# Flow- and concentration variation between the cylinders of a diesel engine

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

by
**David Näsström**

Reg nr: LiTH-ISY-EX -- 07/3957 -- SE

October 31, 2007

# Flow- and concentration variation between the cylinders of a diesel engine

**Master's thesis**

performed in **Vehicular Systems**,
**Dept. of Electrical Engineering**
at **Linköpings universitet**

by **David Näsström**

Reg nr: LiTH-ISY-EX -- 07/3957 -- SE

Supervisor: **Doctor Ola Stenlåås**
Scania CV AB
**Johan Wahlström**
Linköpings Universitet

Examiner: **Associate Professor Lars Eriksson**
Linköpings Universitet

Linköping, October 31, 2007

**Avdelning, Institution**
Division, Department

Division of Vehicular Systems
Department of Electrical Engineering
Linköpings universitet
SE-581 83 Linköping, Sweden

**Datum**
Date

2007-10-31

**Språk**
Language

☐ Svenska/Swedish
☒ Engelska/English
☐ ______________

**Rapporttyp**
Report category

☐ Licentiatavhandling
☒ Examensarbete
☐ C-uppsats
☐ D-uppsats
☐ Övrig rapport
☐ ______________

**URL för elektronisk version**

`http://www.fs.isy.liu.se/Publications/`
`http://www.ep.liu.se/`

**ISBN**
—

**ISRN**
LITH-ISY-EX--07/3957--SE

**Serietitel och serienummer**
Title of series, numbering

**ISSN**
—

**Titel**
Title

Flödes- och koncentrationsvariation mellan cylindrarna hos en dieselmotor
Flow- and concentration variation between the cylinders of a diesel engine

**Författare**
Author

David Näsström

**Sammanfattning**
Abstract

The demands on tomorrows diesel engines regarding fuel consumption and emission levels keep getting more difficult to fulfill. Due to this fact, the control demand is getting bigger and bigger. To be able to comply with the Euro 6 standards, it is believed that engine control need to be conducted individually from cylinder to cylinder if the need for after-treatment systems should be minimized.

Scania's approach to handle emission levels so far has been to use exhaust gas recirculation (EGR). To be able to optimize the use of EGR it is necessary to know how the inert gases, water and carbon dioxide, are distributed between the cylinders. The distribution variation become even more difficult to predict since the EGR is cooled, sometimes leading to condensation of some of the water content. The condensation of water and its behavior in the inlet manifold is studied in this thesis.

Different ways of measuring non-uniformity in the gas composition between cylinders with respect to EGR in general and water content in particular are evaluated. Using these results, measurements have been conducted on an engine and conclusions are drawn from them.

The conclusions are that uneven distribution of above all liquid water, due to puddle formation, have an impact on emission formation that should be accounted for in some of the examined operating conditions.

**Nyckelord**
Keywords

Diesel engine, EGR, liquid water, cylinder to cylinder charge variation, measuring water

# Abstract

The demands on tomorrows diesel engines regarding fuel consumption and emission levels keep getting more difficult to fulfill. Due to this fact, the control demand is getting bigger and bigger. To be able to comply with the Euro 6 standards, it is believed that engine control need to be conducted individually from cylinder to cylinder if the need for after-treatment systems should be minimized.

Scania's approach to handle emission levels so far has been to use exhaust gas recirculation (EGR). To be able to optimize the use of EGR it is necessary to know how the inert gases, water and carbon dioxide, are distributed between the cylinders. The distribution variation become even more difficult to predict since the EGR is cooled, sometimes leading to condensation of some of the water content. The condensation of water and its behavior in the inlet manifold is studied in this thesis.

Different ways of measuring non-uniformity in the gas composition between cylinders with respect to EGR in general and water content in particular are evaluated. Using these results, measurements have been conducted on an engine and conclusions are drawn from them.

The conclusions are that uneven distribution of above all liquid water, due to puddle formation, have an impact on emission formation that should be accounted for in some of the examined operating conditions.

**Keywords:** Diesel engine, EGR, liquid water, cylinder to cylinder charge variation, measuring water

# Contents

**Abstract** **iii**

**1 Introduction** **1**
- 1.1 The Scania EGR concept . . . 2
  - 1.1.1 Water condensation . . . 2
  - 1.1.2 The effect of EGR . . . 2
  - 1.1.3 Importance of uniformity . . . 3
- 1.2 Measuring water . . . 3
- 1.3 Note on confidentiality . . . 4

**2 Theory** **5**
- 2.1 Temperature change . . . 5
  - 2.1.1 Theoretical calculations . . . 5
  - 2.1.2 Conclusion . . . 6
- 2.2 Pressure buildup . . . 7
  - 2.2.1 Theoretical calculations . . . 7
  - 2.2.2 Conclusion . . . 7
- 2.3 Combustion chemistry . . . 8
  - 2.3.1 Theoretical calculations . . . 8
  - 2.3.2 Stability analysis . . . 10
  - 2.3.3 Conclusion . . . 10
- 2.4 Simplified combustion chemistry . . . 11
  - 2.4.1 Theoretical calculations . . . 12
  - 2.4.2 Stability analysis . . . 14
  - 2.4.3 Conclusions . . . 15

**3 Experiments** **18**
- 3.1 Experimental setup . . . 18
  - 3.1.1 Measuring devices . . . 19
- 3.2 Experiment plan . . . 21
  - 3.2.1 Operation points . . . 22
- 3.3 Experiments . . . 24
  - 3.3.1 Steady-state measurements . . . 25

3.3.2 Step-response measurements . . . . . . . . . . . . . . 25
3.4 Actual measurement points . . . . . . . . . . . . . . . . . . . 25

**4 Analysis** **27**
4.1 Stationary . . . . . . . . . . . . . . . . . . . . . . . . . . 27
4.1.1 Result . . . . . . . . . . . . . . . . . . . . . . . . . 32
4.2 Dynamics . . . . . . . . . . . . . . . . . . . . . . . . . . 33
4.2.1 Result . . . . . . . . . . . . . . . . . . . . . . . . . 38

**5 Conclusions and future work** **39**
5.1 Conclusions . . . . . . . . . . . . . . . . . . . . . . . . . 39
5.2 Future work . . . . . . . . . . . . . . . . . . . . . . . . . 40

**References** **41**

**Notation** **42**

# Chapter 1

# Introduction

Emission legislations keep getting more and more stringent. Among the manufacturers of heavy trucks and buses, there are a few concepts used generally to reduce the vehicle out emissions. With the diesel engine, the combustion is typically conducted in an environment with an excess of oxygen. The most difficult emissions to control in modern diesel engines are typically NO$_x$ and particulates, commonly abbreviated PM. Several after-treatment systems exist, the most commonly used are selective catalytic reduction - SCR, NO$_x$-adsorber, oxidizing catalysts, and particulate filters. Scania's approach, however, is mainly focused on preventing the emissions to form already in the combustion chamber.

One issue in the control systems of Scania's engines today is that the emission reducing systems are not controlled for individual cylinders. To do this, more information regarding both the amount, as well as the composition, of the gases that enter each cylinder is needed.

The goal with this thesis is to evaluate different ways of measuring gas composition in the inlet manifold on a cylinder-individual basis. Also covered by the scope of the thesis is performing measurements based on the evaluation, using those measurements to try and determine the localization and behavior of liquid water in the inlet manifold. And looking at the general distribution of EGR gas between the cylinders.

In this report, different methods of measuring the composition of the gases that enter each cylinder is discussed in chapter 2. This results in a measurement method being chosen and performed in an experiment described in chapter 3. The result and the following analysis is presented in chapter 4 and it is finally concluded in chapter 5.1, where ideas and suggestions for future work also can be found.

## 1.1 The Scania EGR concept

To achieve a reduction of emission formation inside the combustion chamber, exhaust gases are used to dilute the charge air (exhaust gas recirculation, from here on the acronym “EGR” will be used both for the gas and for the concept). Because the exhaust gases are very hot (it’s not uncommon with temperatures exceeding $500\,^\circ\mathrm{C}$), they need to be cooled to raise the density and to prevent heating of the charge air mixture.

The system used on the engine on which calculations and measurements has been conducted consists of some different parts. It utilizes an advanced common-rail fuel system that is able to deliver fuel at high pressures, it also incorporates a variable geometry turbo-charger.

For the EGR, the exhaust gases are sampled just before the turbine and led past a proportional valve and on to a cooling system. After being cooled down, the EGR and the fresh air are mixed together before the inlet manifold.

### 1.1.1 Water condensation

When the EGR is cooled from approximately $500\,^\circ\mathrm{C}$ down to temperatures well below $100\,^\circ\mathrm{C}$, water is condensed. This is believed to occur mainly in the cooling system. Theoretical calculations based on $CO_2$ concentrations in the exhaust gas as well as pressure and temperature after the cooler gives a picture of at which operational points condensation should occur. It is potentially a big problem since high levels of EGR is used, resulting in high relative humidity as well as large amounts of water (several per cents volume part) in the intake.

While the EGR in gas phase probably blends quite well with the fresh air (measurements of $CO_2$ concentration in the inlet manifold show steady and reliable values), the liquid water could easily be un-evenly distributed between the cylinders. In this thesis, there will be attempts to measure the distribution of both water as well as carbon dioxide.

### 1.1.2 The effect of EGR

As mentioned earlier, the biggest challenge in large diesel engines today is to reduce particulates and nitrous oxides, and at the same time maintain the high fuel efficiency. The reason that EGR is helpful in doing this is because there is a trade-off between high efficiency and low $NO_x$ formation. This trade-off is based on combustion temperature, where a high combustion temperature will give high efficiency but at the same time oxidize nitrogen. A high combustion temperature also helps oxidizing particulates, therefore introducing another trade-off between particulates and nitrous oxides.

One solution to this problem is to use high injection pressures to increase the length of the fuel-jet before it starts burning, this is called “lift-off”, thereby increasing the air-fuel mixing and reducing particulate formation.

At the same time it is possible to lower the local peak combustion temperature by diluting the fresh air and thereby the oxygen, increasing the distance between the locations where oxidation can occur. This is done by introducing inert gases, either by containing burnt gases from the previous cycle, or by the use of EGR. The use of inert gases also affect the average combustion temperature by increasing the mass in the cylinder. The use of EGR is currently the main course that Scania has taken to comply with future legislation demands.

### 1.1.3 Importance of uniformity

The $NO_x$ formation rate as function of local temperature is approximately exponential, and due to the connection between EGR-fraction and combustion temperature it follows that $NO_x$ formation is dependent on EGR-fraction in an exponential manner. This is shown by Magnus Andersson in [1] where he measures $NO_x$ output as a function of inert EGR (EGR / $\lambda$).

A high use of EGR can affect fuel economy and particulate formation negatively because it requires a more workload from the compressor to keep $\lambda$ at the same level, thereby increasing the pumping-work. Therefore it is important not to use an excessive amount of EGR.

If the inert gases in the EGR is distributed un-evenly it causes some cylinders to have a combustion temperature that is higher than others, which could give a much larger difference in $NO_x$ production than if the temperatures were the same, due to the exponential relationship.

Or you can look at it from another perspective and say that the global EGR-fraction could be lower with preserved $NO_x$ formation if all cylinders received the same amount.

It is very difficult and costly to ensure a perfect distribution using hardware, but if the differences are known, the fuel system could be used to still give the same combustion temperature in all the cylinders by balancing the $\lambda$ to attain a uniform amount of inert EGR.

## 1.2 Measuring water

One of the reasons why the distribution of water hasn't been evaluated in depth earlier at Scania is because it isn't obvious how to measure it. If possible, the measurements should separate liquid water and water vapor. This is difficult because of the ease with which water changes phase. It is also unknown whether the liquid water is mainly present as puddles in the inlet manifold or as an aerosol.

The standard equipment in Scania's test cells does not include a measuring device for measuring water, something that limits the possibilities to approach the problem with a trial-and-error tactics. In chapter 2, different measuring methods are discussed. Each method's pros and cons and also its theoretical accuracy is evaluated and finally a method to use in a practical experiment is chosen.

## 1.3 Note on confidentiality

Because some of the material used during this thesis is considered confidential, measures have been taken to make presentation of data more general. The biggest change is the presentation of at which operating conditions the data is valid for. Instead of describing engine load in Nm, a relative scale is used where 0 represents zero indicated torque and 1 represents maximum indicated torque at that particular engine speed. In the same manner the engine speed is not displayed as rpm but as an index where 0 is assigned to one engine revolution speed and 1 is assigned to another speed larger than the previous. Both transformations are linear though.

# Chapter 2

# Theory

In this chapter, four different methods of measuring the water, or the effects of the water is discussed. Finally, two of them are chosen to be tried out in a test cell at Scania's development laboratory.

## 2.1 Temperature change

The amount of water that theoretically is condensed in the EGR-cooler is fairly easy to calculate. Using the flow of air and the EGR-fraction, it is also possible to calculate the fraction of liquid water entering the inlet manifold. It is much more difficult to determine if, and how much, water is re-evaporated in the inlet manifold. The problem is not made easier by not knowing if the liquid water exists in puddles or as an aerosol or, more likely, both.

If the water doesn't re-evaporate in considerable amounts, the temperature gradient in the inlet manifold should be similar to that of the engine running without EGR altogether. If, on the other hand, the water is evaporating, it might be noticeable as a different temperature gradient in the inlet manifold.

In the following section some basic calculations will be made to estimate the effects of water evaporating under the existing conditions.

### 2.1.1 Theoretical calculations

Consider a gas with a consistency similar to fresh air diluted with some amount of EGR. Here all fractions are displayed as mole fractions. The gas' composition is chosen as 78 % $N_2$, 18 % $O_2$, 2 % $CO_2$ and 2 % $H_2O$ water vapour will be used. If an amount of liquid water equal to 1 % where to evaporate, thus increasing the amount of water vapor with 50 %, how much would that lower the average gas temperature, if all energy required is contributed by the gas itself?

Calculating the total heat capacity of the gas by summarizing the c$_p$ of the substances weighed with their respective mass fractions according to table 2.1, gives a total c$_p$ of 1.05 kJ/(kg · K). Since water has a heat of vaporisation ($L$) of 2260 kJ/kg [7], vaporising an additional amount corresponding to 1 % (1 % mole fraction, corresponding to 0.6 % mass fraction) would lower the average temperature by 13.4 degrees (equation 2.1).

$$\Delta T = \frac{L \cdot X_{m,H_2O_l}}{c_{p,tot}} = \frac{2260\ [kJ/kg] \cdot 0.6\%}{1.05\ [kJ/(kg \cdot K)]} = 13.4\ [K] \qquad (2.1)$$

| Substance | Mole fraction | Weight [AMU] | Mass fraction | c$_p$ [4] [kJ/(kg · K)] |
|---|---|---|---|---|
| $N_2$ | 0.78 | 28 | 75.7 % | 1.04 |
| $O_2$ | 0.18 | 32 | 20.0 % | 0.92 |
| $CO_2$ | 0.02 | 44 | 3.1 % | 1.85 |
| $H_2O$ | 0.02 | 18 | 1.2 % | 1.86 |

Table 2.1: Individual $c_p$ and fractions for the gas mixture

### 2.1.2 Conclusion

The temperature change is pretty large and should be possible to detect using ordinary thermocouples or pt-100 sensors. However, it should be noted that in case the evaporating occurs from water puddles and not from aerosol water, the energy could be provided by the manifold and the puddles themselves, in addition to coming from the gas.

Regardless, the chance of seeing the effects of vaporization using temperature measurements are estimated to be good even if it probably won't be enough to quantify the phenomenon.

In case of puddles forming in the intake manifold, another effect that could be studied is the transient behavior of the water. E.g. a step from much EGR to no EGR could be performed and the temperature behavior for each port can be analyzed. If there are no build-up of water anywhere, the temperature should react almost instantaneously, if on the other hand, there are puddles or wall wetting, it might be possible to see a different behavior of the temperature at the different positions.

The conclusion is that the phenomenon shows promise to being measurable, even if it is probably difficult to quantify the amount of water solely by a temperature measurement. Therefore, different methods of analysis are investigated.

## 2.2 Pressure buildup

The composition of the charge air does not influence the compression stroke in any matter that would be detectable on a pressure trace due to similar heat capacities. However, if liquid water enters the cylinder and then re-evaporates, it should theoretically decrease its density with approximately a thousand times at normal pressure. Calculations have been done to approximate the effect on the pressure trace during the compression stroke.

Two different mechanisms have been considered; first the change in density due to vaporisation, second the lowering of the charge air temperature as the energy is transfered to the liquid water to cause it to re-evaporate. The change in $c_p$ has also been considered.

The following calculations are based on the assumption that the cylinder are filled with a gas mixture consisting of 78 % $N_2$, 18 % $O_2$, 2 % $CO_2$ and 2 % $H_2O$ in gas phase. The comparison is then done when $H_2O$ in liquid phase corresponding to 1 % is added, just as in section 2.1.1. The comparatively high density of water allows for the assumption that it does not affect the initial amount of gas charge in the cylinder.

All numerical values are presented as for $300\,^{\circ}\mathrm{C}$, a point in the compression when the water should have evaporated.

### 2.2.1 Theoretical calculations

The calculations are based on the gas mixture mentioned above. In table 2.2 the substances are listed together with their respective $c_p$. This gives a weighed total $c_p$ of 30.74 kJ/(kmole · K) for the gas. Evaporating the extra per cent of water changes the total $c_p$ to 30.79 kJ/(kmole · K).

Considering the relative weights of the molecules, it is possible to use the $c_p$ and the heat of evaporation to calculate the change in temperature as well as the relative change in molecule quantity.

The result suggests that the temperature difference will be a lowering with approximately $13.4\,^{\circ}\mathrm{C}$ (see equation 2.1) or 2.3 %, considering a temperature of $300\,^{\circ}\mathrm{C}$, if ideal gas is assumed, this would of course result in pressure lowering 2.3 % as well. The added molecules, however, cause a pressure rise of 1 % since the increase in the number of gas molecules is exactly 1 %, giving a total effect on pressure of -1.3 % (1 % - 2.3 %).

### 2.2.2 Conclusion

The resulting -1.3 % in pressure is unfortunately below the measurement uncertainties when using the cylinder pressure sensors that are being utilized at Scania today. On top of that the effect becomes smaller as the temperature

| Substance | Mole fraction | $c_p$ [4] [kJ/(kmole · K)] |
|---|---|---|
| $N_2$ | 0.78 | 29.93 |
| $O_2$ | 0.18 | 31.87 |
| $CO_2$ | 0.02 | 46.69 |
| $H_2O$ | 0.02 | 35.83 |

Table 2.2: Individual $c_p$ and fractions for the gas mixture

increases since the relative temperature change become smaller while the relative increase in molecules remains constant at 1 %.

Therefore this method would not be able to detect differences within the liquid water distribution, at least not at the levels that are expected during normal operation.

Further more, this method would not be able to detect differences in the $CO_2$ distribution.

## 2.3 Combustion chemistry

Using assumptions regarding the combustion chemistry, and allowing concentration measurements, it should be possible to form an equation system which could be solvable for each cylinder. Using the injected amount of fuel, the number of molecules can be locked and the distribution between the cylinders calculated.

### 2.3.1 Theoretical calculations

A simple model of the combustion in a diesel engine is assumed where the only combustion products formed are water and carbon dioxide, as well as $NO_x$, HC and CO.

The assumptions made are reflected in the following equations (capital letters A,B,D-G,I-M and P-U represent different amounts in moles or any similar quantity):

$$FUEL = A \cdot C_a H_b O_c \tag{2.2}$$

$$AIR = B \cdot O_2 + 3.773B \cdot N_2 \tag{2.3}$$

$$\begin{aligned} EGR = {} & D \cdot O_2 + E \cdot N_2 + F \cdot CO_2 + G \cdot H_2O_g + \\ & + I \cdot H_2O_l + J \cdot CO + K \cdot NO_x + L \cdot H_y C_z \end{aligned} \tag{2.4}$$

$$\begin{aligned} EXHAUST = {} & M \cdot O_2 + P \cdot N_2 + Q \cdot CO_2 + R \cdot H_2O + \\ & + S \cdot CO + T \cdot NO_x + U \cdot H_y C_z \end{aligned} \tag{2.5}$$

If the equations above are combined it gives the complete combustion equation:

$$AIR + EGR + FUEL \rightarrow EXHAUST \tag{2.6}$$

The result is a system with 17 unknowns, the system can be used with global measurements but here it will be defined separately for each cylinder. The number of unknowns is under the assumption that the composition of the fuel, emissions and fresh air is completely known. In other words, $a, b, c, x, y$ and $z$ are assumed to be known for each measurement point.

To solve the problem, 17 equations needs to be specified for each cylinder; one equation is given by fuel measuring and an additional four equations is given by the balance of the elements (C, H, N and O). This leaves 12 equations to be determined by measurements.

Three different locations are available for gas sampling. Before the cylinder, after the cylinder and in the common exhausts, at these locations the composition of the gas can be analyzed, of course depending on what instruments are available. Since the EGR is extracted from a point in the exhaust manifold where gas from all cylinders come together, it can be assumed that it has the same composition as the common exhaust during steady-state operation. It is also assumed that all gaseous species mix just as good so that the relative amounts of all species except $H_2O$ in the EGR stay the same regardless at which cylinder the composition is studied.

Not all measurements are performed in the same way; some of them are measuring the species concentration relative to dry air (i.e. with all water condensed and removed, here valid for $O_2$, CO and $CO_2$ concentrations) or to wet air (with water vapor present, $NO_x$ and HC).

Water is avoided for measurements and the following species are assumed to be measurable: $O_2$, CO, $CO_2$, HC and $NO_x$. Measurements before and after the individual cylinders will provide for 10 additional equations (per cylinder). That leaves 2 equations to be filled.

If it is assumed that the EGR in gas phase have the same composition for the cylinder inlets, a concentration measurement (a “dry” measurement of any species) in the common exhausts will provide for one additional equation, corresponding to the amount of EGR that is delivered, leaving only one to completely determine the equation system. It is however not possible to use more than one measurement from the common exhausts since that equation would be a linear combination of the other measurement from the common exhausts and the measurements before the cylinder. This is because the composition of the EGR is known from the measurements at the cylinder inlet and the only additional information that can be extracted is the fraction of exhaust gas that is recirculated. At least if the assumption of equal blending is valid.

The only option that remains is to use one measurement of water vapor concentration, it is chosen to be conducted after the cylinder because all water can be assumed to be in the same phase there due to the high temperatures.

The equation system has the form:

$$\bar{A}\bar{x} = \bar{B} \tag{2.7}$$

where $\bar{A}$ consists of all measurements except the fuel measurement, $\bar{B}$ is formed by zeros and the fuel measurement and $\bar{x}$ is the variables $A$ through $U$ in equations 2.2 to 2.5.

### 2.3.2 Stability analysis

Since the equation system consists of many different measurements it should be tested for stability. Even if the system is linearly independent, it is no guarantee for it not being near-singular. Since all measurements except the fuel measurement are for fractions, the fuel measurement is the one locking the amounts to enable comparison between cylinders.

Since the $\bar{A}$ matrix consists of measurement values, they must be defined before any mathematical analysis can be made. To do this without needing actual measurements, syntetic measurements were created and are presented in table 2.3.

The amounts are recalculated into concentrations for each species and fed into a Matlab script that solves the equation system. The Matlab function “COND” is used to estimate the systems condition number. “COND” returns the 2-norm condition number, a number that predicts the effect of an error in the input, or in the model itself. It can be interpreted as the relation between a relative error on the input and the resulting relative error on the solution.

Unfortunately the condition number was 78,312. That is far too big and it is examplified by a change of 1 % in the prediction of the $NO_x$ composition, from $NO_{1.5}$ to $NO_{1.515}$, which causes the estimation of liquid water before the cylinder to change from 4 to 0.83, a decrease of 83 %.

### 2.3.3 Conclusion

The condition number points to a very ill-conditioned system, this is unfortunate since the system is likely to suffer from both considerable measurement errors as well as errors originating from a too simple combustion model. This leads to the conclusion that this is not a good way to measure EGR distribution. However, the system was solvable and it is possible that with more water measurements and a different combustion model, the condition number could be improved.

| **Fresh air** | |
|---|---|
| Species | Amount |
| $N_2$ | 37.73 |
| $O_2$ | 10 |

| **EGR** | |
|---|---|
| Species | Amount |
| $N_2$ | 12 |
| $O_2$ | 6 |
| $CO_2$ | 5 |
| $H_2O_g$ | 6 |
| $H_2O_l$ | 4 |
| CO | 3 |
| $NO_{1.5}$ | 2 |
| $H_4C_1$ | 1 |

| **Fuel** | |
|---|---|
| Species | Amount |
| $C_4H_6O_2$ | 5 |

| **Exhaust** | |
|---|---|
| Species | Amount |
| $N_2$ | 45.73 |
| $O_2$ | 2 |
| $CO_2$ | 5 |
| $H_2O_g$ | 19 |
| CO | 20 |
| $NO_{1.5}$ | 10 |
| $H_4C_1$ | 4 |

Table 2.3: Model data for construction of measurements

## 2.4 Simplified combustion chemistry

The combustion chemistry in the previous section proved to be too sensitive to model- and measurement error. It also relied on a large amount of measurements, which tends to become costly and a source of errors. Therefore, an attempt is done to assume a more basic model and to allow measuring of water vapor concentration both before and after the combustion chamber on each cylinder. The resulting equation system and system stability will be discussed in the following sections.

### 2.4.1 Theoretical calculations

As in the previous section, 2.3, the engine is modeled using fuel, air, EGR and exhaust gases. This time ideal combustion is assumed, also the fuel is considered to consist only of hydrogen and carbon, resulting in the following equations:

$$FUEL = A \cdot C_a H_b \tag{2.8}$$

$$AIR = B \cdot O_2 + 3.773B \cdot N_2 \tag{2.9}$$

$$\begin{aligned} EGR = {} & D \cdot O_2 + E \cdot N_2 + F \cdot CO_2 + \\ & + G \cdot H_2O_g + I \cdot H_2O_l \end{aligned} \tag{2.10}$$

$$\begin{aligned} EXHAUST = {} & J \cdot O_2 + (3.773B + E) \cdot N_2 + \\ & + K \cdot CO_2 + L \cdot H_2O \end{aligned} \tag{2.11}$$

If the equations above are combined it gives the complete combustion equation:

$$AIR + EGR + FUEL \rightarrow EXHAUST \tag{2.12}$$

The result is a system with 10 unknowns ($A$ through $L$), if the fuel composition, $a$ and $b$, are determinable.

Obviously we would need 10 independent equations to solve the equation system. As before one equation is provided by a fuel measurement. Since the N-balance already is used in the formulation of equation 2.11, only 3 equations can be extracted from element balances (C, H and O). This leaves 6 equations to be supplied by concentration measurements.

The relative volumetric concentrations of $N_2$, $O_2$ and $CO_2$ in the EGR gas is assumed to be exactly those of the common exhausts, CE. Therefore, while running the engine in a stationary mode, measurements in the common exhausts will be representative for the gas mixture in the inlet manifold, as long as water or fresh air is not considered.

$$\frac{D}{D+E+F} = \frac{N_{O_2,CE}}{N_{O_2,CE} + N_{N_2,CE} + N_{CO_2,CE}} \tag{2.13}$$

$$\frac{F}{D+E+F} = \frac{N_{CO_2,CE}}{N_{O_2,CE} + N_{N_2,CE} + N_{CO_2,CE}} \tag{2.14}$$

Unfortunately, the instrument available for measuring oxygen concentration in the common exhausts doesn't dry the gas before measuring. Therefore the right hand side of equation 2.13 is not available. The solution is to assume that the engine has been operating in stationary mode long enough for the $CO_2$ and $H_2O$ in the common exhausts to have stabilized to an equilibrium defined by the carbon/hydrogen balance of the fuel. In that way, equation 2.13 can be reformulated as:

$$\frac{D}{D+E+F\cdot(1+\frac{b}{2a})} = \tag{2.15}$$
$$= \frac{N_{O_2,CE}}{N_{0_2,CE}+N_{N_2,CE}+N_{CO_2,CE}+N_{H_2O,CE}}$$

The measurements in the intake and exhaust manifolds will be done using a FTIR instrument (further information is available in chapter 3). This instrument is not capable to measure homonuclear diatomic gases such as $O_2$ and $N_2$. Therefore the measurements will be limited to $CO_2$ and $H_2O$.

To summarize; with the nomenclature defined in equations 2.8 to 2.11, the measurements will consist of:

$$\dot{m}_{FUEL} = A\cdot\frac{A_{r,FUEL}}{1000\cdot N_A\cdot t_{ref}} \tag{2.16}$$

$$X_{O_2,CE} = \frac{D}{D+E+F\cdot(1+\frac{b}{2a})} \tag{2.17}$$

$$X_{CO_2,CE} = \frac{F}{D+E+F} \tag{2.18}$$

$$X_{CO_2,INT} = \frac{F}{4.773\cdot B+D+E+F+G} \tag{2.19}$$

$$X_{H_2O,INT} = \frac{G}{4.773\cdot B+D+E+F+G} \tag{2.20}$$

$$X_{CO_2,EXH} = \frac{K}{3.773\cdot B+E+J+K+L} \tag{2.21}$$

$$X_{H_2O,EXH} = \frac{L}{3.773\cdot B+E+J+K+L} \tag{2.22}$$

The actual measurements is on the left hand side of the equations above and the equivalent expressions based on the combustion model is on the right hand side.

Worth noting is that the expression for the fuel is in fact not necessary as long as the same amount of fuel is delivered to all the cylinders and only fractions of the different species is of interest, then any number can be used.

Of course, the option of using an actual amount remains, this could be of interest e.g. if the amount of water in to the cylinder were to be calculated. In that case, the amount of fuel per stroke could be used, of course with a sufficiently large base for the number of molecules, to avoid unreasonably large numbers.

### 2.4.2 Stability analysis

Now the 10 equations are specified and combining them gives a solvable system. Just like before, in section 2.3, the matrix include measurements and to do a stability analysis, measurement data needs to be faked.

Here the constructed data has been chosen as to resemble realistic operation of the engine. The data can be found in table 2.4.

| **Fresh air** | |
|---|---|
| Species | Amount |
| $O_2$ | 42 |
| $N_2$ | 158.466 |

| **EGR** | |
|---|---|
| Species | Amount |
| $O_2$ | 8 |
| $N_2$ | 73 |
| $CO_2$ | 9 |
| $H_2O_g$ | 4 |
| $H_2O_l$ | 6 |

| **Fuel** | |
|---|---|
| Species | Amount |
| $C_{12}H_{26}$ | 2 |

| **Exhaust** | |
|---|---|
| Species | Amount |
| $O_2$ | 13 |
| $N_2$ | 231.466 |
| $CO_2$ | 33 |
| $H_2O_g$ | 36 |

Table 2.4: Model data for construction of measurements

Just as before, the Matlab function “COND” is used to estimate the condition number. In this case it is reported as 314. This is a big improvement compared to the previous attempt, but to be certain all the measurements and model parameters are changed with 1 % and 10 % to study the effects on the result.

In table 2.5 relative errors are shown. The errors are constructed by solving the system several times and each time altering one of the input data or model parameters 1 %. The solution is then compared to the “correct” solution.

It is clearly visible that the most sensitive parameter is the one for liquid water, and that it reacts the most to altered measurements in the exhaust manifold and to errors in the fuel composition estimation.

It is also interesting to see how the relative errors behave when subjected to increasing errors. Therefore, another test was conducted, this time with input data that had been altered by 10 $\%$. This data is seen in table 2.6 and it seems as if none of the relative errors grow more than linearly.

| Parameter | $A$ | $B$ | $D$ | $E$ | $F$ | $G$ | $I$ | $J$ | $K$ | $L$ |
|---|---|---|---|---|---|---|---|---|---|---|
| $X_{O_2,CE}$ | 0 | 0 | -1 | 0 | 0 | 0 | 0 | -1 | 0 | 0 |
| $X_{O_2,CE}$ | 0 | 0 | 1 | 1 | 0 | 0 | 0 | -1 | 0 | 0 |
| $X_{CO_2,INT}$ | 0 | 0 | -1 | -1 | -1 | 0 | -2 | -1 | 0 | 0 |
| $X_{H_2O,INT}$ | 0 | 0 | 0 | 0 | 0 | -1 | 1 | 0 | 0 | 0 |
| $X_{CO_2,EXH}$ | 0 | 1 | 1 | 1 | 1 | 1 | 7 | 5 | 0 | 1 |
| $X_{H_2O,EXH}$ | 0 | 0 | 0 | 0 | 0 | 0 | -6 | 1 | 0 | -1 |
| $a$ | 0 | 1 | 1 | 1 | 1 | 1 | 5 | 2 | 1 | 1 |
| $b$ | 0 | 0 | 0 | 0 | 0 | 0 | -4 | -1 | 0 | 0 |

Table 2.5: Result variation due to 1 % variation of input [%]

| Parameter | $A$ | $B$ | $D$ | $E$ | $F$ | $G$ | $I$ | $J$ | $K$ | $L$ |
|---|---|---|---|---|---|---|---|---|---|---|
| $X_{O_2,CE}$ | 0 | 0 | -9 | 1 | 0 | 0 | 0 | -6 | 0 | 0 |
| $X_{CO_2,CE}$ | 0 | -4 | 9 | 11 | 0 | 0 | 0 | -9 | 0 | 0 |
| $X_{CO_2,INT}$ | 0 | 1 | -12 | -12 | -12 | -3 | -17 | -5 | -3 | -3 |
| $X_{H_2O,INT}$ | 0 | 0 | 0 | 0 | 0 | -9 | 6 | 0 | 0 | 0 |
| $X_{CO_2,EXH}$ | 0 | 13 | 13 | 13 | 13 | 13 | 75 | 51 | 4 | 14 |
| $X_{H_2O,EXH}$ | 0 | 2 | 2 | 2 | 2 | 2 | -53 | 6 | 0 | -9 |
| $a$ | 0 | 9 | 8 | 10 | 9 | 9 | 53 | 17 | 10 | 10 |
| $b$ | 0 | 1 | 2 | 0 | 1 | 1 | -43 | -7 | 0 | 0 |

Table 2.6: Result variation due to 10 % variation of input [%]

### 2.4.3 Conclusions

Using an even simpler combustion model and allowing two water vapor measurements increased the theoretical stability of the system more than a hundredfold. This also shows when verifying with faked measurements. At the same time it requires a modest number of measurements and it is feasible to realize with satisfactory results.

The model is based on a pretty heavy assumption however, ideal combustion as in equation 2.11 where the only combustion products are assumed to be $CO_2$ and $H_2O$, is of course not the case in a diesel engine. It will affect the results in a negative manner and to get a picture of how big the problem is some previously collected data is analysed. The main emissions from the engine, apart from $CO_2$ and $H_2O$, is considered to be CO, HC and $NO_x$. If these emissions are summed up (the HC readings are corrected to resemble unburnt diesel fuel), and plotted in a load/engine speed plot (figure 2.1), it shows the potential of the problem.

It is clearly visible that the only area that have the potential to cause larger errors is the high load/low engine speed area with the emissions contributing for more than 0.25 % of the total gas amount. Generally the engine operates with emissions below 0.05 %, a level that hardly affects the measurement accuracy since the levels of $CO_2$ and $H_2O$ is generally at least a few per cents. Also, since the troublesome area is situated at such low engine speeds, it is not likely to coincide with measurement points and the assumption regarding perfect combustion is valid.

Among the different measurement methods discussed, this one clearly shows the highest potential of being able to deliver both a qualitative as well as a quantitative measurement. Therefore, the experiment setup will be based on the measurements mentioned in this chapter.

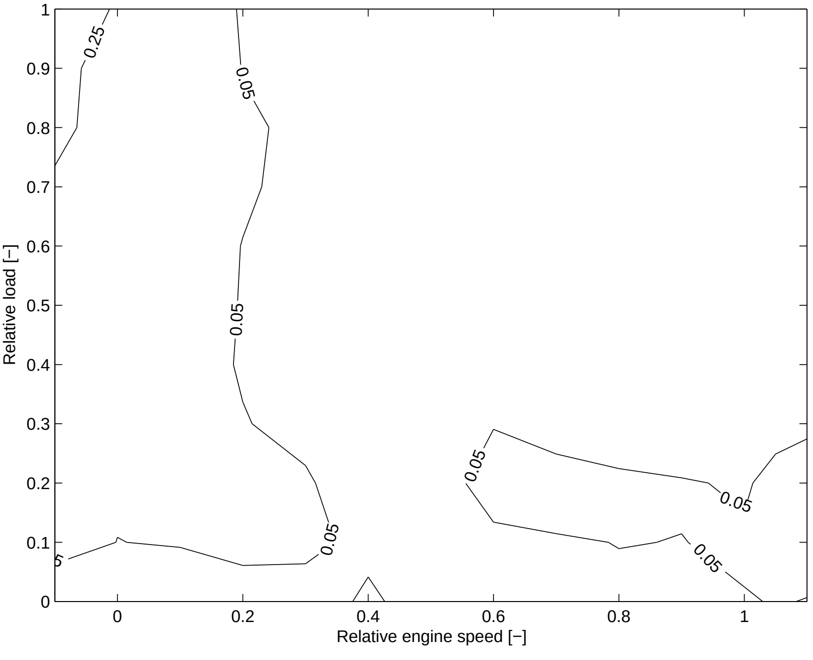

Figure 2.1: Sum of the mole fractions of CO, HC and NOx, in the common exhaust [%]

# Chapter 3

# Experiments

In this chapter, the experiment setup will be described, together with a quick description of the instruments used. An experiment plan will also be defined, where it will be decided at which engine operating points the measurements will be conducted.

Since previous measurements at Scania haven't included cylinder-individual EGR with the species needed to verify these models, additional measurements are required. Because it has not been performed at Scania before, instruments and engine hardware is not already provided for.

Measurements will be conducted in both the intake as well as the exhaust ports of all the cylinders, also the additional oxygen measurement in the global exhaust as mentioned in section 2.4 needs to be arranged since it isn't part of the standard measurement equipment at Scania's test cells.

## 3.1 Experimental setup

The chosen method includes gas sampling from individual exhaust ports as well as inlet ports. Also a gas measurement from the common exhausts is required. This sums up to 13 different measurement locations, it is not realistic to use that many instruments and therefore the same instrument have to be used to analyze gas from several locations.

One alternative is to manually switch the outlet of gas so that it originates from different sources, this would not only be time consuming but could also be hazardous since it requires test cell access during the operation of the engine. A better method is to use valves that can be actuated from a distance and use those to guide gas from the current sampling location to the instrument.

For this experiment, solenoid valves that are normally used on Scania's V8 engines are being used. These valves open fully when subjected to 400 $mA$. Twelve of them are attached in parallel and their respective outlets are connected to the measurement device. The valves are mounted close to the turbocharger to provide passive heating of them (see figure 3.1). In that way, condensation of hydrocarbons and water can be reduced. Figure 3.2 display a principal sketch of the measurement setup and the location of gas-sampling points and location of the temperature sensors in the inlet manifold. Please note that the sketch is not complete, the VGT is not included and neither is the EGR cooling system, note also that each cylinder has two inlet channels but only one exhaust channel.

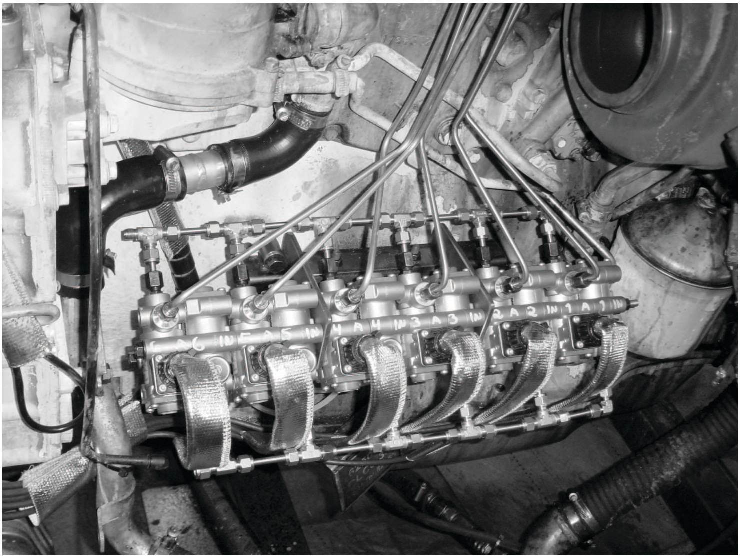

Figure 3.1: The solenoid valves mounted on the engine's hot side

### 3.1.1 Measuring devices

For the oxygen concentration a non-sampling type $NO_x$ analyzer with $O_2$ measurement will provide the data, it will be mounted after the turbine. While an analytical system from Horiba takes care of the $CO_2$ measurements in the common exhausts.

To analyze the $CO_2$- and $H_2O$-concentrations in the intake- and exhaust manifolds a FTIR-instrument is used.

More specific information regarding the instruments can be found in the following sections.

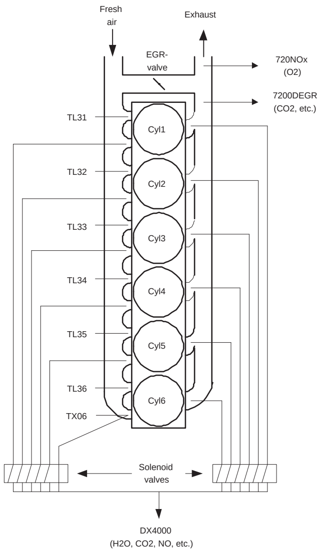

Figure 3.2: Measurement setup

### Gasmet DX4000

“FTIR” is an acronym for “Fourier Transform InfraRed”, it is a way of analyzing samples of gas. The following explanation is drawn from the homepage of the provider of the instrument used [2].

> “In infrared spectroscopy, IR radiation is passed through a sample of gaseous molecules. Some of this radiation is transmitted through while the rest is absorbed by the sample, producing an infrared spectrum, or ‘molecular fingerprint’. Because each molecular structure has a unique combination of atoms, each produces a unique infrared spectrum. From this, identification (Qualitative analysis) and analysis (Quantitative measurement) of the gas is possible, the two major applications of FTIR spectrometry.
>
> FTIR stands for Fourier Transform Infrared, the preferred method of infrared spectroscopy. Since chemical functional groups absorb light at specific frequencies, one can identify the make-up of the sample. As well, through calibration of the spectrometer, one can determine the intensity of the absorption (relates to the component concentration).”

The instrument used is a Gasmet DX4000, it has a accuracy of $\pm < 2$ % [3].

### Horiba MEXA-7200DEGR

Non dispersive Infrared (NDIR) sensors are simple spectroscopic devices often used for gas analysis. The main components are an infrared source (lamp), a sample chamber or light tube, a wavelength filter, and the infrared detector. The gas is pumped (or diffuses) into the sample chamber, and gas concentration is measured electro-optically by its absorption of a specific wavelength in the infrared (IR). The IR light is directed through the sample chamber toward the detector. The detector has an optical filter in front of it that eliminates all light except the wavelength that the selected gas molecules can absorb. Ideally other gas molecules do not absorb light at this wavelength, and do not affect the amount of light reaching the detector.

### Horiba MEXA-720NOx

The instrument utilizes a zirconia sensor [5] which uses an electrochemical cell that allows gas to pass through a diffusion barrier in to a cavity where oxygen is pumped using an ion pump. The pump is kept at a certain voltage and the pumping current is measured [8]. One problem is that $NO_2$ is reduced to NO in the cell, causing additional $O_2$ to form. If the $NO_x$- and $O_2$-levels are compared for the engine, the result is that the error introduced by $NO_2$ reduction is never larger than 3 %, and if only the area of interrest (see section 3.2.1) is looked upon, and it is considered that $NO_x$ in diesel exhausts typically consists of 90 % NO and 10 % $NO_2$ [6], the relative error never exceeds 0.2 %. That can be considered unsignificant when compared to the instrument's accuracy of ± 0.5 % [5].

## 3.2 Experiment plan

There are several interesting variables that could affect, above all, the behavior of the water in the EGR-cooler and inlet manifold. By using already existent measurements of the engine some different quantities were studied. This led to a decision on what measurement point that would be used. Please note that due to confidentiality, the axes has been re-scaled and translated and not all data is presented. This was of course not the case during the work of the thesis.

One interesting variable is relative humidity. If all the water is assumed to be in gas phase, i.e. total re-vaporisation is allowed, the result is that an area of the engine operation map display a relative humidity in excess of 100 %, implying that liquid water must be present. This can be seen in figure 3.3, please note that the area with the higher relative humidity is within the line.

Other quantities that were regarded when choosing measurement points are the mole fraction of liquid water and the relative humidity assuming no re-

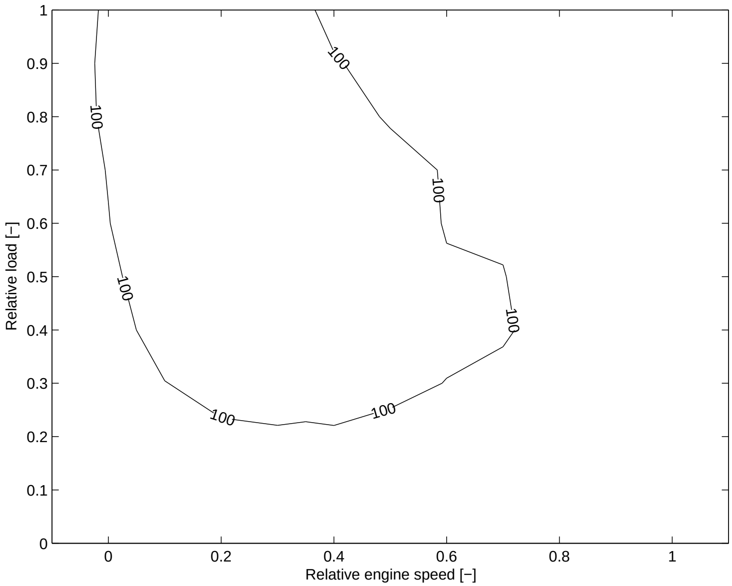

Figure 3.3: Relative humidity in inlet manifold if all water is evaporated [%]

vaporisation, both in the inlet manifold. Figure 3.4 shows the fraction of liquid water and two levels are shown, both the one per cent line as well as the areas with the maximum fractions. Later on, in section 3.2.1, these will be used when placing the measurement locations.

In figure 3.5, the relative humidity provided that no water re-evaporates is plotted. Here it is visible that the point with the least chance of re-evaporation (because of the high relative humidity) is the area just around engine speed = 0, load = 1.

### 3.2.1 Operation points

Based on the parameters in the previous section, measurement points were placed to try to capture the interesting phenomenons that are expected to occur. The coordinates of measurement points discussed here will be referred to as [relative engine speed, relative load].

As can be seen in figure 3.4, [0.2,0.5] and [0.4,0.5] contains the highest fractions of liquid water and therefore the highest potential for distribution variation. Furthermore, it is interesting to follow the line with constant 1 % liquid water share and see if there are any differences due to gas flow speed or EGR fraction, etc. The points placed along the 1 %-line is marked with squares (□) in figure 3.6 and the point at [0,1] coincides with the point with the highest

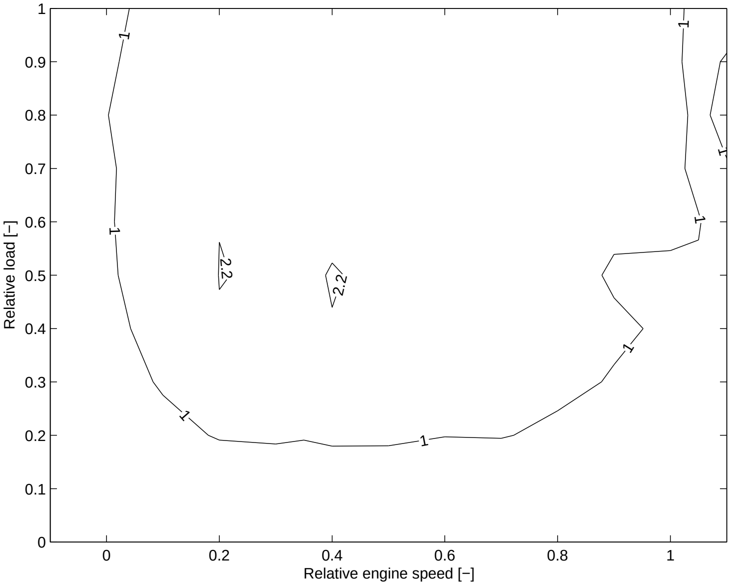

Figure 3.4: Liquid water fraction in inlet manifold [%]

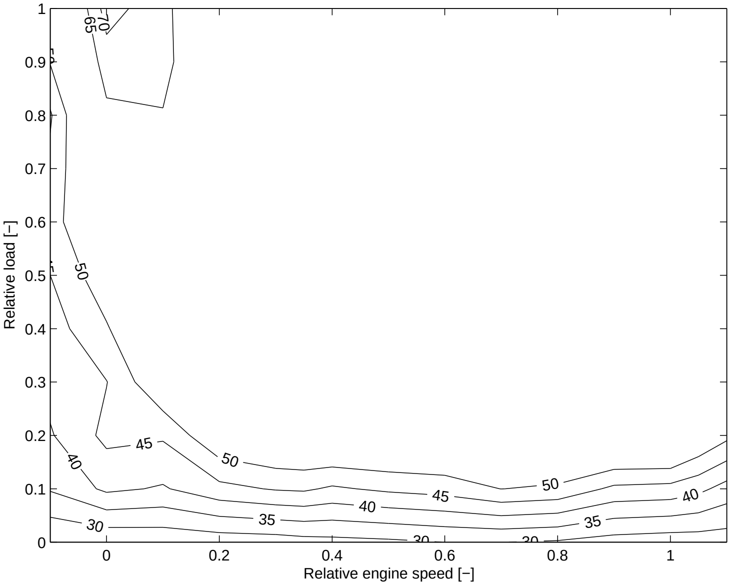

Figure 3.5: Relative humidity in inlet manifold with no re-vaporisation [%]

relative humidity in figure 3.5.

To study the EGR-fraction's impact on the water's behavior and also the distribution of $CO_2$, different levels of EGR is used in the same operation point. This point [0.4,0.5] is marked with a $\oplus$ in the figure.

The rest of the measurement points are chosen to fill out the map to facilitate implementation in the engine control system later on. That is something that won't be covered by the scope of this thesis however.

With the objective to study water puddle occurrence, some dynamic steps in EGR fraction will be made while studying inlet port temperatures.

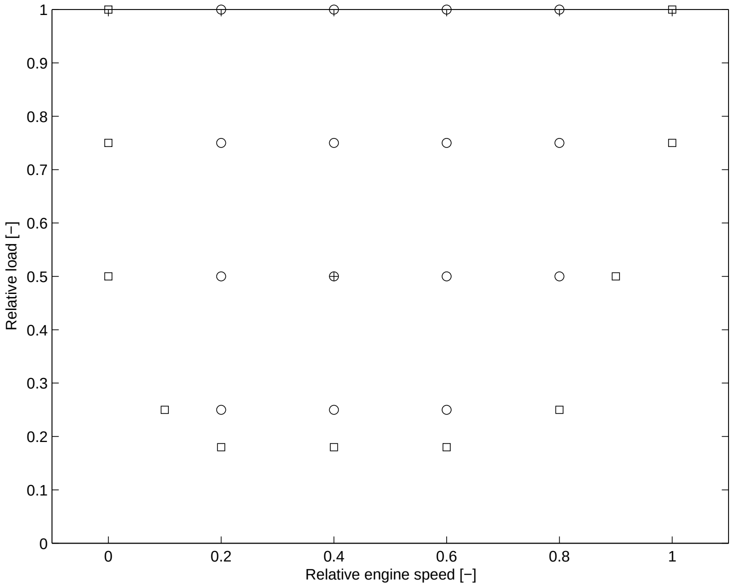

Figure 3.6: Engine operation points for measurements

## 3.3 Experiments

Two different kinds of experiments was conducted, one with steady-state measurements where only the gas concentrations was regarded, and one where step-responses was studied with respect to the temperatures in the different places in the inlet manifold as well as the water concentration at different locations in the inlet manifold. The steps was conducted as a sudden change in EGR fraction at otherwise steady-state operation of the engine.

### 3.3.1 Steady-state measurements

The Gasmet DX4000 was used together with an array of solenoid valves to measure water and carbon dioxide concentrations in the inlet and outlet ports of the 6 cylinders. Due to the single measurement channel of the instrument, the twelve different measurements had to be done sequentially.

At the same time as measuring $H_2O$ and $CO_2$, the instrument measured CO, NO and $NO_2$ concentrations as well.

In the common exhausts, after the turbine, the Horiba MEXA-720NOx measured $O_2$ concentration and the Horiba MEXA-7200DEGR measured the $CO_2$ levels (in dried gas).

The engine was run for a couple of minutes in each new operation point to be stabilized, then an addition amount of time for each new sampling point with the FTIR instrument, to allow condensated water in the piping to evaporate.

### 3.3.2 Step-response measurements

During the transient steps, a valve in the EGR piping was either fully opened, or closed shut, thus limiting the flow of EGR into the inlet manifold. While doing this, seven temperatures in the inlet manifold was recorded with a sample rate of 1 Hz. The steps was performed six times, and each time the FTIR instrument was used to measure the $H_2O$- and $CO_2$-concentrations at a different location in the inlet manifold. These measurements was performed at 1/6 Hz.

Some additional measurements was performed at other operating conditions, but with only temperature logging.

## 3.4 Actual measurement points

Due to problems with the test setup and the Gasmet instrument, the test-series had to be re-runned. This of course caused the time-frame for this thesis to be very limited. Therefore, all the measurements in the experiment plan from section 3.2 wasn't performed. The operating conditions represented in chapter 4 are plotted in figure 3.7

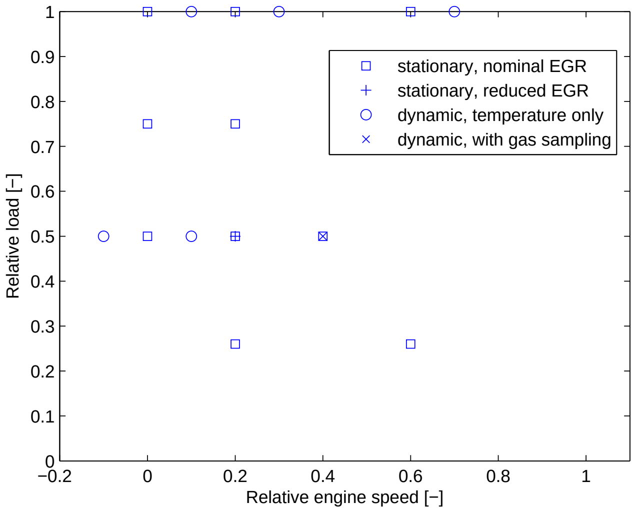

Figure 3.7: Actual operating conditions for measurements

# Chapter 4

# Analysis

Using the collected data, both from the stationary operating points, as well as from the dynamic steps in EGR, it should be possible to draw some conclusions regarding the presence and behavior of water in the inlet manifold. To do this, raw data is plotted and studied and also used as input to the model that calculates liquid water fraction (see section 2.4). For the analysis of the dynamic steps, some calculations based on simple physical relations are made to be able to quantify the amounts of water.

## 4.1 Stationary

The measurements consists mainly of concentration values from intake and exhaust ports of the different cylinders, and also from the common exhausts. Some additional data such as temperatures, massflows and pressures was also stored if they should be needed. The model described in section 2.4 was implemented in Matlab and the data was fed to that program.

Due to some problems with the measurement equipment there was not enough time available to do measurements at all the operating conditioned described in the experiment plan. Instead, ten different loads and engine revolution speeds was evaluated using nominal EGR-levels, as well as three different EGR-levels in one of the operating points.

Unfortunately, one of the solenoid valves used to choose gas sampling point started leaking and therefore there are no gas composition data available for cylinder five.

When studying the results from running the program, it is clear that some of the results can't be plausible. If the fraction of liquid water into the cylinders are studied, some of them show negative values, which of course is impossible. To verify that this is not a result of a sensitive system similar calculations are done just to receive the liquid water concentration. To do that it is assumed that the difference in CO$_2$- and H$_2$O-fractions in the intake and exhaust ports of the individual cylinders is directly a result of the amount of fuel that has been burnt.

In the following equations, index 1 represent conditions before combustion and index 2 after, $N_{tot}$ refers to the total number of gaseous molecules where the respective measurements are made, that being in the inlet manifold and in the exhaust manifold. The fuel is considered to consist of hydrocarbons with a C/H-relation of 1:1.85.

$$\begin{aligned}
&N_{H_2O,2} - N_{H_2O,1} = \frac{1.85}{2}(N_{CO_2,2} - N_{CO_2,1}) \Leftrightarrow \\
&N_{H_2O,2} - \frac{1.85}{2}N_{CO_2,2} = N_{H_2O,1} - \frac{1.85}{2}N_{CO_2,1} \Leftrightarrow \\
&\frac{N_{H_2O,2} - \frac{1.85}{2}N_{CO_2,2}}{N_{tot,1} \cdot N_{tot,2}} = \frac{N_{H_2O,1} - \frac{1.85}{2}N_{CO_2,1}}{N_{tot,1} \cdot N_{tot,2}} \Leftrightarrow \\
&\frac{1}{N_{tot,1}}(X_{H_2O,2} - \frac{1.85}{2}X_{CO_2,2}) = \frac{1}{N_{tot,2}}(X_{H_2O,1} - \frac{1.85}{2}X_{CO_2,1}) \Leftrightarrow \\
&X_{H_2O,2} - \frac{N_{tot,1}}{N_{tot,2}}X_{H_2O,1} = \frac{1.85}{2}(X_{CO_2,2} - \frac{N_{tot,1}}{N_{tot,2}}X_{CO_2,1}) \qquad (4.1)
\end{aligned}$$

Since the total amount of molecules is unknown we need to estimate the fraction $\frac{N_{tot,1}}{N_{tot,2}}$. The largest relative change in molecule amount is if we run at low $\lambda$ without EGR. In the following calculations it is assumed that $A$ molecules of C$_1$H$_{1.85}$ are burnt in fresh air at $\lambda$ = 1. The combustion is assumed to be perfect.

$$\begin{aligned}
&A \cdot C_1H_{1.85} + B \cdot O_2 + 3.773B \cdot N_2 \rightarrow \\
&A \cdot CO_2 + \frac{1.85}{2}A \cdot H_2O + 3.773B \cdot N_2 \Rightarrow \\
\Rightarrow &B = A + \frac{1.85}{2 \cdot 2}A = \frac{5.85}{4}A \Rightarrow \\
\Rightarrow &N_{tot,1} = 4.773\frac{5.85}{4}A; N_{tot,2} = (1 + \frac{1.85}{2} + 3.773\frac{5.85}{4})A \Rightarrow \\
\Rightarrow &\frac{N_{tot,1}}{N_{tot,2}} = 0.94
\end{aligned}$$

This is an underestimation since we know that we run the engine with EGR in most cases and always have $\lambda > 1$. Therefore we don't introduce any large errors if we approximate equation 4.1 with this equation:

$$X_{H_2Oout} - X_{H_2Oin} - X_{H_2Oliquid} = \frac{1.85}{2} \cdot (X_{CO_2out} - X_{CO_2in}) \qquad (4.2)$$

This equation obviously doesn't have any problems with numerical stability and when the calculations of the amount of liquid water into each cylinder, according to equation 4.2, are compared to those of the equation system in section 2.4, it should show whether the negative values is caused by measurement errors or if they are caused by a sensitive equation system. If the two different sets of values agree that there is negative amounts of liquid water entering some cylinders, the errors are in the measurement data itself, otherwise it's more likely numerical instability that is causing the problems.

When the difference between the two sets of results is studied, it can be seen that it keeps within a narrow region (-0.06 to 0.13 percentage points), which shows that it is in fact measurement errors or -uncertainties that cause the strange results.

Even though the results seem to be untrustworthy, they might provide useful information if studied in the right way and by using caution when coming to conclusions. If it is assumed that the variation is somehow related to which operating condition the engine is ran at, it might help to study the cylinder-to-cylinder variation for each operating point. Here it is plotted as a surface, where the shading is determined by the deviation from the mean fraction value for each operating point.

The plot of liquid water distribution (figure 4.1) shows that there seems to be more liquid water coming into the first cylinders in the majority of the operating points. The $CO_2$ fractions in to the cylinders (figure 4.2) differ within a +/- 0.2 vol% interval, while the $H_2O$ fractions in to the cylinders (figure 4.3) vary with +/- 1 vol% between the cylinders within the same operating condition, implying that the variation in water distribution is mainly due to the behaviour of the liquid water, since the variation in water distribution is comparatively large. The small variation of the distribution of $CO_2$ show us that the mixing of gases in the inlet manifold is working satisfactory.

Since the $CO_2$ fraction out from the cylinders (figure 4.4) seems to vary more than the intake manifold values, it points towards injectors that deliver different amounts of fuel, it could possibly depend on varying volumetric efficiency between the cylinders. That effect is not expected to be very big in comparison though.

To further try to conclude how the water is distributed, the NO fraction out from the cylinders is studied (figure 4.5), the surface shows that it seems to be more NO created in cylinder 4 especially, this is interesting because that cylinder also have a higher share of $CO_2$ in the exhaust manifold (figure 4.4). It does not, however, seem to have more water coming out (figure 4.6). This could be explained by an injector that gives a slightly higher amount of fuel

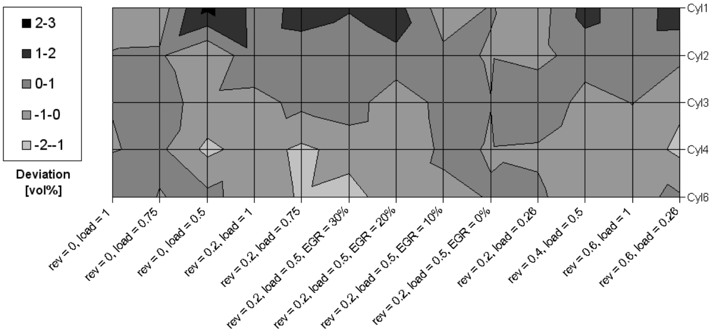

Figure 4.1: Deviation from operating point average in liquid water fraction in to the cylinders

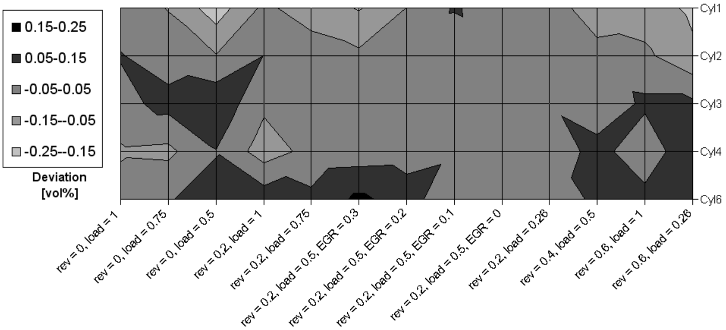

Figure 4.2: Deviation from operating point average in $CO_2$ fraction in to the cylinders

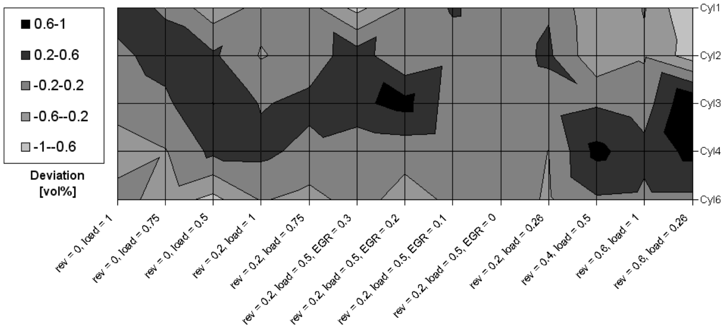

Figure 4.3: Deviation from operating point average in water vapor fraction in to the cylinders

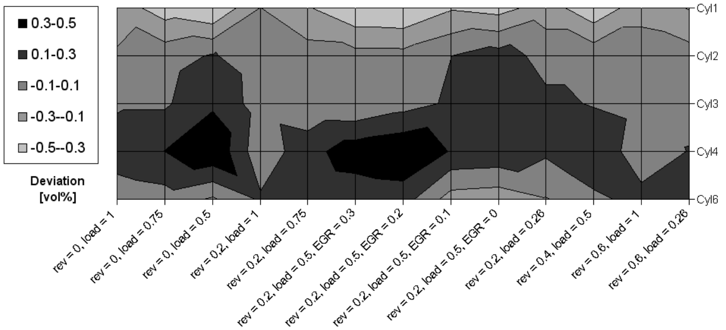

Figure 4.4: Deviation from operating point average in $CO_2$ fraction out from the cylinders

in cylinder four, raising the $CO_2$-levels out from the cylinder. At the same time, it seemed to receive slightly less of liquid water, which would explain why the increase in $CO_2$ out is not accompanied by a higher amount of water vapor out. The increase in NO formation would originate from a higher local temperature even though the extra amount of fuel lower the $\lambda$ value in the combustion chamber. If the operating points with extreme emission values are studied (12 - 20 % and -20 - -12 %) it seems as if the emission formation is a combined effect of a higher amount of fuel as well as a distribution of liquid water that is not uniform but some percent point higher in cylinder 4 and some percent point lower in cylinder 1.

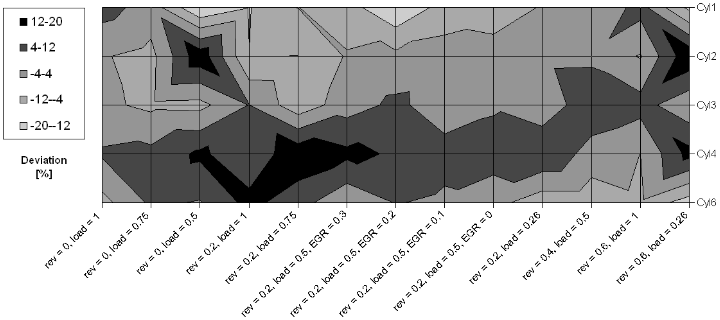

Figure 4.5: Deviation from operating point average in NO fraction out from the cylinders

### 4.1.1 Result

The analysis seem to suggest that there are generally more liquid water entering the first cylinders than the last. It is supported by both the calculations of liquid water directly but also by the distribution of $CO_2$ in the exhaust manifold. Since the deviation of $CO_2$ in the exhaust is bigger than the deviation of $CO_2$ in the inlet manifold and because its distribution is not correlating with the increase in water vapor, it suggests that there are less liquid water entering the last cylinders.

The deviation in liquid water distribution is likely to be a few percentage units, which is a rather large spread. At the same time, the deviation when it comes to $CO_2$ is just a few tenths of a percentage unit, which implies that the construction of the inlet manifold is good when it comes to evenly distribute gas.

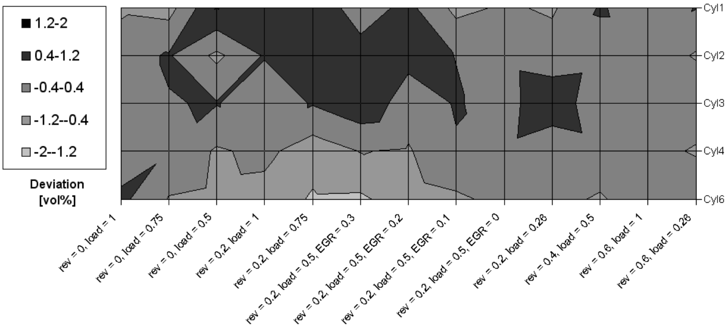

Figure 4.6: Deviation from operating point average in water vapor fraction out from the cylinders

The biggest effect on the $NO_x$ emissions seem to originate from a cylinder-to-cylinder deviation in injected amount of fuel and from a skew distribution of liquid water. As far as distribution of gaseous EGR goes, the deviation seems to be on par with what's possible to reliably measure using the Gasmet instrument and this test setup. It is hard to point directly at the phenomenons when the measurement data displays obvious imperfections, most of the data seem to point in the directions just mentioned, however.

## 4.2 Dynamics

The dynamic sequence consisted of a step in EGR-fraction from nominal level to zero per cent by using a valve that can be closed to eliminate exhaust gas recirculation. Closing of the valve takes a couple of hundred milliseconds, so that dynamics is not considered in the analysis. During the transient, $CO_2$- and $H_2O$-fractions were measured with 1/6Hz and the temperature of the inlet ports where sampled with 1Hz. The placement of the different temperature sensors is displayed in figure 3.2.

Data for the Pt-100 temperature sensors implied that their response time was a few seconds. To verify this, a step from no EGR to nominal EGR was studied, in this case there should be no liquid water present in the inlet manifold which could disturb the measurements. In this way it is possible to verify if there is a big difference in responsiveness due to the different massflows.

Figure 4.7 shows that there is an initial drop in temperature, which is likely due to the cold EGR heat-exchangers. After that there is a rise in temperature as the turbo-charger changes its working point and hot exhaust is starting to circulate. No big variation caused by the different gas mass flows can be seen, which suggests that the dynamics of the sensors are fast enough for the experiment.

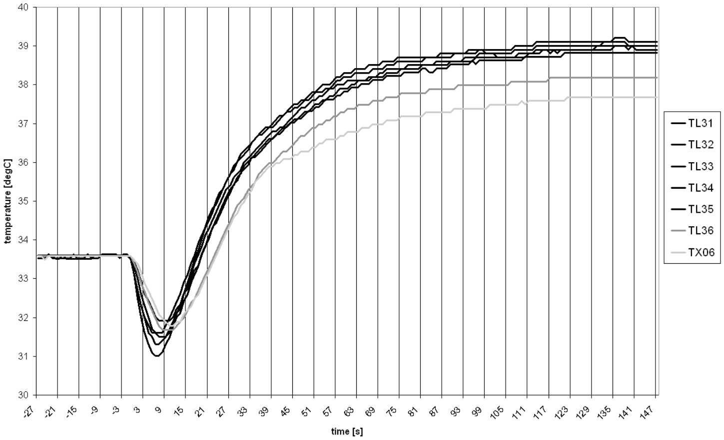

Figure 4.7: Temperatures in inlet manifold during step from no EGR to nominal EGR level, values sampled at 1 Hz and normalized

Six identical steps in EGR was made for the same operating condition, at each one of them, gas from the intake port of one cylinder was sampled. If the $CO_2$-fraction is plotted (figure 4.8) the speed of the sampling equipment is visible, since the $CO_2$-fraction should change almost instantly, and can be used when studying the different $H_2O$-fractions. A quick look at the $H_2O$-fractions compared to the average behavior of the $CO_2$-fractions (see figure 4.9) implies that there is both a larger variation in $H_2O$-fraction before the step as well as a slower decay, a decay that seems to get a little bit slower the further in to the inlet manifold the gas is sampled.

If the temperatures are studied for the same case as above, with the beginning temperatures normalized (figure 4.10), the appearance can be explained by the following reasoning: The initial drop in temperature is due to the fact that the EGR gas after the air-to-air heat exchanger still is warmer than the fresh air from the inter-cooler. When the EGR is shut off, the flow through the inter-cooler is increased and the turbo-charger changes operating point, both resulting in the air after the inter-cooler getting warmer. What is not explained is why there is such a large deviation between the different temperature sensors. As could be seen in figure 4.7, mere location doesn't cause

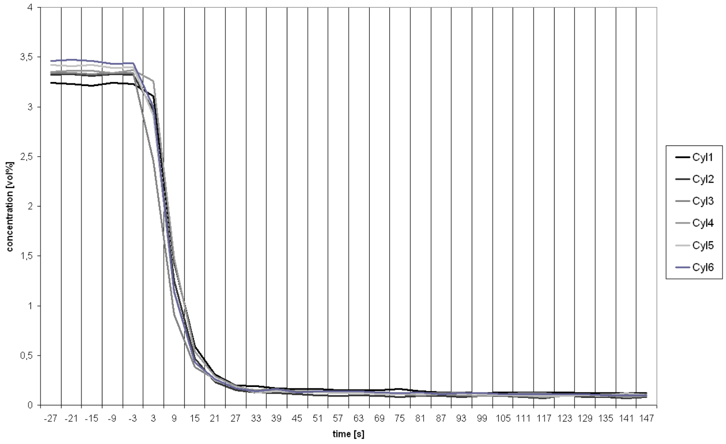

Figure 4.8: $CO_2$ fraction when going from nominal EGR level to no EGR, values sampled at 1/6 Hz

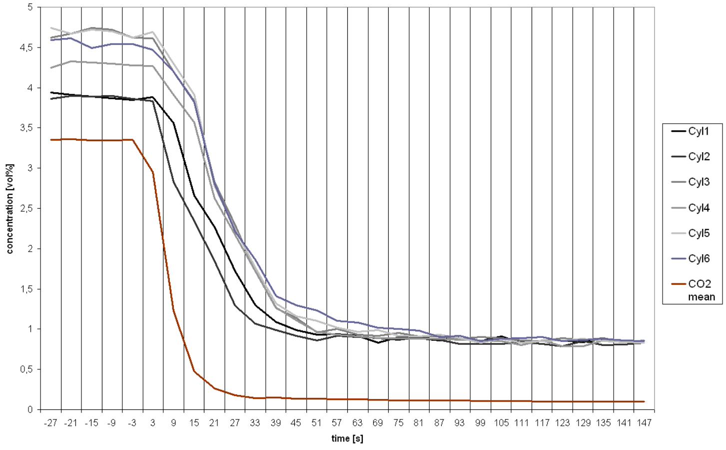

Figure 4.9: $H_2O$ and the average $CO_2$ fraction when going from nominal EGR level to no EGR, values sampled at 1/6 Hz

more than a few seconds delay due to different gas mass flows. The theory is that the temperature difference between TL31 and the following temperature sensors is a result of water evaporating, either from puddles and wet walls of the inlet manifold, or by droplets on the sensors themselves.

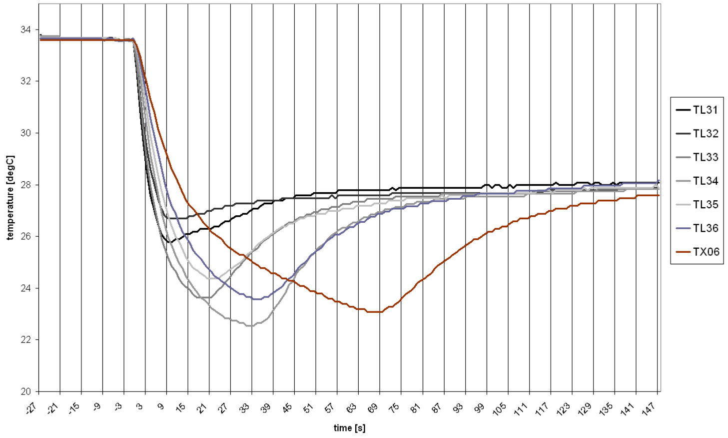

Figure 4.10: Temperatures in inlet manifold during step from nominal EGR level to no EGR, values sampled at 1 Hz and normalized

To quantify this phenomenon, both the temperature differences and the delay of the $H_2O$-fractions settling can be integrated and used to estimate the corresponding amount of water needed to explain the behavior.

To calculate the amount of water from the temperature difference the first step is assuming that TL31 is showing the “real” temperature without the influence of water, after that the difference between the temperature sensors can be calculated and together with the measured air mass flow the energy needed to lower the temperature during a period of time estimated from the temperature appearance is determined. If all the energy is assumed to come from the water, the corresponding mass of liquid water is given by simply comparing the specific heat of air with the heat of vaporization of water.

$$m_{H_2O} = \int (T_{i+1} - T_i) \cdot \dot{m}_{AIR} \cdot \frac{c_{p,H_2O}}{L_{H_2O}} dt \tag{4.3}$$

The same analysis can be made based on the concentration measurements, although here we obviously use another equation.

$$m_{H_2O} = \int \Delta X_{H_2O} \cdot \dot{m}_{AIR} \cdot \frac{M_{H_2O}}{M_{AIR}} dt \qquad (4.4)$$

The results of the calculations are displayed in the following table (table 4.1).

| Location of the water | Amount of water (temperature) | Amount of water (fraction) |
|---|---|---|
| In front of cyl. 1 | 8.49 g | 54.26 g<br>(also before cyl. 1) |
| In front of cyl. 2 | 0 g | 16.10 g |
| In front of cyl. 3 | 5.73 g | 0 g |
| In front of cyl. 4 | 0 g | 18.6 g |
| In front of cyl. 5 | 1.43 g | 1.07 g |
| In front of cyl. 6 | 1.82 g | 0.59 g |

Table 4.1: Location and amount of water calculated from temperature and fraction measurements in inlet manifold

It can be seen that the two kinds of measurement isn't complying about the exact location of the water. What they do agree about is that the amount is diminishing further into the inlet manifold, they also show values of approximately the same order of magnitude, spanning from single grams to tenths of grams.

As mentioned in section 3.4, some additional transient measurements was done without gas sampling. These temperature measurements are evaluated in the same manner as above and the results are displayed both in tabular form and as a surface plot (table 4.2 and figure 4.11). Even though the variation is percentually very large, the results seem to be somewhat consistent with what we saw in the previous data.

| rev. index<br>load index | -0.1<br>0.5 | 0.1<br>0.5 | 0.1<br>1 | 0.3<br>1 | 0.7<br>1 |
|---|---|---|---|---|---|
| In front of cyl. 1 & 2 | 3.38 | 2.64 | 10.05 | 1.57 | 0 |
| In front of cyl. 3 | 7.22 | 3.97 | 0.01 | 3.43 | 0 |
| In front of cyl. 4 | 1.86 | 0 | 0.11 | 0 | 0 |
| In front of cyl. 5 | 0.42 | 0.77 | 0.74 | 0.02 | 0 |
| In front of cyl. 6 | 0.55 | 1.50 | 1.68 | 1.46 | 1.14 |

Table 4.2: Location and amount of water calculated from temperature measurements in inlet manifold

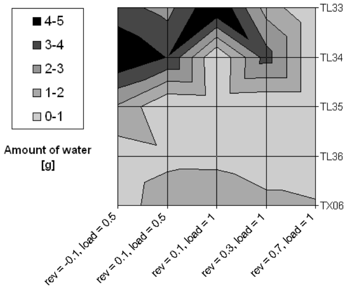

Figure 4.11: Amount and location of water calculated from temperature measurements

### 4.2.1 Result

From studying the measurements it is clear that liquid water exists stationary in the inlet manifold. It is also obvious that there are some sort of puddles present, and not just liquid water as aerosol. This is apparent when considering the amount of time it takes for the $H_2O$ fractions and temperatures to stabilize after a step in EGR fraction.

The two different attempts to quantify the phenomenon does not agree completely regarding the amounts and placement of the water but both suggests that the water puddles are mainly present in the beginning of the manifold and that the amount per cylinder is in the tenths of grams.

# Chapter 5

# Conclusions and future work

## 5.1 Conclusions

The work shows that there are ways to measure liquid water presence in the inlet manifold when using large amounts of cooled EGR. To accurately and with certainty quantify the exact amounts and with good precision determine the location of the water was shown to be the major difficulty. The results further show that the water form “puddles” that can take as long as two minutes to dry if the engine is run without recirculating humid exhaust gases.

All results seem to agree that the water is mainly localized in the beginning of the inlet manifold and that evaporating water increase the fraction of water vapor further down the manifold.

It was shown that the distribution of gaseous EGR works good. Measurements of cylinder out emissions, mainly NO and $NO_2$, suggest that the effect on emissions from uneven water distribution can be comparatively large and that also differences in fuel distribution is likely to have a greater impact on emission formation than that of gaseous EGR distribution.

In this thesis it is shown that the best way for measure differences in distribution is to actively measure the species of interest. It is also concluded that if faster response times are demanded i.e. for transients in EGR-fraction or engine load, temperature measurements are an option that in some way enables quantification possibilities.

Finally, it is shown that the impact of liquid water distribution on emission formation should be taken into account when tomorrows emissions legislation is to be met but also that the current way of mixing and distributing the inlet gas is working satisfactory.

## 5.2 Future work

Much of the data collected here is uncertain and it would be wise to use familiar equipment and to devote more time to verify that reliable results can be guaranteed.

Using a valve system with valves that was not developed for the task did not cause any major problems and it seem to be good and cost-effective alternative when the utilizing time does not exceed a few weeks.

Measurements in the exhaust channels were far more stable than measurements in the relatively cold gases in the inlet manifold. If fuel delivery can be controlled more accurately so that that source of uneven distribution of emissions can be reduced, it could be an option to directly study emission formation from individual cylinders, if the impact of engine operation should be more thoroughly investigated.

One idea is to run the test at an one-cylinder setup and manually add liquid water into the manifold in a controlled manner to see which amounts are needed to make a considerable impact on emissions.

For the calculations regarding water condensation it was assumed that the cooled EGR held the same temperature as the charge air mixture, but when the temperature readings of the transient steps are analysed it is clear that the EGR is significantly warmer. This should be taken into consideration when planning potential future investigations. Separate temperature measurements of the fresh air and EGR before the point of mixing would be helpful when evaluating which operating conditions are associated with large fractions of liquid water.

The calculations of volumetric efficiency that are done in the engine control system today compensates in some way for liquid water. Using the knowledge that was obtained during this thesis together with the additional temperature measurements mentioned above, it should be possible to improve that model.

# References

[1] M. Andersson. Fast nox prediction in diesel engines, September 2006.

[2] GASMET. Faq on http://www.gasmet.fi/. internet, 2006.

[3] GASMET. Gasmet dx-4000 technical data. internet, pdf, 2006. version 1.

[4] G. Hellsten. *Tabeller och diagram*. Almqvist & Wiksell Förlag AB, 1 edition, 1992.

[5] Ltd HORIBA. Instruction manual for horiba non-sampling type nox-a/f analyzer mexa-720nox, 2003.

[6] C. Künkel. *Catalytic Reduction of NOx on Heavy-Duty Trucks*. Phd thesis, Department of Chemical Engineering II, Lund University, Lund, Sweden, September 2001.

[7] C. Nordling and J. Österman. *Physics Handbook for Science and Engineering*. Studentlitteratur, Lund, Sweden, 6 edition, 1999.

[8] O. Stenlåås. *On the Impact of Piston Motion and In-Cylinder Charge Composition on Energy Release, Auto Ignition and Emission Formation in Premixed Charge Internal Combustion Engines*. Phd thesis, Department of Heat and Power Engineering, Lund Institute of Technology, Lund, Sweden, 2004.

# Notation

Symbols and abbreviations used in the report.

## Variables and parameters

| | |
|---|---|
| $c_p$ | Specific heat capacity (at constant pressure) |
| $L_*$ | Heat of vaporization |
| $M_*$ | Mole weight |
| $N_*$ | Amount of substance |
| $X_*$ | Mole fraction |

## Abbreviations

| | |
|---|---|
| AMU | Atomic mass units |
| EGR | Exhaust gas recirculation |
| HC | Hydro carbons |
| PM | Particulate matter |
| SCR | Selective catalytic reduction |
| TL3* | Temperature sensor in inlet manifold |
| TX06 | Temperature sensor in inlet manifold |