/P}]_{SS} - T_{V/P} \right) \tag{27}$$

5.3.1 Measurements

To be able to study the transient behaviour, steps in mass flow (2 -> 5Nm^3/h, 5 -> 2Nm^3/h, 2 -> 8Nm^3/h and 8 -> 2Nm^3/h) were made. Each step was done twice, because these measurements were combined with the transient measurements for the connection between the vaporiser/pressure regulator and the hoses (they were done for two different hose lengths). The coolant was about 80^{o}C.

Unfortunately, at the time the measurements were done, it was not clear that the ratio $p_{stage1}/p_{V/P}$ was an important parameter for the calculation of $T_{V/P}$, so no measurements were done for steps in inlet manifold pressure.

The results of the measurements are shown in *Appendix H*.

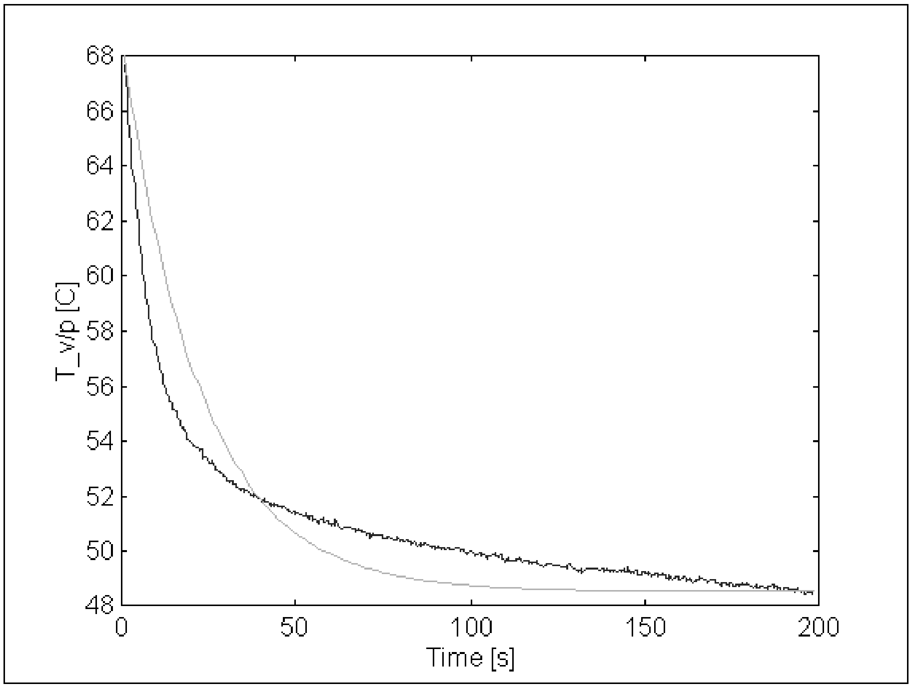

Figure 16: *The response from a step in mass flow (2->8Nm^3/h). The lighter line shows the measured temperature, while the darker line shows the predicted temperature.*

5.3.2 Determining 1/τ

The solution to equation (27) is

$$T_{V/P}(t) = [T_{V/P}]_{SS} - \left([T_{V/P}]_{SS} - T_{V/P}(0)\right)\cdot e^{-\frac{t}{\tau}} \tag{28}$$

where $\left[T_{V/P}\right]_0$ is the value given by (19) when using the previous flow. τ can thus be calculated with the following expression

$$\frac{1}{\tau} = \frac{1}{t}\ln\left(\frac{T_{V/P}(0) - \left[T_{V/P}\right]_{SS}}{\left[T_{V/P}\right]_{SS} - T_{V/P}(t)}\right) \tag{29}$$

The calculated $1/\tau$ values were between 0.025 and $0.06 s^{-1}$, the higher values for the higher flows. A compromise was to set $1/\tau$ to $0.045 s^{-1}$.

5.3.3 The models ability to describe the temperature

To set $1/\tau$ to $0.045 s^{-1}$ gave the result that the difference between the measured temperature and the one given by the transient model was never higher than $4.5^{o}C$ (see *Figure 16* which shows the step response that had the worst correspondence with the model). This will cause an inaccuracy of the density of about 1.5%.

The model derived above is very simple. In fact, the heat has a two dimensional spreading through the wall that separates the fuel from the coolant. A combination of two first order filters with different time constants might give a higher correspondence to the measured data. On the other hand, this might make the model unnecessarily complicated.

5.4 Summary

When p_{im} is close to 100kPa are

$$p_{stage1}[kPa] = p_{im} + 142 - 3.3 \cdot 10^3 \cdot m_{fuel}$$

$$p_{V/P}[kPa] = p_{im} + 121 - 1.0 \cdot 10^3 \cdot m_{fuel} - 0.068 \cdot T_{cool}$$

$$\frac{dT_{V/P}}{dt} = \frac{1}{\tau} \cdot \left([T_{V/P}]_{SS} - T_{V/P} \right)$$

$$[T_{V/P}]_{SS} = T_{stage1} \left(\frac{p_{stage1}}{p_{V/P}} \right)^{\frac{\kappa - 1}{\kappa}}$$

$$T_{stage1} = T_{cool} - (T_{cool} - T_{boil}) \cdot e^{-\frac{UA}{\dot{m}_{fuel} c_{p,gas}} \left(\frac{f(T_{boil}, T_{Tank})}{T_{stage1} - T_{boil} + f(T_{boil}, T_{Tank})} \right)}$$

$$f(T_{boil}, T_{Tank}) = \frac{r - c_{p,liq} (T_{Tank} - T_{boil})}{c_{p,gas}}$$

$UA = 50$ W/K and $1/\tau = 0.045 s^{-1}$

6 Modelling the connections between the vaporiser/pressure regulator and the injectors

The temperature of the fuel close to the point where the fuel is injected (T_{inj}) is a very important parameter in the calculations of the injection time, because when the temperature changes, the density will change. After the fuel leaves the vaporiser/pressure regulator and before it enters one of the injectors, it will pass through a couple of hoses and a rail. Naturally, the fuel will either warm up or cool down during this path if its temperature differs from the surrounding.

The locations of the temperatures that are mentioned in the subchapters below are shown in *Figure 17*.

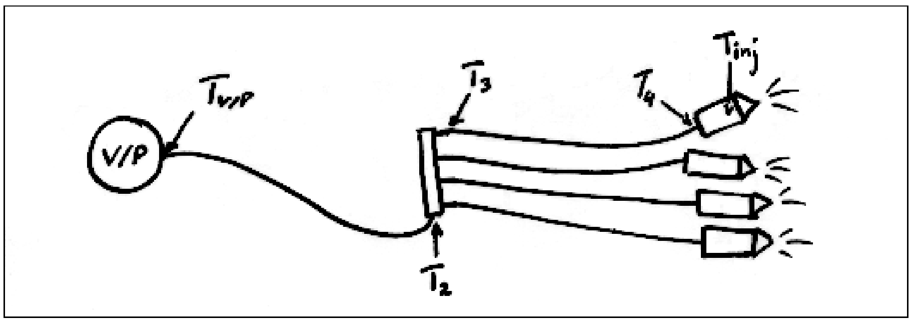

Figure 17: The different temperatures of the connector

6.1 Temperature - Steady state

The heat flow to/from the surrounding air is

$$Q_{surround,i} = U_i A_i \Delta T_i \tag{30}$$

where U_i is a heat transfer constant for the part between T_{i-1} and T_i, A_i is the area the heat is transferred through and

$$\Delta T_i = \frac{T_i - T_{i-1}}{\ln\left[\left(T_{eng.c.} - T_{i-1}\right)/\left(T_{eng.c.} - T_i\right)\right]} \tag{31}$$

if assumed that the temperature of the surrounding air will be the same everywhere around the connection[4]. This temperature is called $T_{eng.c.}$ as in engine compartment temperature.

The heat absorbed/emitted by the fuel is

$$Q_{fuel,i} = c_{gas} m_{fuel} \left(T_i - T_{i-1}\right) \tag{32}$$

Since the heat transferred to/from the surrounding air is absorbed/emitted by the fuel, the two energy flows above are equal, i.e.

$$Q_{surround,i} = Q_{fuel,i} \tag{33}$$

This results in

$$T_i = T_{eng.c.} - \left(T_{eng.c.} - T_{i-1}\right) \cdot e^{-\frac{U_i A_i}{\dot{m}_{fuel} c_{p,gas}}} \tag{34}$$

Simple calculations give that

$$T_{inj} = T_{eng.c.} - \left(T_{eng.c.} - T_{V/P}\right) \cdot e^{-\frac{1}{\dot{m}_{fuel} c_{p,gas}} \sum U_i A_i} \tag{35}$$

[4] See Appendix D

$T_{V/P}$ is the temperature of the fuel coming out of the vaporiser/pressure regulator.

There will be a temperature drop over the injectors caused by the decrease in pressure. If assumed that T_{inj} = 50^{o}C, p_{inj} = 196kPa and p_{im} = 100kPa, the temperature drop is as big as 22^{o}C. However, this temperature drop is of no importance to the algorithm.

6.1.1 Measurements

For the calculation of the different heat transfer factors, temperatures were measured at five places: Just after the vaporiser/pressure regulator, before the rail, after the rail, before one of the injectors and after one of the injectors. This was done for three flows: 2, 4 and 6Nm^3/h[5].

The sensor measuring the temperature after the rail was in all cases but one placed at the hose farthest from the hose connecting the regulator to the rail. So were also the sensors for measuring the injector temperatures. To see the difference in temperature between the hose closest to the hose connecting the regulator to the rail and the farthest, a set of measurements were done with the sensors placed at the former and at the injector belonging to it.

Different hose lengths were used. The hose connecting the regulator to the rail was 0.5, 1.0, 1.5 or 2.57m long, while the hoses between the rail and the injectors were 0.3, 0.5, 0.75, 0.97 or 2.0m. To make sure that all the connections between the rail and the injectors were of the same length although sensors were placed before and after one of the hoses, the measured hose was shortened. This resulted in the following hose lengths; 0.22, 0.38, 0.63, 0.85 and 1.88m. Note that the difference in lengths for the shortest hose is only 8cm compared to 12cm for the other hoses. This is because we would not have been able to assemble the different parts with a shorter hose.

The coolant temperature was between 80 and 90^{o}C.

The experiments were made in a special test set-up. It consisted of a vaporiser/-pressure regulator, a rail made of metal, metal injectors, metal inlet manifold and rubber hoses.

6.1.2 Results of the measurement

The results of the measurement are presented in *10Appendix C*.

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[5] Normal cubic meter per hour

An important factor for the calculation of U_iA_i is the ambient temperature. It was during these experiments about 14^{o}C, but varied somewhat depending on where it was measured. For example, it was a couple of degrees higher close to the vaporiser/pressure regulator.

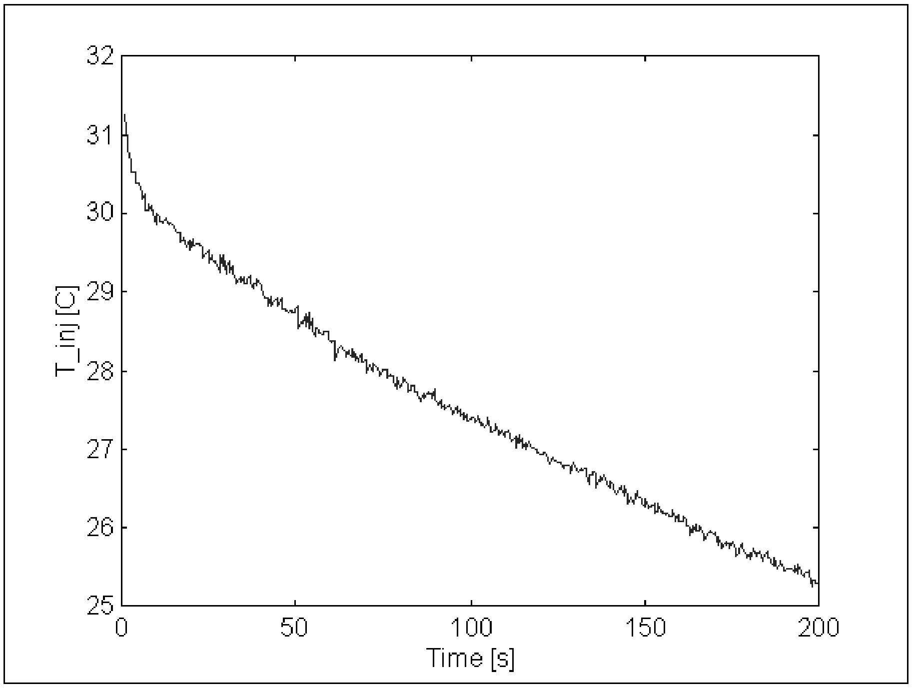

Figure 18: Response for a step in fuel mass flow (8->2Nm^3/h). The lengths of the hoses used are 2.57 and 2.00m (the hose between the V/P and rail mentioned first).

It took a long time for the system to stabilise. The temperature at the injectors increased/decreased very slowly, about one degree a minute, which made it difficult to notice the change. A step response for a change in fuel mass flow is shown in *Figure 18*. The response was measured during only 200 seconds, and it is obvious that the temperature did not reach steady state during that time. It is possible that the steady state measurements were done too early.

The values for the temperature of the fuel after the injectors are unreliable. They are highly correlated to the temperature of the manifold, since the manifold as well as the injectors are made of metal and joined together. The manifold warms up/cools down very slowly. That means that if there has been a big fuel flow for a time, the manifold will have been heated up and when the flow changes to a much lower value, the temperature after the injectors will still be high for a very long time. Luckily, the heat transfer factor over the injectors are not of much interest. The new injectors will be made of plastic, which do not conduct heat very well. The most interesting temperature is thus the temperature just before the injectors.

6.1.3 Determining the heat transfer factors

U_iA_i can be calculated using the following equation

$$U_i A_i = m_{fuel} c_{p,gas} \ln\left(\frac{T_{eng.c.} - T_{i-1}}{T_{eng.c.} - T_i}\right) \tag{36}$$

As the temperature values might have been measured before steady state was reached, one have to be careful when using the measured values. In addition to the measurements defined above, there were also some step responses made which will be described later on (*6.2.1*). By studying these step responses, it is possible to see that one heat transfer constant converges towards a certain value.

The transient temperature behaviour is depending both directly and indirectly on a change in fuel mass flow. A higher flow means a lower temperature loss/increase in the connection, but it also means that the temperature after the vaporiser/pressure regulator will drop. A change in temperature will spread its effect slowly through the connection, which means that with shorter hoses, it will reach steady state faster than with long hoses. Thus, the results for short hoses can often be considered more reliable than the ones for long hoses.

6.1.3.1 The hose between the vaporiser/pressure regulator and the rail

The heat transfer factors (calculated from the measured data) for the hose that connects the vaporiser/pressure regulator with the rail are shown in *Table 4*. By studying the step responses mentioned earlier (one of them is shown in *Figure 19*), one can see that the heat transfer factor relatively early converges towards a specific value. This value is about the same as in *Table 4*. It seems like the calculated values are reliable.

The heat transfer factor can also be written *kl*, where $k = 2\pi rU$ and l is the hose length. This is convenient in practical use, because there is no fixed value for the hose length. The length of the hose depends on the type of car in which the GSI is implemented.

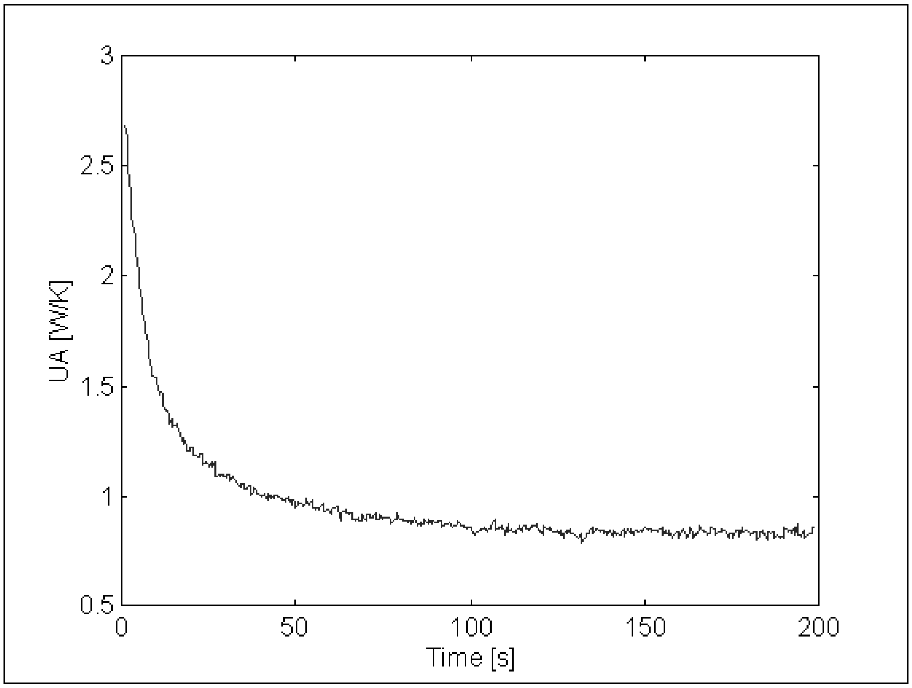

Figure 19: The heat transfer factor for the hose between the regulator and the rail as a function of time after a step in fuel mass flow (2->8Nm3/h). The hose is 1.0m

Table 4: Heat transfer factor for the hose connecting the regulator with the rail

Hose length [m]	*UA* [W/K]
0.5	0.68
1.0	0.88
1.5	1.2
2.57	1.9

Regression analysis [6] gives the following expression for the heat transfer factor

$$UA[{}^{W}\!/\!_{K}] = 0.60l + 0.33 \tag{37}$$

which means that $k = 0.60$ W/Km and that the error caused by the test set-up (for example extension of the hose) can be identified to 0.33W/K. The heat transfer factors calculated from the measured data compared to the ones predicted by the model can be seen in *Figure 20*.

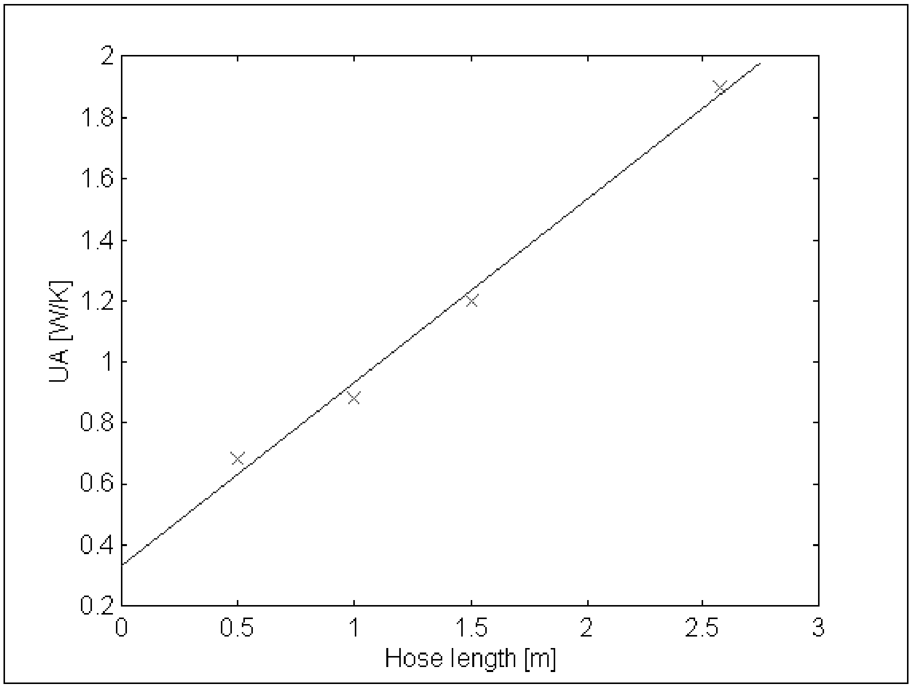

Figure 20: Heat transfer factor as function of hose length. The line shows the values predicted by the model and the x represents the measured values.

6.1.3.2 The rail

Step responses for the heat transfer factor show that the factor for the rail does not converge as quick as the factor for the hose. The information the step responses made with long hoses can give, is that the factor is higher than 0.4W/K and lower 0.7W/K. For the short hoses however, there is a tendency to converge towards a value close to 0.55W/K.

There were no step responses made when the sensors were placed closest to the hose connecting the regulator to the rail. The only available data is therefore *Measurement 1a, Appendix E*. This gives a heat transfer factor of 0.37W/K.

Table 5: Heat transfer factor for the hose connecting the rail with the injectors

Hose length [m]	*UA* [W/K]
0.22	0.82
0.38	1.13
0.63	1.95
0.85	2.77
1.88	4.37

It is not possible, based on the available data, to determine exactly how big the influence is from the measurement equipment. If assumed that half the heat transfer for the short part of the rail is caused by the measurement equipment, the resulting factors becomes 0.37 and 0.18W/K respectively.

6.1.3.3 The hoses between the rail and the injectors

Studies of the step responses show that the heat transfer factor that converges most slowly towards a stable value is the factor for the hoses between the rail and the injectors. The values could therefore be suspected to be inaccurate. Nevertheless, the information given by the step responses is that *UA* has a value between 0.9 and 1.2W/K when the hose is 0.5m, and between 2.2 and 4.5W/K when the hose is 2.0m. This is according to the values in *Table 5*.

Regression analysis results in the following relationship between *UA* and hose length

$$UA[{}^{W}\!/_{K}] = 2.15l + 0.50 \qquad (38)$$

k in *UA = kl* is thus 2.15W/Km and the error caused by the test set-up 0.50W/K. The values predicted with this model compared to the measured ones are shown in *Figure 21*.

6.2 Temperature -Transient behaviour

A change in mass flow has both a direct and an indirect effect. If the flow is reduced, the fuel will spend longer time in the connection, which causes a higher rate of heat transferred through the walls. On the inside of a hose, the temperature is close to the fuel temperature, while at the outside it is close to the surrounding temperature. As a consequence, the temperatures of the walls have to change when the fuel temperature changes. This might take some time.

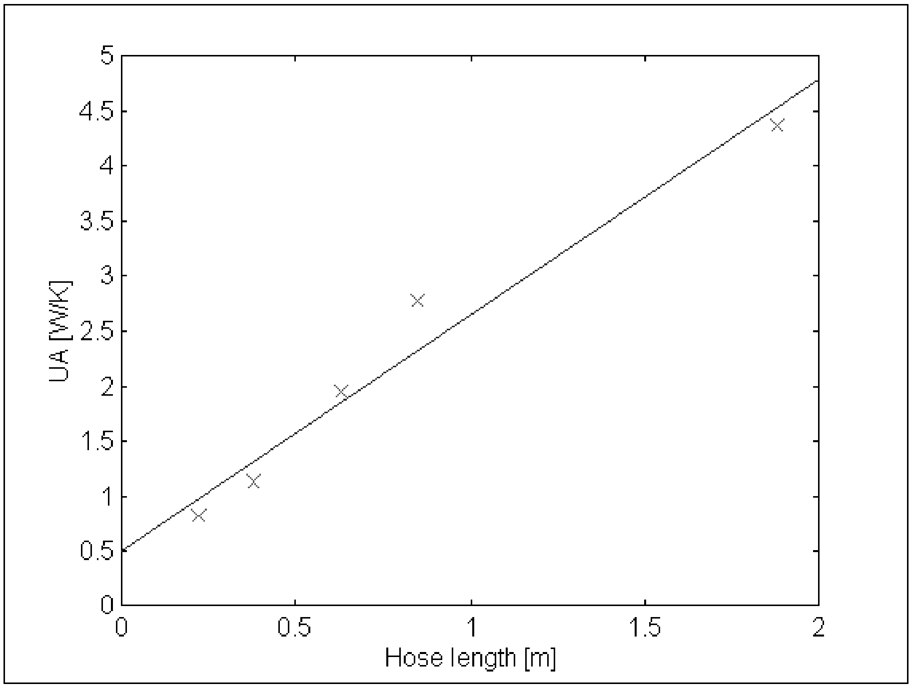

Figure 21: Heat transfer factor as function of hose length. The line shows the values predicted by the model and the x represents the measured values.

The indirect effect of a decrease in fuel flow is that $T_{V/P}$ will be closer to the coolant temperature than before. The temperature of the part of the connection closest to the rail will be about the same as $T_{V/P}$ while the part on the other side, closest to the injectors, will be related to T_{inj}. When $T_{V/P}$ changes, the temperature of the connection walls also has to change. This will take quite some time, because the change in temperature has a long way to travel.

A very simple description of the transient behaviour of T_{inj} is

$$Q \sim [T_{inj}]_{SS} - T_{inj} \tag{39}$$

$$\frac{dT_{inj}}{dt} \sim Q \tag{40}$$

(39) and (40) gives

$$\frac{dT_{inj}}{dt} = \frac{1}{\tau}\left([T_{inj}]_{SS} - T_{inj}\right) \tag{41}$$

6.2.1 Measurements

Steps in mass flows (8 -> 2Nm^3/h, 5 -> 2Nm^3/h, 2 -> 8Nm^3/h and 2 -> 5Nm^3/h) were made. Two combinations of hose lengths were used; 1.0m together with 0.5m and 2.57m together with 2.0m (the hose between V/P and the rail mentioned first). The temperatures in *Figure 17* were all measured as well as the mass flow.

The sample time was 0.5s.

6.2.2 Determining τ

As explained before, there are different causes for the transient behaviour. Steps in fuel mass flow gives both direct and indirect effects and the temperature change in the walls are two-dimensional (radial direction and along the lengths of the hoses). The transient model above, represented by equation (41), is too simple to describe the behaviour sufficiently. However, if the first quick change is ignored (see *Appendix C*), the model describes the changes quite well.

(41) was implemented in MatrixX, a software application that allows you to simulate models. $[T_{inj}]_{SS}$ was calculated from the measured $T_{V/P}$ according to (35). This was done for each sample. By trying various time constants and making a qualitative comparison between the results, the time constants shown in *Table 6* proved to be the ones who described the slow changes best. The graphs generated for the medium long hoses are shown in *Appendix C*.

Table 6: *$1/\tau$ [s^{-1}]*

	Medium long hoses	Long hoses
8->2Nm^3/h	0.0065	0.0019
5->2Nm^3/h	0.007	0.002
2->8Nm^3/h	0.0095	0.005
2->5Nm^3/h	0.007	0.0045

Note that the temperature adjusts much faster to the steady state value when the shorter hoses are used. This is exactly what could be expected. When the hose length increases, the effect of a change in $T_{V/P}$ will have a greater length to pass through. Note also that there seems to be a correlation between the time constant and mass flow.

When doing the fittings above, the first quick change had to be removed in some way. This was done by setting the initial temperature value to a value such as the calculated and the measured values became equal at the point where the quick change faded out. The differences between the initial temperatures used in the calculations and the measured temperatures are shown in *Table 7*. It was difficult to make a good fit with the measurements done for the long hoses. Thus, the values for the long hoses should not be put to much weight to. It is good, though, to observe that these values are about the same as the ones for the medium sized hoses.

Table 7: *Correction values*

	Medium long hoses	Long hoses
8->2Nm^3/h	-1.3	-1.55
5->2Nm^3/h	-2.0	-0.7
2->8Nm^3/h	5.8	1.2
2->5Nm^3/h	2.7	0.4

What can these values be related to? By studying the expected change in T_{inj} steady state values at $T_{V/P}(t=0)$, caused by a change in fuel mass flow (*Table 8*), one might suspect that there is a correlation between this difference and the correction values. This is not the complete explanation. Sensor characteristics influence the shape of the temperature curve. These are unfortunately unknown.

Table 8: Changes of T_{inj} steady state value at steps in fuel mass flow

	Medium long hoses	Long hoses
8->2Nm^3/h	-15.2	-12.7
5->2Nm^3/h	-15.5	-9.8
2->8Nm^3/h	26.1	22
2->5Nm^3/h	18.4	12

The transient model, consisting of a first order filter, leaves a great deal to be desired. With a correction of the initial value, it can give a good correspondence with the reality. A more physically correct model would perhaps give better results, but it is both difficult to develop and probably much more time consuming in the software.

6.3 Temperature in the engine compartment

When air enters the engine compartment, it has ambient temperature. Because of the hot engine, the air heats up inside the engine compartment. The magnitude of the air draught depends on the speed of travel. From this we can draw some conclusions. The temperature in the engine compartment depends on coolant temperature, ambient temperature and speed of travel. When the car is going very fast, the temperature in the engine compartment is close to ambient temperature. On the other hand, when the car has been standing still for a while, the temperature will be close to coolant. The temperature will not be the same in the whole engine compartment. Some places are more protected from air draught than others, which will result in a higher temperature at those locations.

6.3.1 Measurements

Temperatures of the air in the engine compartment and coolant were measured at 30km/h, at 100km/h and when the car was standing still. Ambient temperature was also measured.

The sensor for measuring the temperature in the engine compartment was located at a place well exposed to draught. The reason for this was that the hoses in the experiment car were located in such a place.

6.3.2 Results

The results are shown in *Table 9*. The ambient temperature was 16^{o}C.

Table 9: Measurements for the temperature in the engine compartment

	Standing still	30km/h	100km/h
$T_{eng.c.}$	65^{o}C	33^{o}C	27^{o}C
T_{cool}	85^{o}C	80^{o}C	72^{o}C

6.3.3 Conclusions

It can be noticed that there is a big difference between a moving and a stationary car. This was expected. A decrease in temperature in the engine compartment caused by an increase in speed of travel is not confirmed by the measurements however. It is true that the temperature in the engine compartment decreased when the speed increased, but so did the coolant temperature[6].

6.4 Pressure

Pressure waves travel with the same velocity as the speed of sound. The connection parts have such shapes that the restrictions are minimal at maximum flows. A change of pressure at one place of the connection will thus change the pressure at another place almost instantly (within 100 ms). Hence, the pressure will be approximately the same in the whole connection at all times.

[6] The reason why the measured coolant temperature decreases with increasing speed, might be that the part where the temperature of the coolant is measured is exposed to draught.

6.5 Summary

$$\frac{dT_{inj}}{dt} = \frac{1}{\tau}\left(\left[T_{inj}\right]_{SS} - T_{inj}\right)$$

$$T_{inj}(t,[T_{inj}]_{SS}) = T'_{inj}(t) + f([T_{inj}]_{SS})$$

$$\left[T_{inj}\right]_{SS} = T_{eng.c.} - \left(T_{eng.c.} - T_{V/P}\right)\cdot e^{-\frac{1}{\dot{m}_{fuel}c_{p,gas}}\sum U_i A_i}$$

$$UA = \sum U_i A_i = 0.60 l_1 + 0.28 + 2.15 l_2$$

l_1 = length of hose between V/P and rail [m]
l_2 = length of hose between rail and injector [m]
$1/\tau$ is about $0.007 s^{-1}$ when $l_1 = 1.0$m and $l_2 = 0.5$m

$$p_{inj}(t) = p_{V/P}(t)$$

7 Implementation

In this chapter there is a discussion of how to implement the ECM algorithms based on the models in chapter *3-6*. The models are sometimes very complicated and would take a long CPU time to calculate in the ECM (*Figure 22*). Most calculations must thus be made on another computer and only the results, in form of look-up tables, will be implemented.

Things that are easy for the ECM to calculate are addition and subtraction of two variables. It is also easy to do a multiplication with a constant. Most other mathematical operations are time consuming, like multiplication/division between two variables or exponential expressions.

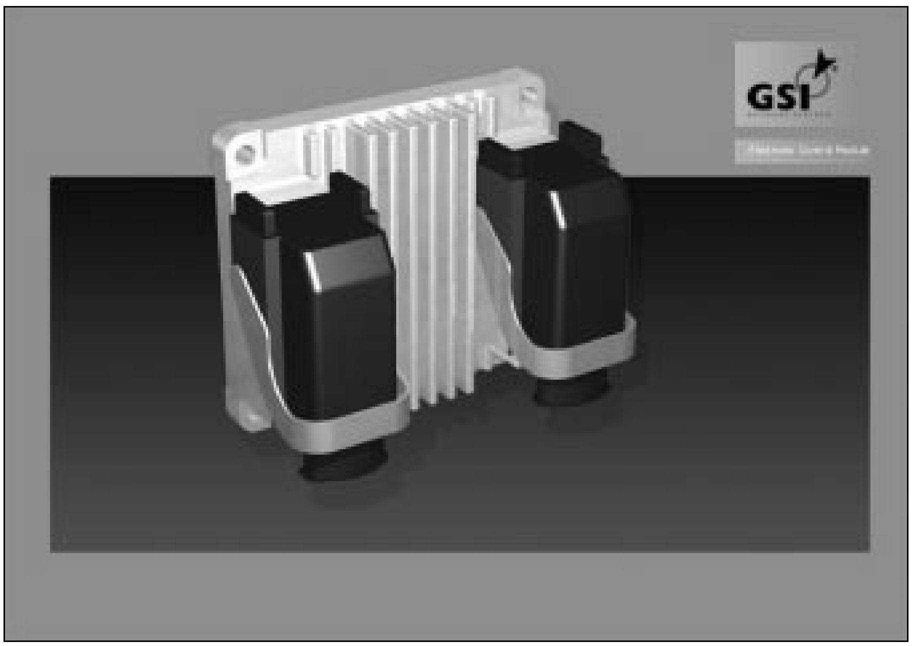

Figure 22: The ECM

Look-up tables can be designed in different ways. The values the look-up tables give can either be the values corresponding to the table inputs that are closest to the actual input, or interpolated ones. There are both two-dimensional and multi-dimensional tables, but it takes much longer time to look something up in a multi-dimensional table than in a two-dimensional one. Therefore, it is better to use a couple of two-dimensional tables than one multi-dimensional.

7.1 Air and fuel flow

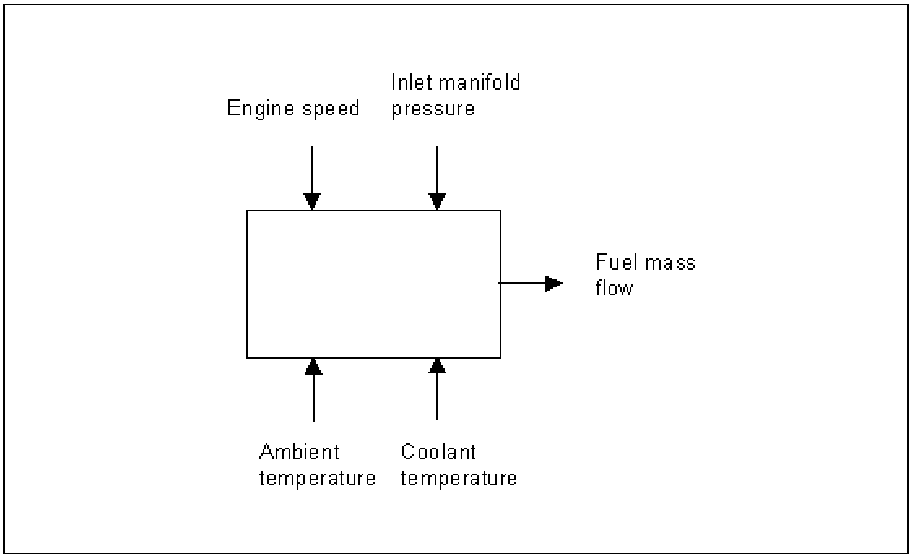

Figure 23: Flow chart for desired fuel mass flow

When calculating the desired amount of fuel, the following data is needed: coolant temperature, ambient temperature, engine speed and inlet manifold pressure (*Figure 23*). The calculations are done with the help of four look-up tables. The first one contains a map of the volumetric efficiency, according to

$$\eta_{vol} = \eta_{vol}(N, p_{im}) \tag{42}$$

The second look-up table takes m_{air} and $v = T_{amb} - T_{cool}$ as input, and gives

$$g_1(m_{air}, v) = f(m_{air}) \cdot v \tag{43}$$

as output. The current m_{air} is of course unknown, but the latest calculated airflow serves as a good approximation. The ambient temperature is determined according to (3).

The third table gives g_2 as a function of p_{im} and $T_{ch} = T_{cool} + g_1$

$$g_2(p_{im}, T_{ch}) = \frac{V_d n_{cyl} p_{im} N}{60 n_i R T_{ch}} \tag{44}$$

The resulting airflow can be read from the last table, using g_2 and η_{vol} as inputs.

$$m_{air}(\eta_{vol}, g_2) = \eta_{vol} \cdot g_2 \tag{45}$$

Since $\lambda = 1$, the fuel mass flow is

$$m_{fuel} = m_{air} \frac{1}{(m_{air} / m_{fuel})_S} \tag{46}$$

7.1.1 Table properties

The values used in the tables have the following ranges:

- N: 600 -> 6400rpm
- p_{im}: 10 -> 100kPa
- T_{amb}: 273 -> 308K (-10 -> 35^{o}C)

– T_{cool}: 273 -> 373K (0 -> 100^{o}C)

7.2 The tank

The temperature of the tank is determined according to (7).

7.3 Vaporiser/pressure regulator

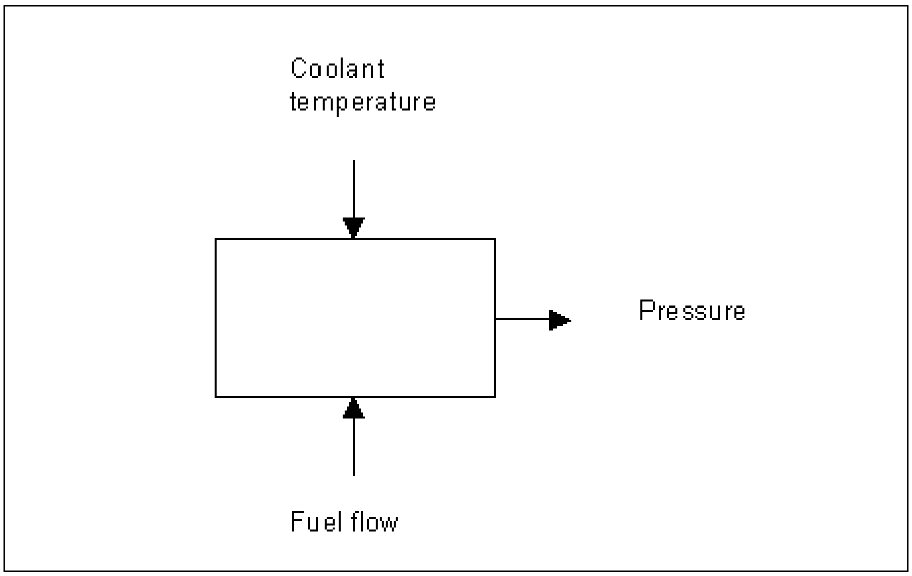

Figure 24: Flow chart for the V/P pressures

7.3.1 Pressure

The data needed to calculate the first and second stage pressures of the V/P are: coolant temperature, fuel mass flow and inlet manifold pressure (*Figure 24*). In *5.1* it was shown that p_{stage1} could be approximated by

$$p_{stage1}[kPa] = p_{im} + 142 - 3.3 \cdot 10^{3} \cdot m_{fuel} \tag{47}$$

and $p_{V/P}$ by

$$p_{V/P}[kPa] = p_{im} + 121 - 1.0 \cdot 10^3 \cdot m_{fuel} - 0.068 \cdot T_{cool} \tag{48}$$

(47) and (48) are simple expressions, which means that it will take hardly any time to evaluate them. They can be implemented without modification into the software.

7.3.2 Temperature

The data needed to calculate the temperature of the fuel after the vaporiser/pressure regulator are: coolant temperature, boiling temperature, tank temperature, first stage pressure, second stage pressure and fuel mass flow (*Figure 25*).

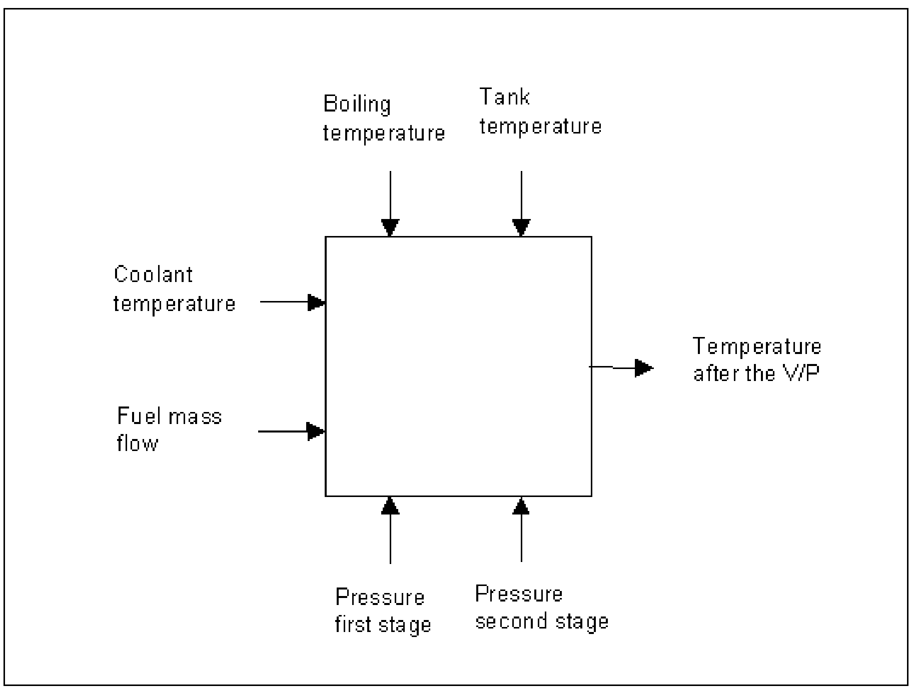

Figure 25: Flow chart for $T_{V/P}$

The temperature after the vaporiser/pressure regulator is (according to *5.3*)

$$\frac{dT_{V/P}}{dt} = \frac{1}{\tau} \cdot \left(\left[T_{V/P} \right]_{SS} - T_{V/P} \right) \tag{49}$$

This can be implemented as follows

$$\begin{aligned} T_{V/P}(t) &\approx T_{V/P}(t-t_s) + t_s \cdot \frac{dT_{V/P}(t-t_s)}{dt} = \\ &= T_{V/P}(t-t_s) + t_s \cdot \frac{1}{\tau} \left(\left[T_{V/P} \right]_{SS} - T_{V/P}(t-t_s) \right) \end{aligned} \tag{50}$$

The expressions for the temperature after the V/P at steady state (derived in *0*) are very complicated. It would take a long time to calculate them, too long to be of any use in a real time running algorithm. The solution is to use look-up tables.

The temperature has to be determined in several steps. The first table takes as input $v_1 = T_{stage1} - T_{boil}$ and $v_2 = T_{Tank} - T_{boil}$, and gives as an output a value called f_1.

$$f_1(v_1, v_2) = 1 - \frac{v_1}{v_1 + \frac{r - c_{p,liq} v_2}{c_{p,gas}}} \tag{51}$$

f_1 is used as input to the next table together with m_{fuel}. The output of the table is called f_2.

$$f_2(f_1, m_{fuel}) = e^{-\frac{UA}{\dot{m}_{fuel} c_{p,gas}} f_1} \tag{52}$$

The next table includes the multiplication between f_2 and $v_3 = T_{cool} - T_{boil}$. Multiplication of two variables takes a long time to calculate. That is why a look-up table is used.

$$f_3(f_2, v_3) = v_3 \cdot f_2 \tag{53}$$

The value in the last table represents the temperature at steady state. Its input variables is $T_{stage1} = T_{cool} - f_3$ and p_{im}.

$$[T_{V/P}]_{SS}(T_{stage1}, p_{im}) = \left(\frac{p_{im} + \Delta p_{stage1}}{p_{im} + \Delta p_{V/P}} \right)^{\frac{\kappa - 1}{\kappa}} \cdot T_{stage1} \tag{54}$$

Δp_{stage1} and $\Delta p_{V/P}$ are the mean differences between the inlet manifold pressure and $p_{1:st\ stage}$ respective $p_{V/P}$. According to the measurements, these values are 132kPa and 95kPa. The error caused by this approximation is less than 1.5^{o}C.

7.3.2.1 Table properties

The ranges for the temperatures are

– T_{boil}: 243 -> 263K (-30 -> -10^{o}C)

– T_{cool}: 273 -> 373K (0 -> 100^{o}C)

– T_{Tank}: 273 -> 308K (0 -> 35^{o}C)

– T_{stage1}: 243 -> 368K (-30 -> 95^{o}C)

which means that v_1 to v_3 are

– v_1: -20 -> 125K

– v_2: 10 -> 65K

– v_3: 10 -> 130K

The remaining inputs have the following ranges:

- p_{im}: 10 -> 100kPa

- m_{fuel} : 0 -> 6.4g/s (0 -> 10Nm^3/h)

f_1 does not have to be very accurate. A 4x4 look up table where the output is the value given by the table inputs closest to the actual input values might be good enough. The outputs from the other three tables must be more accurate. This can be achieved by a linear interpolation or more points in the tables.

7.4 Connection between the V/P and the rail

7.4.1 Temperature

The inputs to the temperature algorithm are: temperature in the engine compartment, temperature after the vaporiser/pressure regulator and fuel mass flow (*Figure 26*).

According to *6.2*, the temperature at the injectors can be estimated by

$$\frac{dT_{inj}}{dt} = \frac{1}{\tau} \cdot \left(\left[T_{inj} \right]_{SS} - T_{inj} \right) \tag{55}$$

This can be implemented in the following way

$$T_{inj}(t) \approx T_{inj}(t-t_s) + t_s \cdot \frac{dT_{inj}(t-t_s)}{dt} =$$
$$= T_{inj}(t-t_s) + t_s \cdot \frac{1}{\tau}\left(\left[T_{inj}\right]_{SS} - T_{inj}(t-t_s)\right) \tag{56}$$

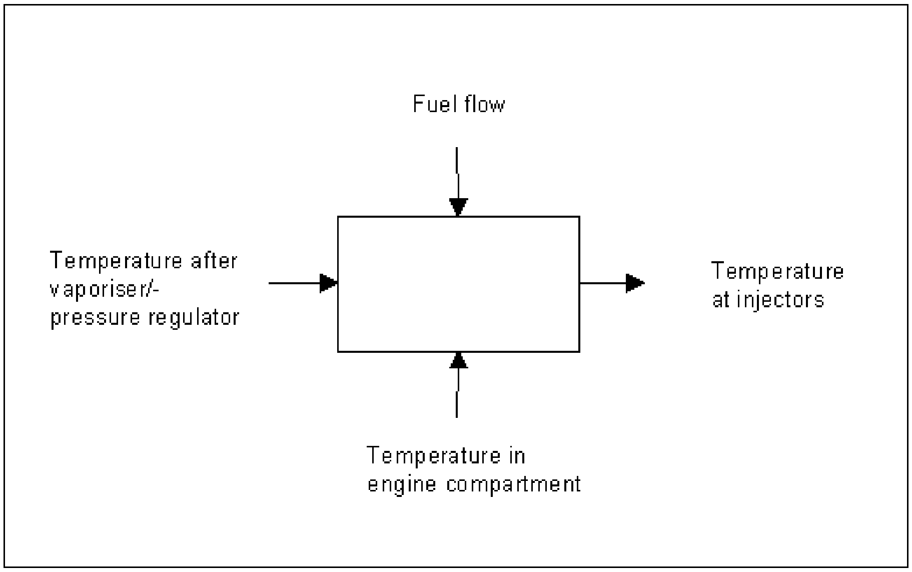

Figure 26: *Flow chart for T_{inj}*

To make this model work properly, an offset has to be added. This is done by first determining the offset

$$f = f([T_{inj}]_{SS}) \tag{57}$$

and then adjusting T_{inj}

$$T_{inj}(t) := T_{inj}(t) + f([T_{inj}]_{SS}(t)) - f([T_{inj}]_{SS}(t-t_s)) \tag{58}$$

$[T_{inj}]_{SS}$ can be calculated with the help of a look-up table, which gives g as output and takes $v = T_{eng.c.} - T_{V/P}$ and m_{fuel} as input.

$$g(v, m_{fuel}) = v \cdot e^{-\frac{UA}{\dot{m}_{fuel} c_{p,gas}}} \tag{59}$$

$[T_{inj}]_{SS}$ is then given by a simple subtraction

$$[T_{inj}]_{SS} = T_{eng.c.} - g \tag{60}$$

7.4.1.1 Table properties

The values used in the tables have the following ranges:

- m_{fuel} : 0 -> 6.4g/s (0 -> 10Nm^3/h)

- v: -85 -> 75K, since $T_{eng.c.}$: 283 -> 348K and $T_{V/P}$: 273 -> 368K

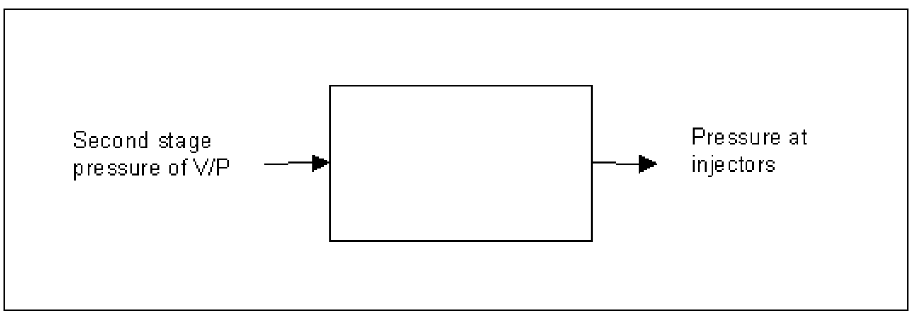

Figure 27: Flow chart for p_{inj}

7.4.2 Pressure

The pressure at the injectors is the same (approximately) as just after the vaporiser, i.e.

$$p_{inj} = p_{V/P} \tag{61}$$

8 Factors that might effect the system

A model can never describe the reality exactly. It is usually wise to keep models as simple as possible. Thus, a lot of factors that affect the modelled system should be ignored as long as their impact is not significant.

Below, there is a discussion about factors that have not been taken into account when deriving the models, but still might have an effect on the system behaviour.

Radiation – The exhaust manifold, cooling system and other hot parts radiate heat to the colder parts of the system. This can have consequences for the temperatures of the GSI.

Installation – The GSI components are placed where there is enough space to put them. The actual location depends on the type of car that is used. In some cars the hoses might end up close to the usually very hot exhaust manifold, while in other cars they are placed relatively far from the heat sources. This will have effect on the temperature at the injectors, because the temperature of the air surrounding the hoses will be different.

Fuel type – The ratio between propane and butane is not always 60/40, as assumed in this thesis. A different fuel content will affect a lot of parameters, for example the density, lambda and thermal properties. This will have consequences for the temperatures in the system, the amount of fuel injected as well as the lambda control. In *Figure 28* is shown how the temperature after the V/P varies depending on fuel type. As can be seen, the difference is 5 degrees or less between two fuels with very unequal propane/butane ratios (70/30 and 40/60).

Pressure oscillations – There are some pressure oscillations in the connection between the V/P and the injectors. These are caused by the opening and closing of valves. The magnitude is about 10kPa. The pressure oscillations will reduce the accuracy of the pressure calculations, and thus the density of the injected fuel.

Air contents – Variations in air composition might be suspected to be of some importance for the air/fuel equivalence ratio. However, in the calculations of the stoichiometric proportions of air and fuel, oxygen and nitrogen are used. These are nearly constant at heights lower than 90 kilometres (less than 1%). [8]

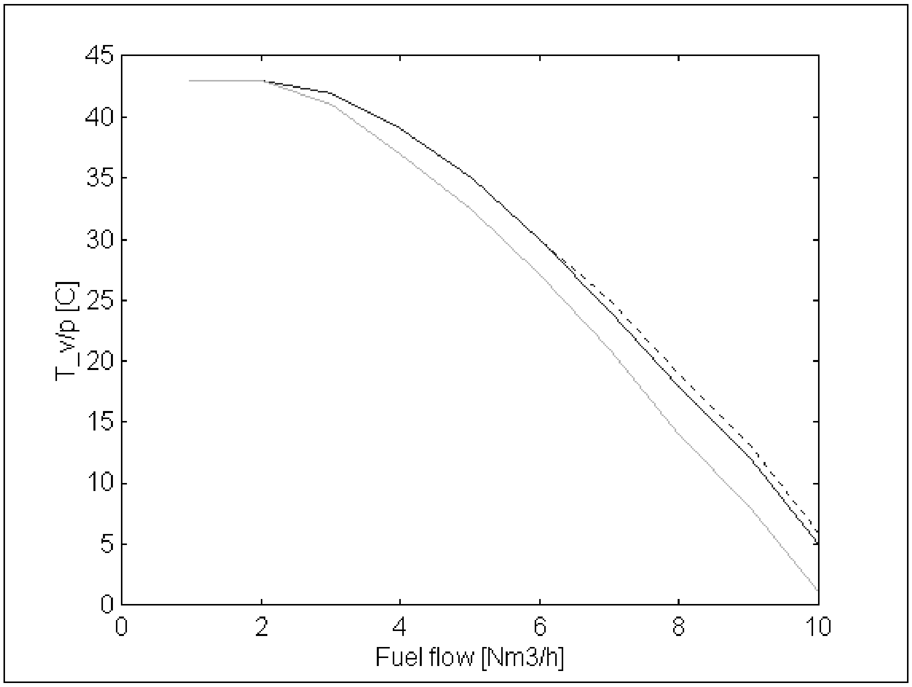

Figure 28: *Simulated $T_{V/P}$ for three different fuel types at T_{cool} = 50°C – dotted line: 70% propane/30% butane, dark solid line: 60% propane/40% butane, light solid line: 40% propane/60% butane*

Ambient pressure – The barometric pressure decreases with increasing height. The decrease is about 350Pa per 30 metres near the surface of the earth [8]. That means that on a height of 1000 metres, the pressure has dropped about 12 percent. This affects the airflow into the engine. A change in pressure can be identified, because at full throttle the inlet manifold pressure is the same as the ambient pressure.

Engine wear - The quality of an engine will change during its lifetime due to wear. What consequences this has depend on how the engine changes.

Differences between hoses – When the system is implemented in a car, the hoses between the rail and the injectors are usually not of equal length. That means that the injection temperature will differ from one injector to another. The position of the hose at the rail (close to or far away from the hose connecting the V/P to the rail) will also be of some importance.

In *Figure 29* is shown the injection temperature at steady state simulated for two different lengths of the connection. The biggest deviation between the two lines is almost 10^oC. Since it is not possible to control the injectors individually, some problems might occur if the length of the hoses between the rail and injectors vary much.

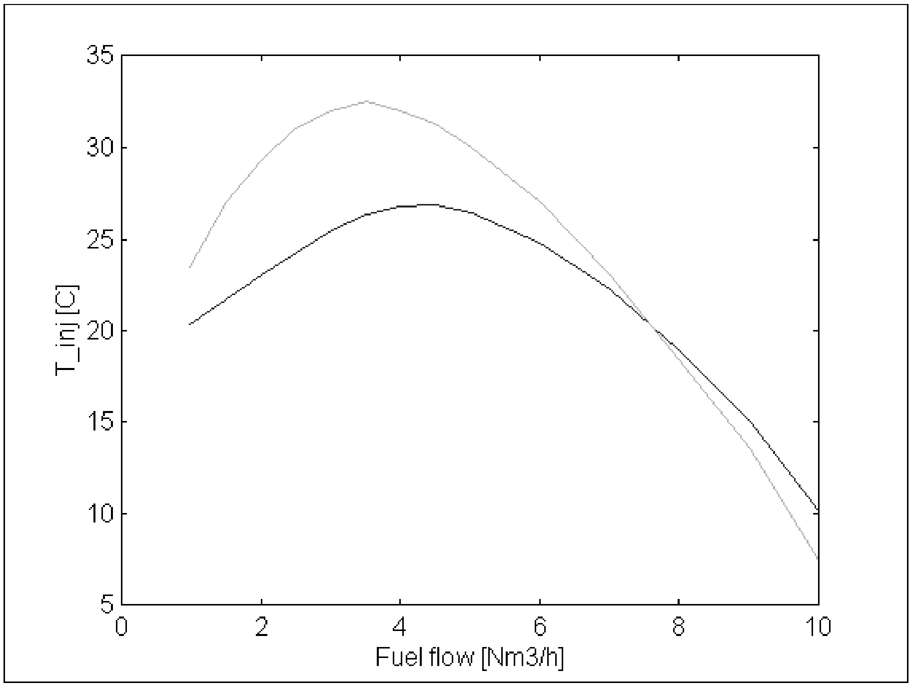

Figure 29: *Simulation of T_{inj} for two different lengths of the connection. The darker line shows the values for when the hose between the rail and injector is 1.5m and connected to the part of the rail that is as far as possible from the hose between the V/P and the rail. The lighter line is when the hose is 0.5m and connected to the opposite side of the rail. ($T_{Tank} = 5^oC$ and $T_{eng.c} = 20^oC$)*

9 Conclusions and recommendations

The derived models describe the measured data well, especially the steady state models. However, to make the mathematical description of the GSI complete, there are some things that need to be described. There are also some things that can be done to improve the accuracy of the models.

The derived models can be transformed into ECM algorithms and can be run in real-time.

As long as the injectors are made of metal, the temperature of the fuel after the injectors will be highly correlated to the temperature of the inlet manifold. The fuel temperature changes a lot over the injectors, which makes it very difficult to determine the fuel temperature exactly at the injection valve. It is better in a modelling point of view to use plastic injectors. The injection temperature will then be about the same as the fuel temperature just before the injectors.

Differences in hose lengths of the hoses connecting the rail to the injectors will result in different injection temperatures for each injector. Not only the steady-state values will be unequal; there will also be a difference in time constants. It is recommended to keep the hoses of as equal lengths as possible.

If short hoses are used, the errors in predicted temperatures caused by air draught and uncertain engine compartment temperature will be smaller then when long hoses are used. On the other hand, the shorter the hoses are, the bigger the relative impact of the hose position at the rail is. In this aspect, a rail made of plastic or rubber is better then a rail made of metal.

The fuel content (the proportion between propane and butane) does not matter much for the injection temperature.

There are some things that can go very wrong. One example is when the tank temperature is estimated. Therefore, the ECM has to use the information given by the lambda sensor and do temporary corrections to the algorithms. Permanent changes, for example system wear, have to be learned in the software.

9.1 Things needed to complete the description of the system

Injector model – The behaviour of the new injectors has to be described.

Minimum temperature – The LPG did not vaporise completely, even if its temperature was much higher than the boiling temperature. The lowest possible fuel temperature, for which the fuel still becomes fully vaporised, must be determined.

Transient behaviour of T_{inj} for different hose lengths – With longer hoses, it takes longer time for T_{inj} to stabilise after a change in $T_{V/P}$. To make it possible to use different hose lengths, the relationship between the time constant and hose lengths has to be described. It may be wise to include fuel mass flow in the model, since it seems like it has a big influence on the time constant when long hoses are used. (*6.2*)

Offset for the transient behaviour of T_{inj} – The offset as function of $[T_{inj}(T_{V/P})]_{SS}$ has to be determined. A look up-table might be an appropriate way to do this, because the physical background to the offset factor is complex. (*6.2*)

Temperature in the engine compartment – The temperature in the engine compartment has to be described, as it is of importance for the temperature at the injectors. Either a function given by regression analysis or a look-up table may be good ways to do this. It is important to study the transient behaviour, because the changes are relatively slow. (*6.3*)

Transient behaviour of p_{stage1} and $p_{V/P}$ - The pressures in the V/P changes relatively fast, but it will still take a couple of engine cycles before the pressures have reached their steady state values. A model of the V/P transient behaviour would result in a better estimation of the V/P pressures. (*4*)

Relation between p_{im} and p_{stage1} or $p_{V/P}$ - With decreasing pressures, the density decreases. This will effect the performance of the V/P. Thus, the relation is not linear.

9.2 Recommended improvements

Sensor behaviour – The influences the sensors have on the measured data have not been compensated for when the different parameters were calculated. To make the models more accurate and reliable, the behaviour of the sensors have to be studied and compensated for.

$T_{V/P}$ response after a step in inlet manifold pressure – A step in fuel mass flow is usually a result of a change in inlet manifold pressure. $T_{V/P}$ is depending on the fuel mass flow and indirectly on the inlet manifold pressure, due to that the pressure ratio $p_{stage1}/p_{V/P}$ is included in the model. It would thus give a more reliable model if not only responses for steps in fuel flow are studied, but steps in both inlet manifold temperature and fuel flow.

Time constants as function of fuel flow - It seems like the time constants used in the transient temperature models are depending on the fuel flow. The physical background to this can be well worth studying. It might both give some valuable knowledge as well as help improving the transient models. (*6.2* and *5.3*)

Ambient temperature model – Under unfortunate circumstances, the estimated ambient temperature can be as much as 20 degrees off. Assume that it is a very nice and sunny day and the temperature is about 30 ^{o}C. If the coolant temperature has not had time to cool down enough since it was last used, the coolant temperature might be above T_{max} = 35 ^{o}C. T_{amb} is thus set to 10^{o}C. Another example of when the estimation can give wrong values, is when the car has been standing in a garage when started.

Errors in the estimated ambient temperature have effect on the calculation of air and fuel mass flows as well as the temperature in the engine compartment. (*3.1* and *6.3*)

Tank temperature model – The model for the tank temperature is made in the same way as the one for ambient temperature, which means that it causes the same kind of problems. (*6*)

Air draught – Apart from conductive heat transfer, there is also convective heat transfer between the GSI parts and its surrounding. The major part of the convective heat transfer is caused by the air draught through the engine compartment. This increases with increasing speed. The part that is most sensitive to air draught is the connection. It has a big area for a not so big volume.

The air draught depends on the speed of travel, but it is difficult to get good information about the speed. One way to compensate for the air draught is to simply add a correction factor to the *UA* value of the connection when the car is moving.

10 References

1 Heat transfer, J P Holman

2 Physics handbook, Carl Nordling and Jonny Österman

3 Course material Vehicular Systems, Lars Nielsen and Lars Eriksson

4 Electronic Gas Injection, KOLTEC

5 Alternative Fuels Guidebook, Richard L. Bechtold

6 Grundläggande regressionsanalys, Eva Enqvist

7 Landi den Hartog b.v.

8 Encyclopedia Brittanica

Appendix A Used symbols and units

A-1 Sensor signals

T_{cool} Coolant temperature [K]
V_{supp} Voltage of the car battery [V]
N Engine speed [rpm]
P_{im} Inlet manifold pressure [Pa]
λ Air/fuel equivalence ratio = $\dfrac{m_{air} / m_{fuel}}{\left(m_{air} / m_{fuel}\right)_S}$ (S = stoichiometric)

A-2 Other symbols

T_{inj} Fuel temperature at the injectors [K]
$T_{V/P}$ Fuel temperature just after the vaporiser/pressure regulator [K]
T_{Tank} Fuel temperature in the tank [K]
$T_{surround}$ Temperature of the air surrounding the connection [K]
T_{amb} Temperature outside the car [K]
T_{boil} Boiling temperature [K]

p_{stage1} Pressure in the V/P first stage chamber [Pa]
$p_{V/P}$ Pressure in the V/P second stage chamber [Pa]
p_{inj} Pressure at the injectors [Pa]

$\dot{m}$ Mass flow [kg/s]
t_{key_off} The amount of time the engine has been turned off

$[\ ]_{SS}$ Steady state value

A-3 Abbreviations

GSI Gaseous sequential injection system
LPG Liquefied petroleum gas
V/P Vaporiser/pressure regulator

ECM	Electronic control module

Appendix B LPG properties

The LPG used in this study consisted of 60% propane and 40% butane. 25% of the butane was i-butane, the rest was n-butane.

Table 10: Properties [7]

Constants	Propane	i-butane	n-butane
R [J/kgK]	188.53	143.1	143.1
$c_{p,liq}$ [kJ/kgK]	2.76	2.35	2.39
$c_{p,gas}$ [kJ/kgK]	1.63	1.57	1.59
κ [-]	1.13	1.11	1.11
ρ_{gas} [kg/m^3]	2.011	2.697	2.708
r [kJ/kg]	427.0	366.4	385.6

Appendix C Steady state measurements for the connection

Table 11: Measurement 1a

	m_{fuel} = 2Nm³/h	m_{fuel} = 4Nm³/h	m_{fuel} = 6Nm³/h
m_{fuel} [Nm³/h]	2.00	4.00	5.99
T_{cool} [°C]	91.7	90.6	84.2
$T_{V/P}$ [°C]	81.1	75.7	61.3
$T_{b.rail}$ [°C]	60.1	64.5	55.7
$T_{a.rail}$ [°C]	51.6	60.5	54.3
$T_{b.inj}$ [°C]	36.4	50.0	48.6
$T_{a.inj}$ [°C]	30.7	43.7	45.7

Sensor on hose closest to hose connecting the V/P with the rail

Hose length v/p->rail: 1.0m

Hose length rail->injectors: 0.5m

Ambient temperature = 14°C

Table 12: Measurement 1b

	m_{fuel} = 2Nm³/h	m_{fuel} = 4Nm³/h	m_{fuel} = 6Nm³/h
m_{fuel} [Nm³/h]	2.00	4.00	5.95
T_{cool} [°C]	91.5	90.0	85.9
$T_{V/P}$ [°C]	80.8	75.1	63.1
$T_{b.rail}$ [°C]	58.7	63.3	57.8
$T_{a.rail}$ [°C]	43.6	55.6	55.2
$T_{b.inj}$ [°C]	33.1	47.3	50.4
$T_{a.inj}$ [°C]	28.9	43.1	47.5

Sensor on hose most farthest from hose connecting the V/P with the rail

Hose length v/p->rail: 1.0m

Hose length rail->injectors: 0.5m

Ambient temperature = 14°C

Table 13: Measurement 2

	m_{fuel} = 2Nm^3/h	m_{fuel} = 4Nm^3/h	m_{fuel} = 6Nm^3/h
m_{fuel} [Nm^3/h]	2.08	3.99	6.03
T_{cool} [^{o}C]	83.7	81.3	80.4
$T_{V/P}$ [^{o}C]	70.7	67.4	60.0
$T_{b.rail}$ [^{o}C]	45.6	55.5	53.7
$T_{a.rail}$ [^{o}C]	32.2	46.4	49.6
$T_{b.inj}$ [^{o}C]	23.0	32.8	39.0
$T_{a.inj}$ [^{o}C]	21.0	23.1	26.3

Sensor on hose farthest from hose connecting the V/P with the rail

Hose length v/p->rail: 1.5m

Hose length rail->injectors: 0.97m

Ambient temperature = 14^{o}C

Table 14: Measurement 3

	m_{fuel} = 2Nm^3/h	m_{fuel} = 4Nm^3/h	m_{fuel} = 6Nm^3/h
m_{fuel} [Nm^3/h]	1.99	4.01	6.04
T_{cool} [^{o}C]	81.7	80.5	80.6
$T_{V/P}$ [^{o}C]	75.9	67.1	58.9
$T_{b.rail}$ [^{o}C]	56.6	59.9	55.8
$T_{a.rail}$ [^{o}C]	43.8	53.8	53.8
$T_{b.inj}$ [^{o}C]	34.8	47.0	50.1
$T_{a.inj}$ [^{o}C]	22.4	25.3	29.1

Sensor on hose farthest from hose connecting the V/P with the rail

Hose length v/p->rail: 0.5m

Hose length rail->injectors: 0.3m

Ambient temperature = 14^{o}C

Table 15: Measurement 4

	m_{fuel} = 2Nm^3/h	m_{fuel} = 4Nm^3/h	m_{fuel} = 6Nm^3/h
m_{fuel} [Nm^3/h]	2.03	4.00	6.00
T_{cool} [^{o}C]	81.3	81.5	81.6
$T_{V/P}$ [^{o}C]	67.7	64.1	56.1
$T_{b.rail}$ [^{o}C]	47.2	52.9	51.0
$T_{a.rail}$ [^{o}C]	33.8	42.7	46.5
$T_{b.inj}$ [^{o}C]	21.8	31.1	38.3
$T_{a.inj}$ [^{o}C]	16.6	16.7	17.4

Sensor on hose farthest from hose connecting the V/P with the rail

Hose length v/p->rail: 1.0m

Hose length rail->injectors: 0.75m

Ambient temperature = 14^{o}C

Table 16: Measurement 5

	m_{fuel} = 2Nm^3/h	m_{fuel} = 4Nm^3/h	m_{fuel} = 6Nm^3/h
m_{fuel} [Nm^3/h]	2.00	4.00	6.01
T_{cool} [^{o}C]	81.5	82.3	80.7
$T_{V/P}$ [^{o}C]	70.1	65.0	55.5
$T_{b.rail}$ [^{o}C]	37.2	44.8	44.9
$T_{a.rail}$ [^{o}C]	29.9	37.9	41.8
$T_{b.inj}$ [^{o}C]	17.3	20.7	27.0
$T_{a.inj}$ [^{o}C]	14.1	13.5	14.3

Sensor on hose farthest from hose connecting the V/P with the rail

Hose length v/p->rail: 2.57m

Hose length rail->injectors: 2.00m

Ambient temperature = 14^{o}C

Appendix D Transient behaviour of the connection, measurements and modelling

The following graphs show (top to bottom): $[T_{inj}]_{SS}$ calculated from the measured temperature of the vaporiser/pressure regulator, calculated T_{inj} (including transient behaviour), measured T_{inj} and difference between calculated and measured T_{inj}. The time constants used are mentioned in the figure texts.

The hoses were 1.0 and 0.5m (the hose between the V/P and the rail mentioned first) when the measurements were done.

The important outcome of this study is a first order filter time constant that gives a good correspondance with the measured data. To avoid that the first quick change (of both known and unknown origin) has an effect on the choice of time constant, the initial value of the calculated T_{inj} is set so that the calculated and measured values are the same when the quick change fades out.

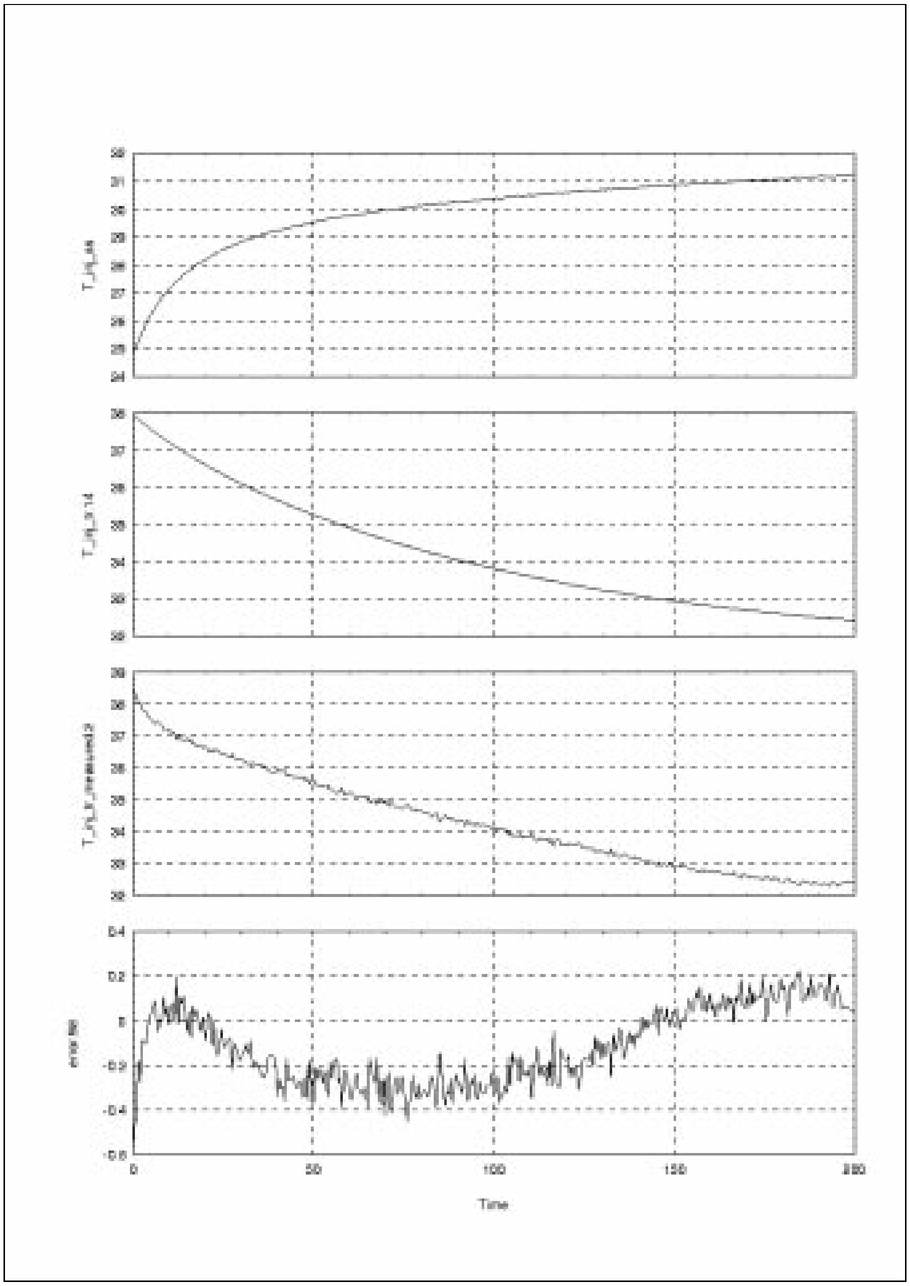

Figure 30: *Step in fuel mass flow 8->2Nm^3/h. $1/\tau = 0.009s^{-1}$.*

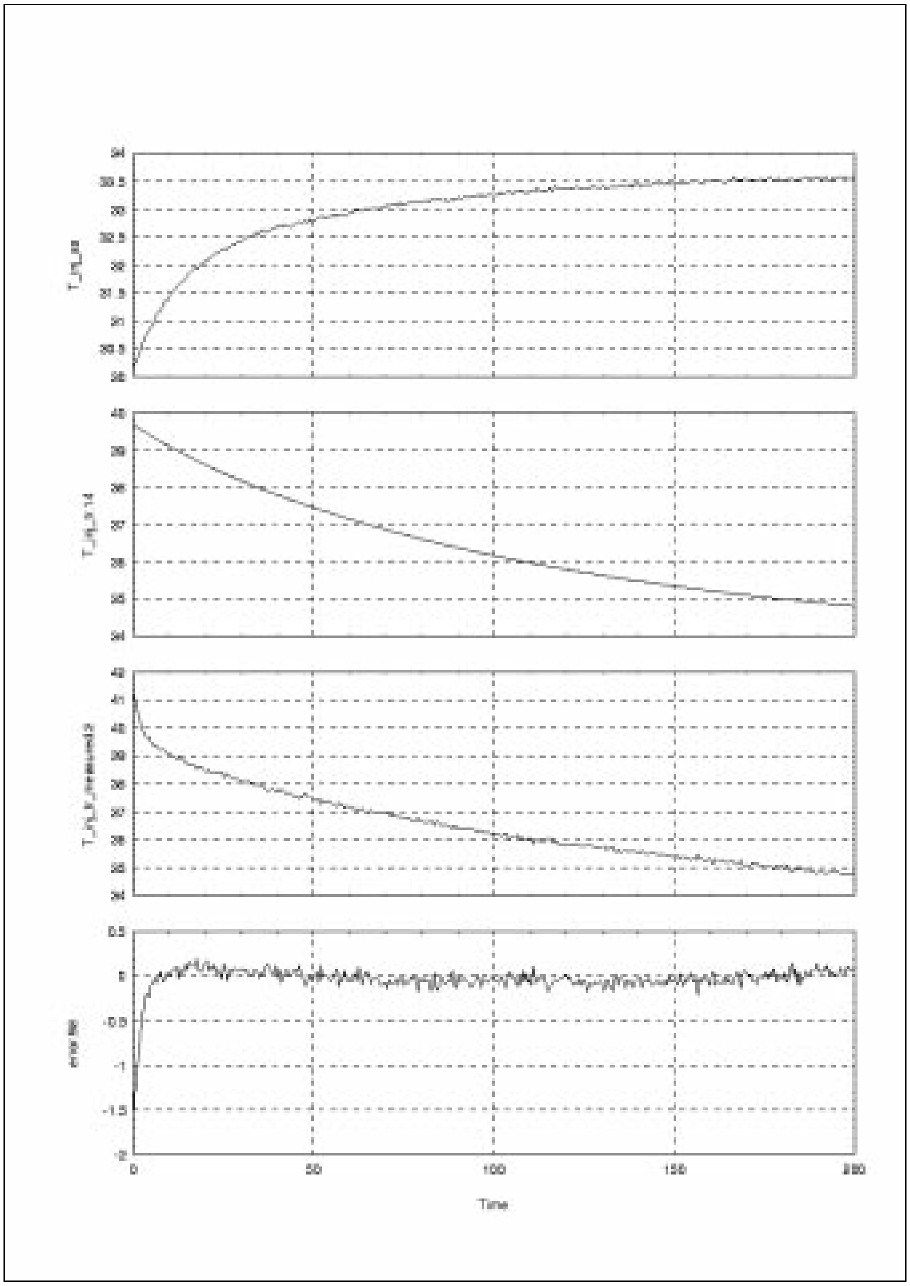

Figure 31: *Step in fuel mass flow 5->2Nm^3/h. 1/τ = 0.007s^{-1}.*

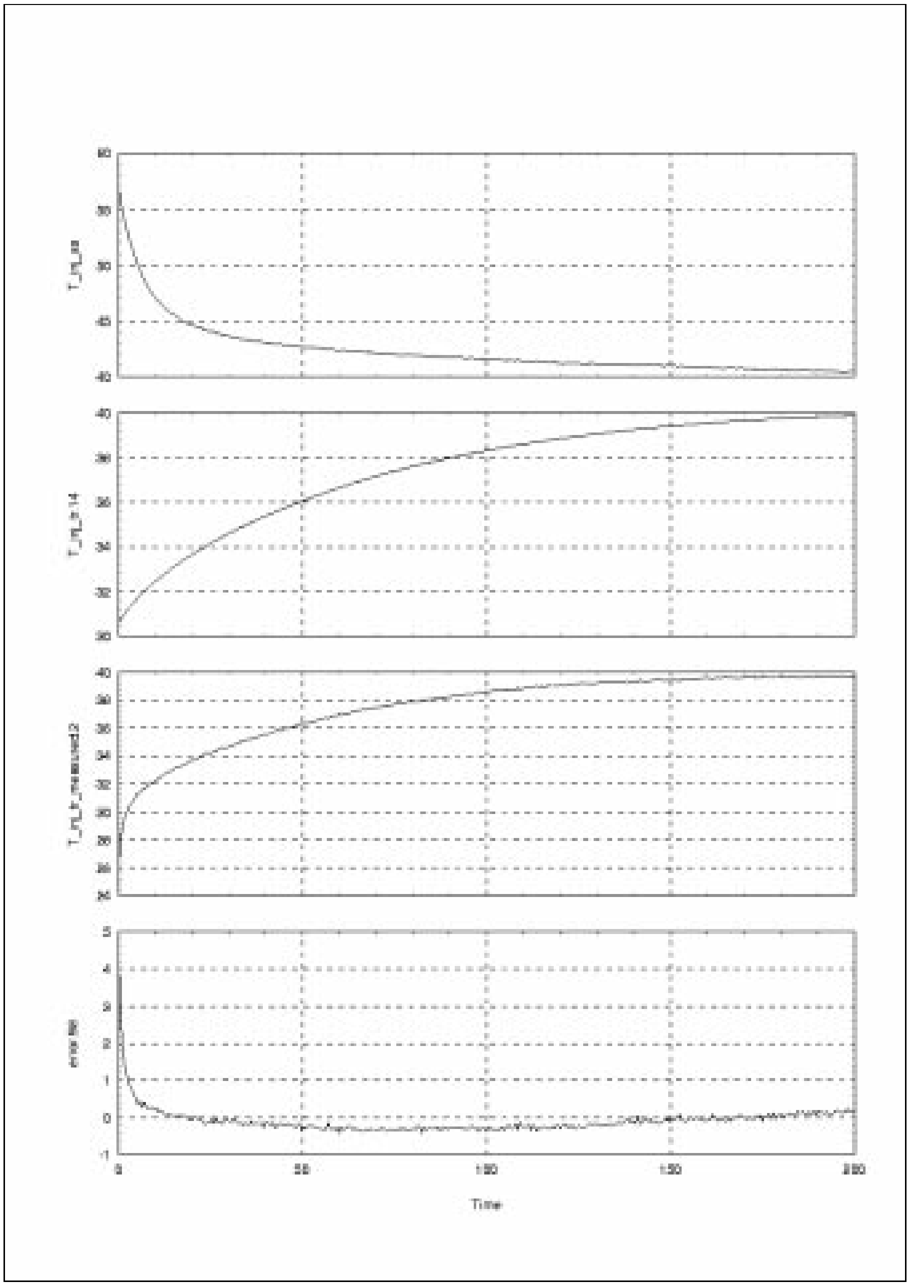

Figure 32: *Step in fuel mass flow 2->8Nm^3/h. 1/τ = 0.0095s^{-1}.*

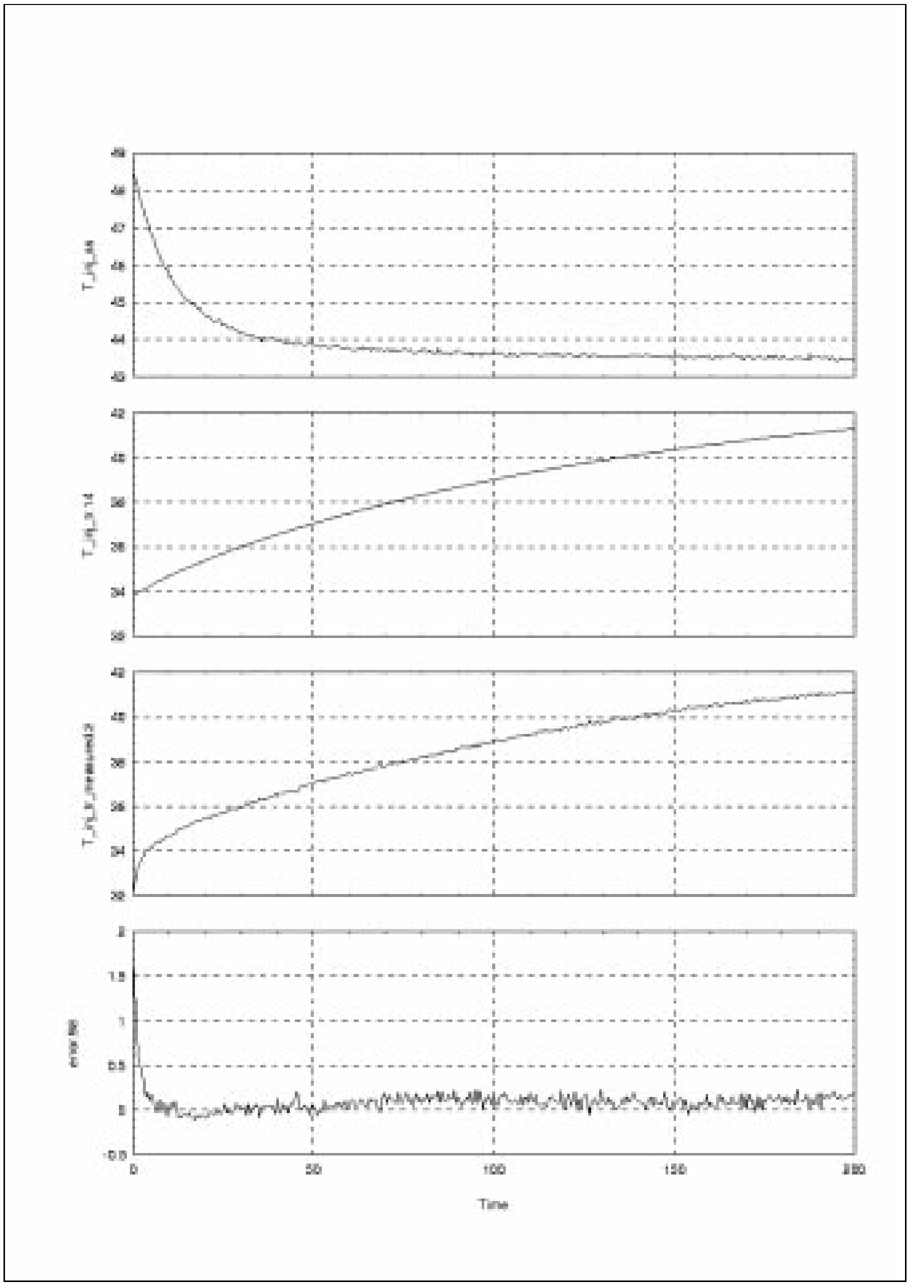

Figure 33: *Step in fuel mass flow2->5Nm^3/h. 1/τ = 0.007s^{-1}.*

Appendix E The log mean temperature difference

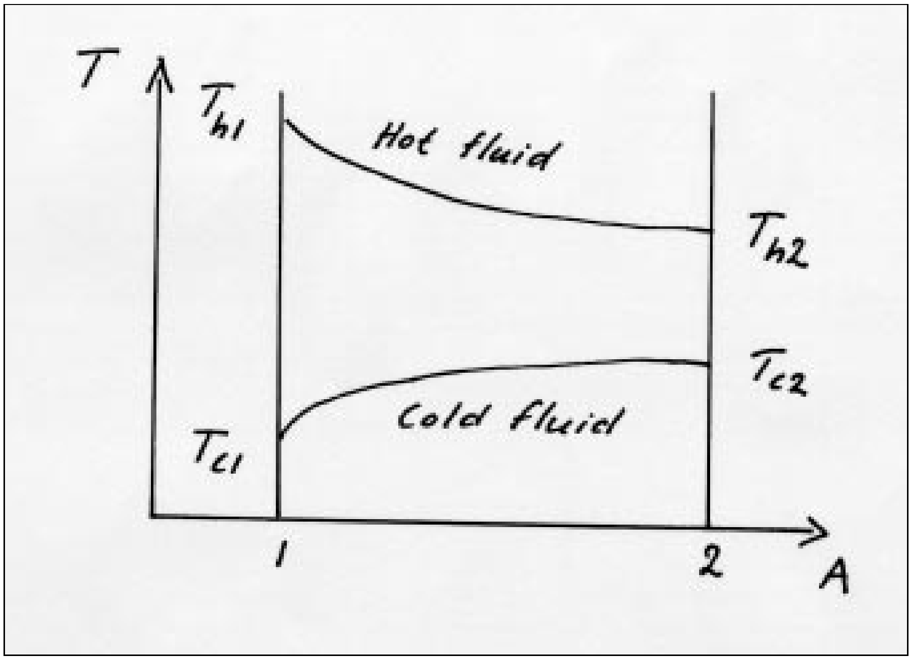

Figure 34: Temperature profiles

It can be shown [1] that the mean temperature difference in a heat exchanger is

$$\Delta T = \frac{(T_{h2} - T_{c2}) - (T_{h1} - T_{c1})}{\ln[(T_{h2} - T_{c2})/(T_{h1} - T_{c1})]} \tag{62}$$

Appendix F Measurements for T_v/p

Table 17: $T_{V/P}$ *[^oC] (mass flow: 1-5Nm^3/h)*

	m_{fuel} = 1Nm^3/h	m_{fuel} = 3Nm^3/h	m_{fuel} = 5Nm^3/h
T_{cool} = 20^{o}C	17	L	L
T_{cool} = 30^{o}C	27	20	L
T_{cool} = 40^{o}C	37	32	26
T_{cool} = 50^{o}C	45	41	35
T_{cool} = 60^{o}C	55	53	46
T_{cool} = 70^{o}C	64	62	57
T_{cool} = 80^{o}C	75	69	63
T_{cool} = 90^{o}C	85	82	75

Table 18: $T_{V/P}$ *[^oC] (mass flow: 6-10Nm^3/h)*

	m_{fuel} = 1Nm^3/h	m_{fuel} = 3Nm^3/h	m_{fuel} = 5Nm^3/h
T_{cool} = 20^{o}C	L	L	L
T_{cool} = 30^{o}C	L	L	L
T_{cool} = 40^{o}C	19	0	L
T_{cool} = 50^{o}C	31	20	4
T_{cool} = 60^{o}C	42	35	26
T_{cool} = 70^{o}C	52	45	37
T_{cool} = 80^{o}C	59	52	44
T_{cool} = 90^{o}C	67	60	54

L = Liquid

Appendix G UA values

Table 19: UA [W/K] (mass flow: 1-5Nm^3/h)

	m_{fuel} = 1Nm^3/h	m_{fuel} = 3Nm^3/h	m_{fuel} = 5Nm^3/h
T_{cool} = 20°C	-	L	L
T_{cool} = 30°C	-	48	L
T_{cool} = 40°C	-	54	56
T_{cool} = 50°C	-	45	51
T_{cool} = 60°C	-	63	50
T_{cool} = 70°C	-	49	51
T_{cool} = 80°C	-	34	41
T_{cool} = 90°C	-	51	44

Table 20: UA [W/K] (mass flow: 6-10Nm^3/h)

	m_{fuel} = 6Nm^3/h	m_{fuel} = 8Nm^3/h	m_{fuel} = 10Nm^3/h
T_{cool} = 20°C	L	L	L
T_{cool} = 30°C	L	L	L
T_{cool} = 40°C	52	45	L
T_{cool} = 50°C	52	52	49
T_{cool} = 60°C	51	56	57
T_{cool} = 70°C	49	54	56
T_{cool} = 80°C	44	49	52
T_{cool} = 90°C	40	46	51

Appendix H Measurements for the V/P pressures

Table 21: p_{stage1} *[kPa] (mass flow: 1-5Nm^3/h)*

	$\dot{m}_{fuel}$ = 1Nm^3/h	$\dot{m}_{fuel}$ = 3Nm^3/h	$\dot{m}_{fuel}$ = 5Nm^3/h
T_{cool} = 20°C	242	L	L
T_{cool} = 30°C	237	234	L
T_{cool} = 40°C	242	237	232
T_{cool} = 50°C	240	240	235
T_{cool} = 60°C	239	232	231
T_{cool} = 70°C	238	237	231
T_{cool} = 80°C	242	235	230
T_{cool} = 90°C	241	234	235

Table 22: p_{stage1} *[kPa] (mass flow: 6-10Nm^3/h)*

	$\dot{m}_{fuel}$ = 6Nm^3/h	$\dot{m}_{fuel}$ = 8Nm^3/h	$\dot{m}_{fuel}$ = 10Nm^3/h
T_{cool} = 20°C	L	L	L
T_{cool} = 30°C	L	L	L
T_{cool} = 40°C	230	228	L
T_{cool} = 50°C	231	226	224
T_{cool} = 60°C	229	225	220
T_{cool} = 70°C	231	226	221
T_{cool} = 80°C	228	225	220
T_{cool} = 90°C	230	225	220

Table 23: $p_{V/P}$ [kPa] (mass flow: 1-5Nm^3/h)

	$\dot{m}_{fuel}$ = 1Nm^3/h	$\dot{m}_{fuel}$ = 3Nm^3/h	$\dot{m}_{fuel}$ = 5Nm^3/h
T_{cool} = 20°C	199	L	L
T_{cool} = 30°C	198	199	L
T_{cool} = 40°C	198	197	196
T_{cool} = 50°C	198	196	195
T_{cool} = 60°C	198	197	195
T_{cool} = 70°C	197	195	195
T_{cool} = 80°C	195	195	195
T_{cool} = 90°C	193	195	194

Table 24: $p_{V/P}$ [kPa] (mass flow: 6-10Nm^3/h)

	$\dot{m}_{fuel}$ = 6Nm^3/h	$\dot{m}_{fuel}$ = 8Nm^3/h	$\dot{m}_{fuel}$ = 10Nm^3/h
T_{cool} = 20°C	L	L	L
T_{cool} = 30°C	L	L	L
T_{cool} = 40°C	196	195	L
T_{cool} = 50°C	194	193	193
T_{cool} = 60°C	195	192	190
T_{cool} = 70°C	195	192	190
T_{cool} = 80°C	193	191	189
T_{cool} = 90°C	193	191	189

Appendix I Transient measurements of the temperature after the V/P

The measurements shown were made for one combination of hose lengths; 1m and 0.5m (the one between the V/P and the connection mentioned first).

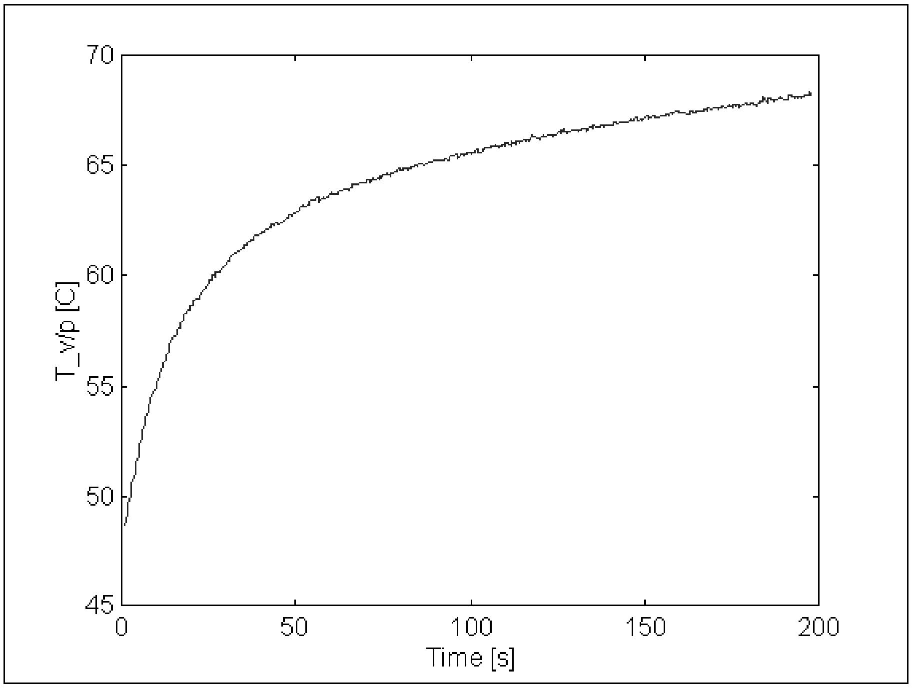

Figure 35: *Step response: 8->2Nm^3/h*

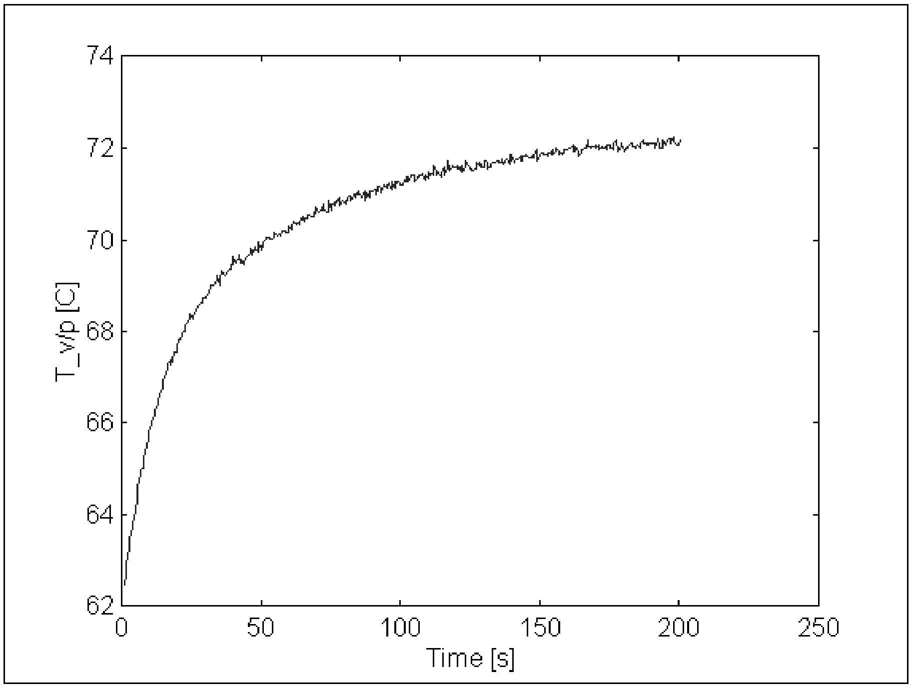

Figure 36: *Step response: 5->2Nm3/h*

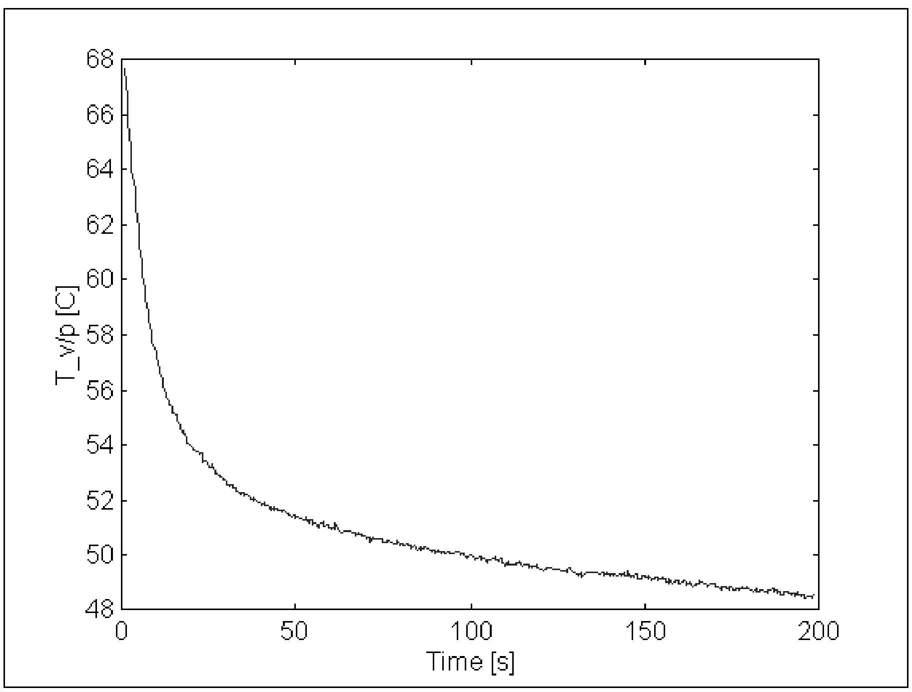

Figure 37: *Step response: 2->8Nm3/h*

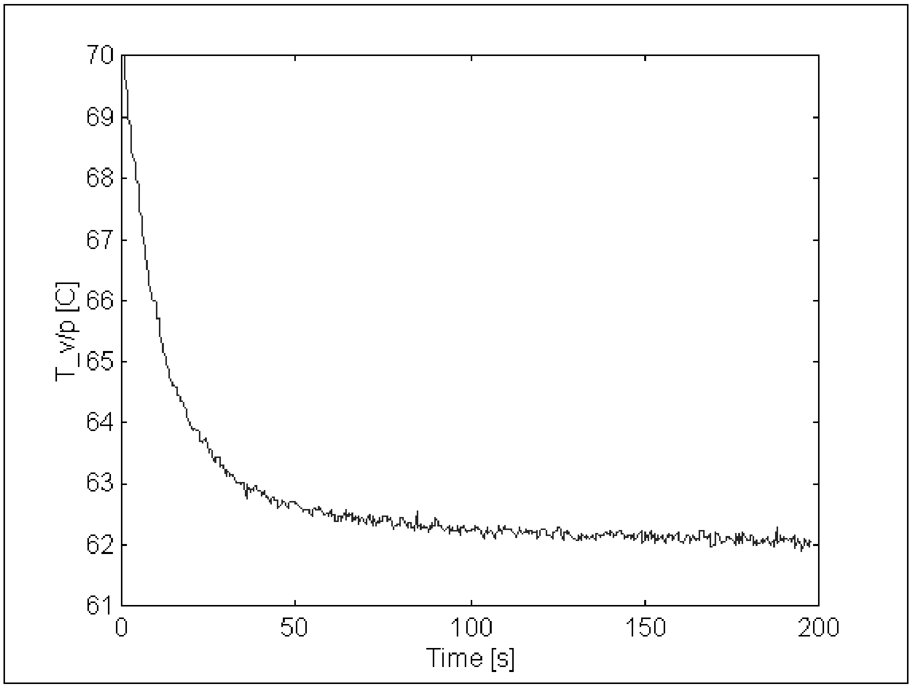

Figure 38: Step response: 2->5Nm3/h