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Abstract:

The goal of this M.Sc. thesis has been to study the NOx-reduction measure: water in fuel emulsion. There has been done a literature study to look into the function, system components and effect upon NOx reduction, specific fuel oil consumption and other emissions.

The operational aspects of water in fuel emulsion have been looked into. It has been found that running on water in fuel emulsion does not have many operational problems. However there is a need for having a "fail safe" system that can shut down the water supply if the engine has a failure and if the dosage system fails to give the right amount of water. This has to be done in order to prevent the separation of water and fuel that will happen if the emulsion is stationary for too long.

The possibility of using the engine simulation program GT-Power in order to examine NOx emissions and specific fuel oil consumption has been investigated. For this there has been used a Rolls Royce KRG-6 engine model. The results of these investigations are a number of modifications that needs to be done to the engine model in order to achieve reasonable results. It has also been found that there is a need for more specific engine data if one wants to achieve good results. However reasonable results for relative NOx- reduction have been found. This supports the idea of GT-Power being suitable for this kind of simulations.

Engine runs in the lab at MARINTEK, running the KR-3 engine on white diesel have been performed and the results from these test are that the emissions of NOx is reduced, but not nearly as much as expected. Particulate matter measurements showed a large increase in the size distribution around diameters of 20nm. This is the size of particles in the nucleation mode. The result for filter smoke number is a very large reduction; this supports the theory that soot in the exhaust gas is reduced when introducing water to the combustion chamber. However looking at the particle size distribution does not support this theory. These measurements show that the particle number within the range of accumulation mode particles (80nm) is about the same. This raises a question about how reliable the smoke number is when determining the contents of soot in the exhaust gas.

Keyword:

NOx reduction
Water in fuel emulsion
PM emissions

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MASTER THESIS
for
Stud.techn. Ragnar Dystvold Landet
Spring 2010

PM emissions and NO_x–reduction due to water in fuel emulsions in marine diesel engines

PM utslipp og NO_x-reduksjon ved hjelp av vanntilsetning i brennoljen på marine dieselmotorer

Background

While emissions from land based sources in many cases have been reduced in recent years, emissions of NOx, SOx, and PM from ships has increased. International shipping now contributes to somewhere around 15% of global NOx emissions, and there is a substantial pressure to reduce NOx emissions from ships. EU and USA have both established air quality standards with established maximum levels of fine particles. These levels are exceeded in many coastal and harbour areas, and reductions of PM emissions from ships is an important objective. Nevertheless, while NOx emissions from ships is regulated and more stringent NOx emission limits are forthcoming, emissions of PM from ships is not likely to be regulated.

It is widely accepted that many NOx reduction measures will result in increased levels of other emissions. In particular, reductions of NOx may result in increased emissions of CO2, CO, unburned hydrocarbons and particulates. The magnitude of the effect on CO2 is fairly well established, however the consequence of NOx reduction measures on the various aspects of PM emissions form ships is largely unknown.

To get a better understanding of the effect of PM emissions from ships these emissions must be more accurately characterised and more refined emissions factors need be developed. This will enable more accurate emissions modelling and health impact studies in general and also allow different NOx reduction technologies to be modelled separately.

Adding water to the fuel in terms of water-oil emulsions has been demonstrated reduced NO_x emissions. This assignment has main emphasis on the water-in-oil emulsions and their affects on exhaust emissions.

Overall Aim and Focus

The overall objective of the task is to get an understanding of the impact of water-fuel emulsion on PM and NOx emissions from marine diesel engines.

The assignment should be prepared based on following points:

- Review literature on NOx and PM emissions from marine diesel engines.
- Discuss current methods for NOx emission reduction by adding water.
- Discuss application of water-fuel emulsions, expected influence on NOx and PM emissions, and special considerations concerning engine operation with water-fuel emulsions.
- Investigate the effect of water-fuel emulsion by means of engine performance simulations using GT-POWER.
- Perform engine experiments in the Machinery laboratory by using "White diesel" fuel in a four stroke medium speed diesel engine. Measurement of engine performance and emissions of NOx and PM.
- Discuss experimental results and simulation results.

The project will be performed in cooperation with MARINTEK, where senior engineer Erik Hennie is his contact person.

The MSc assignment text should be included in the report.

The report should be written like a research report, with an abstract, conclusions, contents list, reference list, etc.

During preparation of the report it is important that the candidate emphasizes easily understood and well written text. For ease of reading the report should contain adequate references at appropriate places to related text, tables and figures. On evaluation, a lot of weight is put on thorough preparation of results, their clear presentation in the form of tables and/or graphs, and on comprehensive discussion.

All used sources must be completely documented. For textbooks and periodicals, author, title, year, page number and eventually figure number must be specified.

In accordance with current regulations NTNU reserves the right to use any results from the project work in connection with teaching, research, publication, or other activities.

Three -3- copies of the report are required. One complete copy of all material on digital form on CD-ROM in Word-format or other relevant format should be handed in together with the written material.

The project report must be delivered to Department of Marine Technology no later than June 14 2010.

Trondheim, January 21 2010

Harald Valland
professor

Preface

This report is the results of my M.Sc. thesis work as a part of NTNUs Master of Science program in Marine Technology. The laboratory part of the thesis was done in cooperation with MARINTEK.

MARINTEK has been running engine tests to measure particulate matter emissions for over a year. They are testing to see how different NOx reduction measures affect particulate matter emissions. In order to measure particle matter they have two different measurement instruments capable of giving particulate matter size distribution. The different instruments are a DEKATI ELPI and a TSI SMPS.

The goal of my master thesis work has been to look into how running marine diesel engines on water in fuel emulsion affect the emissions of NOx and particulate matter. I have been working on two main tasks. I have used an engine simulation program called GT-POWER in order to check if water in fuel emulsion could be implemented to the program. And if there could be achieved reasonable results for NOx emissions. The second task has been to analyze results from engine tests done at MARINTEK. The KR3 engine at the engine lab has been run on white diesel to measure emissions. The two interests here has been to see how this affects the NOx and particulate matter emissions from the engine.

I would like to thank my supervisor at Institute for Marine Technology Harald Valland and my supervisor at MARINTEK Erik Hennie for their support and help during my work. I would also like to thank PH.D student Nabil Al Ryati for help with GT-POWER and Jørgen B. Nielsen from MARINTEK for his help with the particulate matter measurements.

Trondheim 21. June

Ragnar Dystvold Landet

List of abbreviations

WFE	Water in fuel emulsion
DWI	Direct water injection
SAM	Scavenging air moistening
BMEP	Brake mean effective pressure
IMO	International maritime organization
NECA	NOx emission control area
DI	Direct injection
CFD	Computational fluid dynamics
EGR	Exhaust gas recycling
SCR	Selective catalytic reduction
SFOC	Specific fuel oil consumption
BSFC	Brake specific fuel consumption
MGO	Marine gas oil
PM	Particulate matter
FSN	Filter smoke number
DEKATI	Manufacturer of particle measurement instruments
TSI	Manufacturer of particle measurement instruments
ELPI	Electrical low pressure impactor
FPS	Fine particle sampler
SMPS	Scanning mobility particle sizer
DMA	Differential mobility analyzer
CPC	Condensation particle counter

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1 Introduction

In recent years emissions of NOx, SOx and particulate matter from ships have increased. International shipping now contributes to about 15% of the global NOx emissions and there is a substantial pressure to reduce NOx emissions from ships. EU and USA have established air quality standards with maximum levels of fine particles. These levels are exceeded in many costal and harbour areas. However it is not likely that particulate matter emissions from ships will be regulated, but it is still an important objective to minimize these emissions. NOx emissions from ships are regulated and the limits on NOx emission are getting stricter.

The limitation on NOx emission means that different measures for NOx reduction will have to be used. These measures might have adverse effects upon the levels of other emissions like CO2, CO, unburned hydrocarbons and particulate matter. The magnitude of the effect upon CO2 is widely known; while the effects upon particulate matter are largely unknown.

The aim of this master thesis is to look at the effect water in fuel emulsion have upon emissions and specific fuel oil consumption. The emissions that I have looked into have been NOx and particulate matter. The effects of WFE on NOx emissions are widely known, while the effects upon particulate have not been documented to the same extent.

This first chapter of my master thesis is more or less information gather during my work with the project thesis. It consists of information on why ship owners want to reduce NOx emissions. How NOx emissions form and what the effect of introducing water into the combustion chamber has on the formation process. I have looked into the effects upon soot in the exhaust gas. Different systems for introducing water into the combustion chamber and there has been given a general description on a WFE system and its different parts. There has also been looked into what effect WFE has on the emission of NOx and what effect this has on the specific fuel oil consumption of the engine.

I have looked into operational aspects of running on WFE and have in this connection been in contact with Magnus B. Småvik from MOTORCONSULT AS. He gave me information on the operational and maintenance aspects of their WFE system.

GT-POWER has been used to look into modelling of adding water to the fuel in order to investigate the effects on NOx emission and specific fuel oil consumption. In this work I have used a model of a Rolls Royce KRG-6 engine that I received from Nabil Al Ryati a PH.D student at NTNU department of marine technology.

To study the effects water has on the particulate emissions in diesel engines, the KR-3 engine at MARINTEK lab has been run on white diesel.

1.1 NOx emissions from diesel engines

NOx emissions from diesel engines consist of different nitrogen oxides. Nitric oxide (NO), Nitrogen Dioxide (NO2) and nitrous oxide. From these three gases it is nitric oxide which is the largest contributor to the emission while nitrogen dioxide contributes to about 5% of the emissions and nitrous oxide contributes to about 1%.

1.2 Reasons to reduce NOx emissions

There are three main reasons to reduce NOx emission from internal combustion engines. These reasons are the health hazards and the environmental impact from the emissions. But ultimately regulations and taxes are the reason why ship owners will reduce the emissions.

1.2.1 Health Hazards

NO_X molecules in the atmosphere react with ammonia, moisture and other compounds to form nitric acid vapour and particles. Small particles penetrate deeply into lung tissue and damage it. NO_X also react with volatile organic compounds in the presence of heat and sunlight to form Ozone. Ozone can damage lung tissue and reduce lung function and is also a contributor to the formation of smog.

1.2.2 Environmental impact

N_2O which is a small amount of the NO_X emissions destroys the ozone layer. While nitric acid along with sulphuric acid is the cause of acid rain which has a harmful effect on plants, aquatic animals and infrastructure.

So there is no doubt that reducing NO_X emissions is an important task in keeping the population and environment healthy. Reduced emissions also have positive effect on the ship owner's economy as NO_X taxes are expensive.

1.2.3 Rules and regulations

NO_X reduction is getting more and more important as regulations are getting stricter. The limits set by Imo on NO_X emissions is shown in Table 1 and graphically in Figure 1. Tier 1 is the current emission limit set by IMO annex VI regulation 13. This regulation applies to each diesel engine with a power output of more than 130kW which is installed on a on a ship constructed on or after 1 January 2000 (IMO, 2005). Also engines with a power output of more than 130 kW which undergoes a major conversion on or after 1 January 2000 has to comply with this regulation.

Tier II is to be effective from the 1 January 2011 and is a global limit. Tier III is a more massive reduction of NOx emissions and is only meant to apply to ships sailing within special NOx emission control areas (NECA).

		NOx Limit, g/kWh		
Tier	Date	n<130	130<n<2000	n>2000
Tier I	2000	17	45*n^-0.2	9.8
Tier II	2011	14.4	44*n^-0.23	7.7
Tier III	2016	3.4	9*n^-0.2	1.96

Table 1: IMO NOx limits.

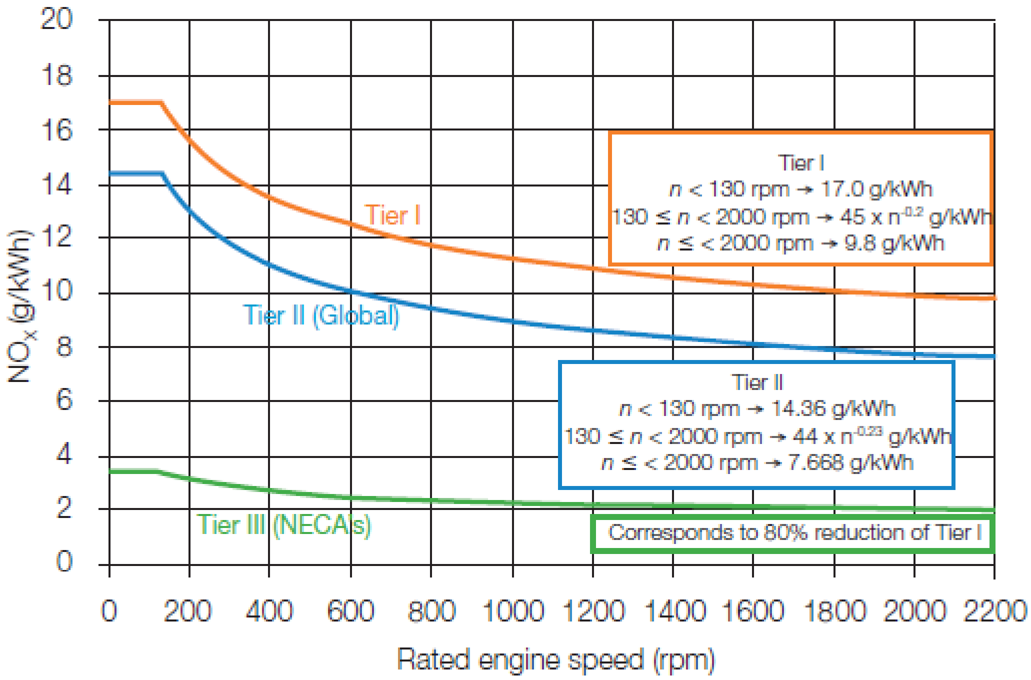

Figure 1: Imo limits on NOX emissions (Man B&W, 2008).

In addition to the limits set by IMO there is local limits and taxes many places in the world. In Norway there is there is a NOx emission tax where the ship owner is taxed by how much NOx is emitted pr. year. If the engine is delivered after 1 January 2000, it will have an EIAPP certificate and the specific NOx emission will be given in the certificate. If not there need to be certified personnel onboard to measure the specific NOx emissions of the engine(s).

The annual NOx emissions are calculated by using a NOx factor. The calculation of the NOx factor can be seen in Equation 1. To calculate the annual NOx emission the NOx factor is simply multiplied with the annual fuel consumption. If the vessel uses a SCR installation the calculation will be a bit more complicated. The NOx tax is 16.14 NOK per kg NOx emitted (The Norwegian Customs Service, 2010).

$$F = \frac{SpecificNOxemission * 1000}{SpecificFuelConsumption}\left[\frac{kgNOx}{tonFuel}\right]$$

Equation 1: NOx factor F (Norwegian Maritime Directorate, 2010).

1.3 NOx formation in diesel engines

To understand how WFE helps reduce NO_X emissions in a diesel engine we have to take a look at how NO_X forms in the combustion chamber.

1.3.1 NO_X formation

The formation of NO_X in a combustion engine is highly dependent on local temperatures inside the combustion chamber. NO and NO_2 are usually grouped together under the term NO_X, while NO is the predominant oxide of nitrogen produced inside the engine cylinder. “The principal source of NO is the oxidation of atmospheric (molecular) nitrogen.”(Heywood, 1988). So the NO_X emissions come mainly from NO formation in the engine cylinder. Later in the process, during expansion and in the exhaust system, part of the NO will convert to form NO_2 and N_2O, typically 5% and 1%, respectively, of the original NO amount (Man B&W, 2003).

Further (Heywood, 1988) chapter 11.2.1 has this to say about NO formation: “NO forms in both the flame front and the post flame gases. In engines, however, combustion occurs at high pressure so the flame reaction zone is extremely thin ($\approx$0.1 mm) and residence time within this zone is short. Also, the cylinder pressure rises during most of the combustion process, so burned gases produced early in the combustion process are compressed to a higher temperature than they reach immediately after combustion. Thus, NO formation in the post flame gases almost always dominates any flame-front produced NO. It is, therefore, appropriate to assume that the combustion and NO formation processes are decoupled and to approximate the concentrations of O, O2, OH, H, and N2 by their equilibrium values at the local pressure and equilibrium temperature. “

In chapter 11.2.4 of the same book says that the most critical time for NO formation is when burned gas temperatures are at a maximum. This is between the start of combustion and shortly after maximum cylinder pressure. After maximum cylinder pressure the temperature decreases as expansion work in preformed by the cylinder gas. The formation rate of NO is directly coupled with the temperature as shown in Equation 2. Without going into too much detail this equation shows us that the formation of NO needs a high temperature and that the formation rate increases fast with increasing temperature. This is due to the high activation energy needed for the Zeldovich mechanism shown in Equation 3.

$$\frac{d[NO]}{dt} = \frac{6\times10^{16}}{T^{1/2}}\exp\left(\frac{-69090}{T}\right)[O_2]_e^{1/2}[N_2]_e$$

Equation 2: Initial NO formation rate (Heywood, 1988).

$$\begin{aligned} O + N_2 &\rightarrow NO + N \\ N + O_2 &\rightarrow NO + O \\ N + OH &\rightarrow NO + H \end{aligned}$$

Equation 3: Zeldovich mechanism relations (Heywood, 1988).

According to figure 16-8 in the compendium Diesel engines 4 written by D. Stapersma, the NO formation in the cylinder is virtually zero with temperatures below 2000°C. When the temperature rises above 2000°C the NO formation will have a rapid growth.

1.3.2 The effects of introducing water in the engine cylinder

The waters function in decreasing NO_X emissions is related to two phenomena. The first is that vaporisation of the injected water droplets will decrease the internal energy proportionally to the vaporisation enthalpy of the water. The second is that specific heat of the cylinder gas will also increase with increasing water content. These two phenomena lead to lower temperatures where the WFE has been sprayed. Since fuel and water is sprayed together this means that the temperature in the post flame gases, that earlier in this report were identified as the main area where NO is formed, will decrease. As NO formation increases rapidly with increasing temperature above 2000°C, this means that less NO will be formed. Equation 4 shows that energy is drawn from the cylinder gas (environment) to evaporate the water droplets.

$$dQ = -m_w * h_{w-vap}$$
$$dU = dQ - dW$$
$$dU = -m_w * h_{w-vap}$$

Equation 4: Internal energy changing with water vaporization.

Equation 5 shows that as the specific heat of the gas is increased more energy is needed to increase the temperature. Or said with other words: for a given transfer of heat to the gas, temperature will not increase as much if the specific heat is higher.

$$Q = Cp_g * m_g * \Delta T$$

Equation 5: Energy for gas heating.

As a rule of thumb a change of 100°C in the combustion temperature may change the NO_X formation by a factor of 3 (Man B&W, 2003).

1.3.3 Micro explosion

According to a SAE paper (Chadderton, 2000) a phenomenon called micro-explosion occurs when water droplets in WFE instantaneously vaporizes within fuel droplets as the fuel is exposed to increasing cylinder temperatures during injection. This happens when the temperature of the fuel droplets increase above the boiling point of water. The water will quickly and violently evaporate; this will break the fuel droplets into smaller droplets, which results in a more complete vaporization and turbulent mixing of the fuel. Figure 2 shows this effect. As the fuel atomization is enhanced, it will lead to a selective reduction of insoluble carbon fractions of the particulates.

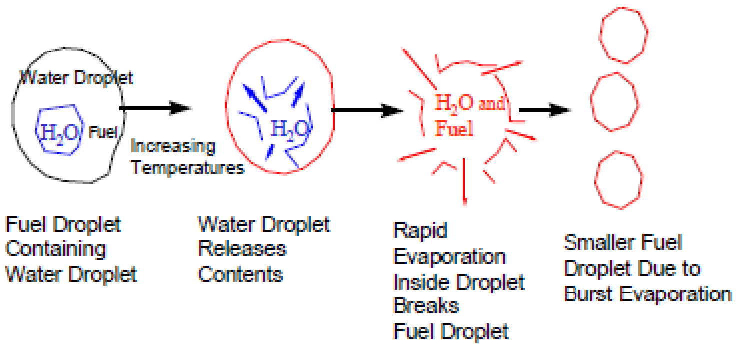

Figure 2: Micro-explosion phenomenon in diesel fuel continuous emulsions (Chadderton, 2000).

1.3.4 Soot

A study with an optically-accessible DI diesel engine (Ki Hoon Song, 2000) showed that the addition of water in diesel creates a delay in the start of combustion and a reduction in the amount of soot as the water content is increased. The combustion images with 1 and 20% water can be seen in Figure 3.

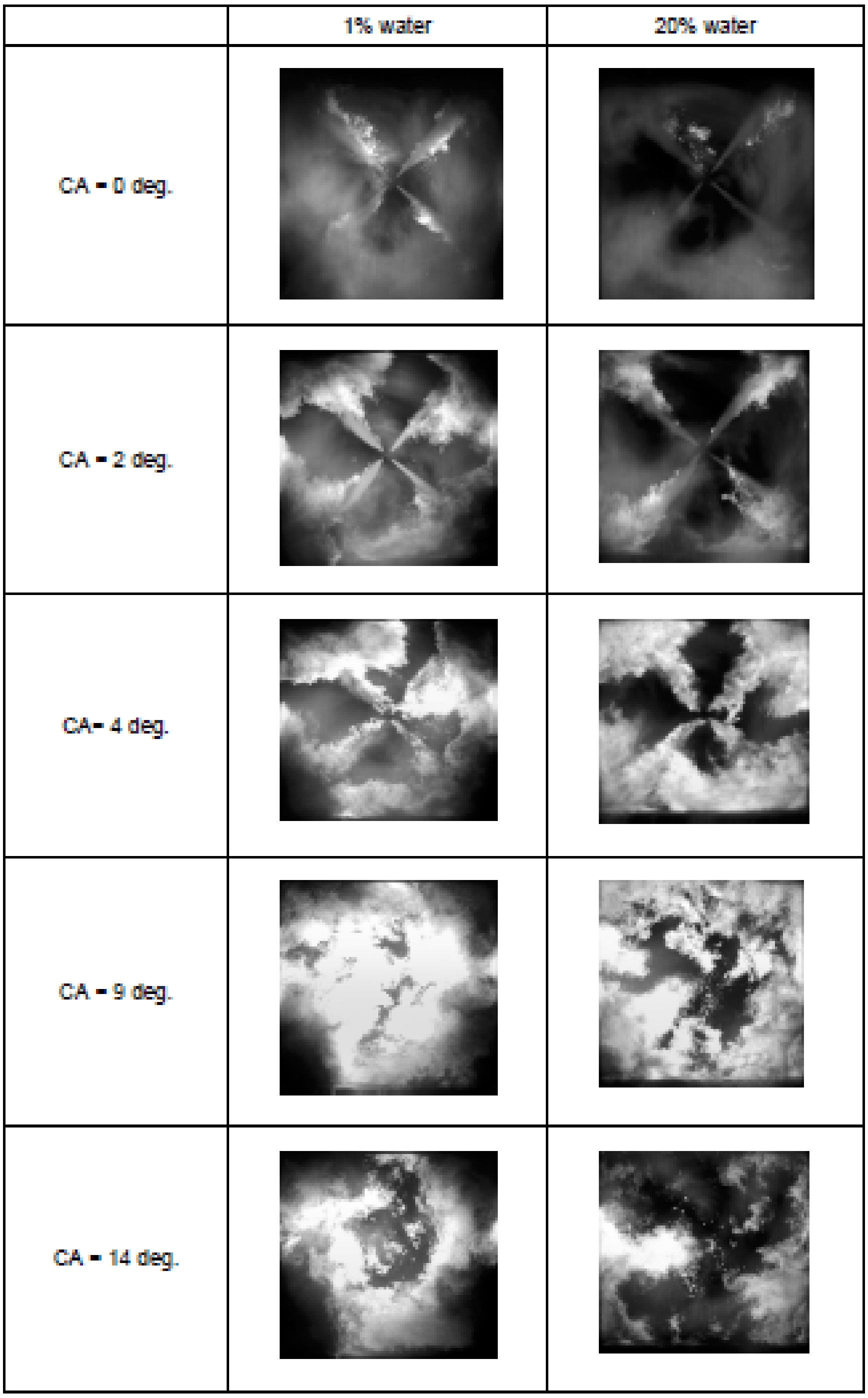

Figure 3: Combustion images from an optically-accessible DI diesel engine (Ki Hoon Song, 2000).

1.4 System

1.4.1 Different systems for introducing water to the combustion chamber

There are three main principles for water injection in diesel engines: WFE, direct water injection and intake air humidification. This report will cover the WFE system, but in this chapter I will give an overview of the tree different systems for water injection.

1.4.1.1 Direct water injection

This system requires a separate injection system as the water is injected by its own injectors. This means that the system requires its own pumps and possibly common-rail system. With the DWI system the water spray will be directed to the areas that need cooling. One advantage of this system is that the fuel and water injecting systems are separated, so the amount of water added can be controlled over a much larger spectre than with an emulsion-system. Figure 4 shows NO_X reduction in a CFD simulation of DWI. Figure 5 shows that NO_X emissions can be reduced by as much as 50% by injecting 70% water compared with fuel into the cylinder.

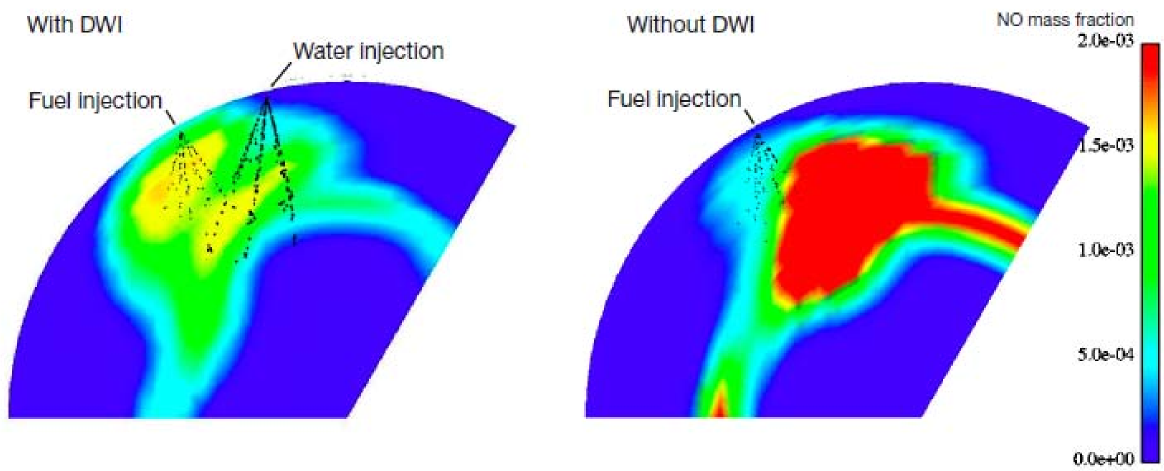

Figure 4: Wärtsilas CFD simulation of NOX reduction using DWI (Heinrich Schmid and German Weisser, 2005).

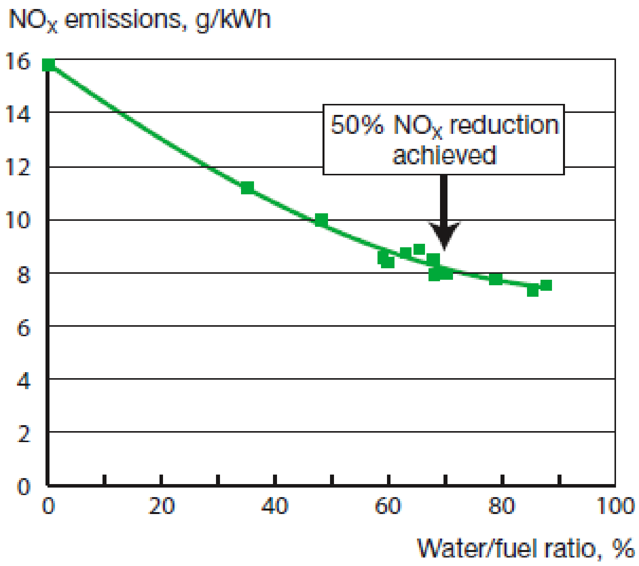

Figure 5: Reduced NOX emissions for Wärtsilä Sulzer RT-flex research engine using DWI (Heinrich Schmid and German Weisser, 2005).

1.4.1.2 Intake air humidification

This system injects water to the intake air and therefore there will be a homogeneous distribution of the water inside the engine cylinder.

Wärtsilä has their wetpack system where pressurized water is added to the intake air after the turbocharger compressor. Test on a pilot vessel has shown reduction of NO_X emissions of 40% (Berisa, 2007).

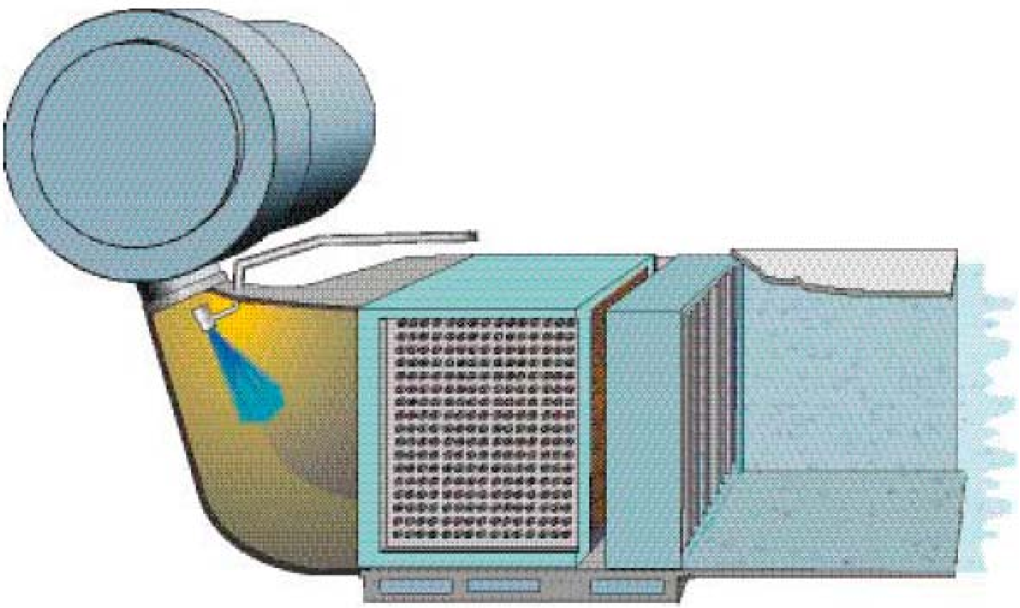

Figure 6: Wärtsilä wetpack system (Berisa, 2007).

Man B&W have tested a SAM system. This system also injects water into the air after the compressor of the turbocharger. This system however uses seawater and the effects upon components after long time use has not yet been investigated. The system first injects seawater for cooling and near 100% humidification of the intake air. The freshwater stages as seen in Figure 7 are not for cooling of the air or further humidification, but for cleaning of the salt. The drainage system for cleaning out salt has shown an efficiency of 99.6% cleaning on a pilot plant.

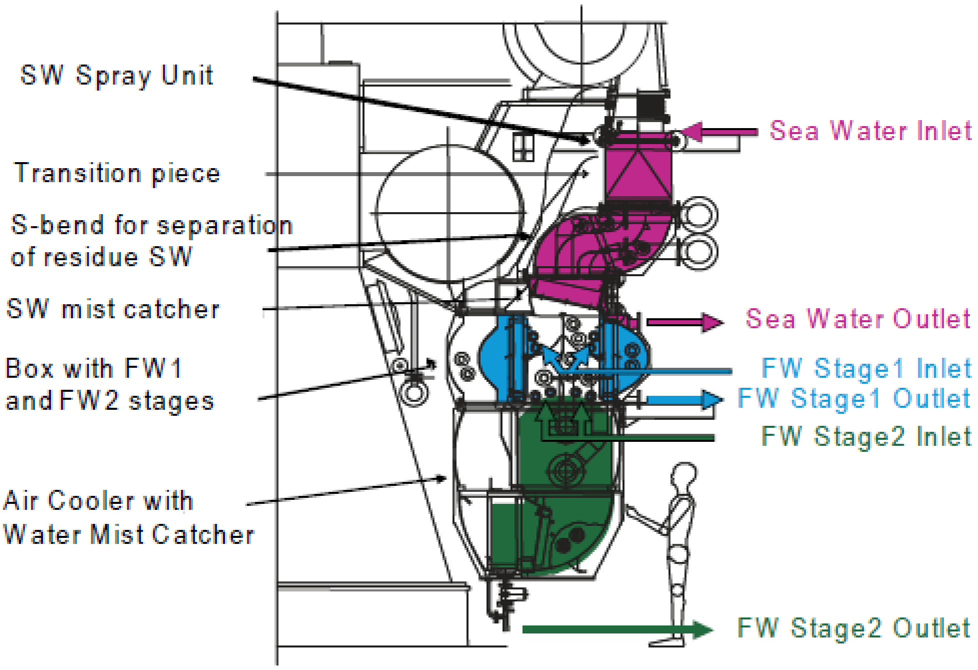

Figure 7: Man B&W's SAM system (Man B&W, 2008).

1.4.1.3 Water-fuel emulsion

This system is based on mixing of water into the fuel. The mixing takes place in a homogenizer and the emulsion is injected into the engine cylinder using the fuel injection system. Therefore additional injecting equipment is not needed. This is of course dependant on the installed injection system and the desired effect. MAN B&W has done test with up to 50% water added on stationary engines (Man B&W, 2008). The need for new components in the case of a retrofit installation is dependent on the installed injection system and the desired NO_X reduction. Normally 10 to 20% NO_X reduction can be achieved without changes to the injection pumps.

1.4.2 System Description

A system for WFE consists of different components. Foremost there is the water supply. This system needs to be able to feed water of a good enough quality with a high enough quantity. There need to be pumps to feed water to the homogeniser. The homogeniser mixes the water into the fuel to create the water-fuel emulsion. Next there is a pre-heater. And there is a measurement instrument for the control system and at last we have the injection system.

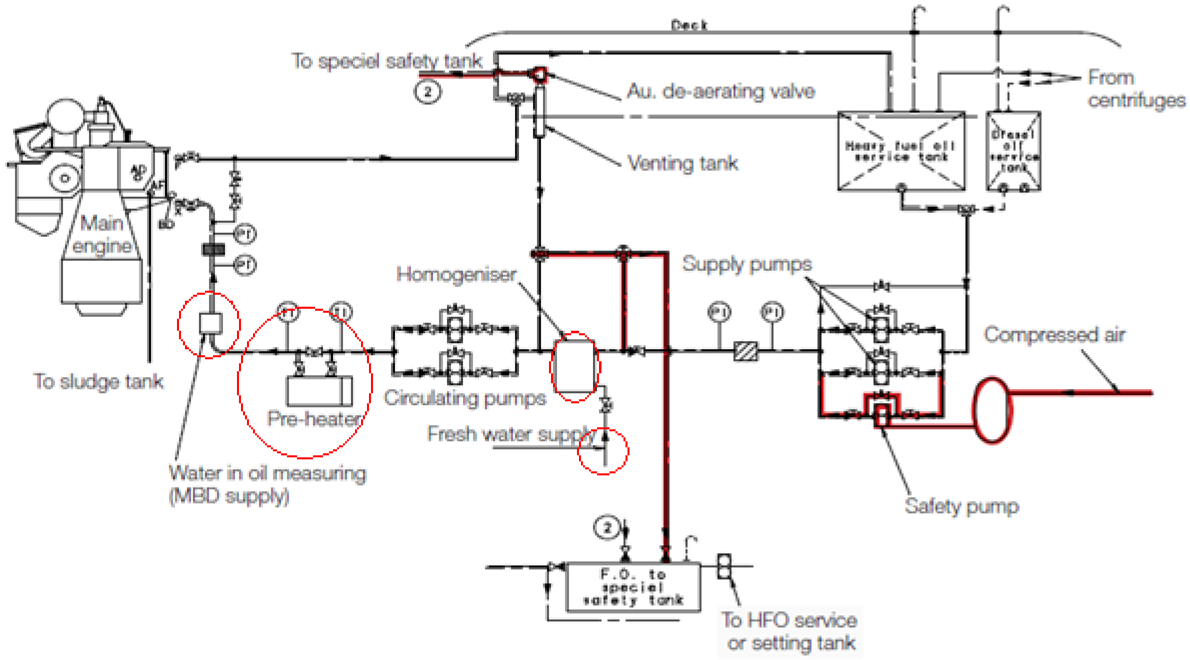

Figure 8: System drawing (Man B&W, 2008).

1.4.2.1 Water supply

There is two ways to ensure water supply to the system. Water can either be supplied from freshwater tanks that are filled while the ship is in port. This means that the vessel has to have a considerably large freshwater tank capacity. The size of these is of course dependent on how long the sailing route of the ship is.

The other alternative is to have desalination plants on board the ship. These plants will take seawater from the sea chests and clean it for salts to make it suitable for injection into the engine. It is of great importance to remove all salts from the water to avoid that sodium from the salt reacts with vanadium from the fuel and create particles/deposits that can accumulate on the valve spindles and seats. These deposits can result in leaks and they have to be avoided.

Desalination plants onboard ships are subject to a number of factors not experienced in land-based plants. There is inertial forces and torques associated with ship movement, vibrations and shock from other equipment, variations of feed water temperature and salinity. So they have to be designed sturdier and more flexible than shore based installations. This said desalination plants are the most realistic alternative for water supply, at least for ships with a long sailing route.

1.4.2.2 Homogeniser
A homogeniser ensures the mixing of water into the fuel from the supply pump. The homogeniser injects water into the fuel stream from nozzles at high pressure to create small droplets of water in the fuel. The homogeniser is as shown in Figure 8 positioned between the fuel supply pump and circulation pumps.

There are two types of homogenisers that can be used: Ultrasonic and mechanical types. The goal is to create as small droplets of water as possible to ensure optimal spray in the combustion chamber. Heavy fuel oil and water emulsify readily, but using distillates (gas oil), there is a need for an emulsifying agent. These agents are based on low-cost vegetable proteins and are abundantly available.

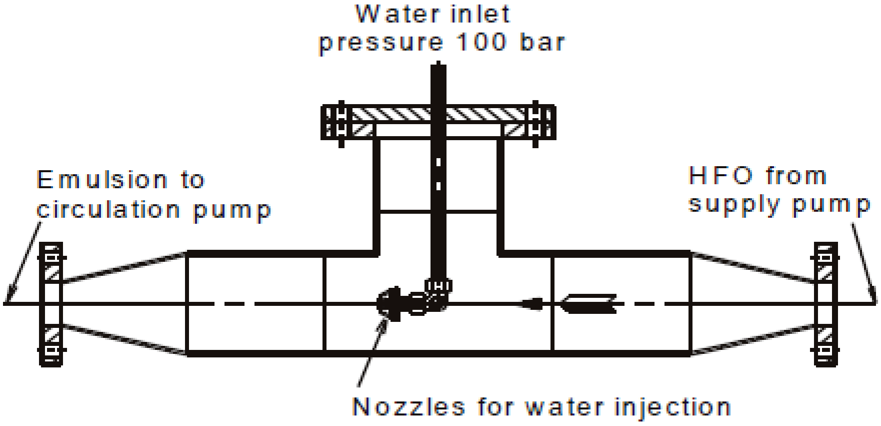

Figure 9: Homogeniser (Man B&W, 2008).

1.4.2.3 Pre-heater
After the circulation pumps there is a pre-heater. This pre-heater is necessary because when adding water droplets to the fuel the viscosity is increased. To maintain a viscosity of 10-15 cst. for injection it is necessary to raise the temperature by pre-heating (Man B&W, 2008). When raising the temperature there is a risk of the water boiling so there is also a need to increase the pressure. This pressure is ensured by the circulation pumps.

1.4.2.4 Injection system
The injection system has to handle more mass flow when injecting water-fuel emulsion. If the installation of emulsion system is a retrofit there might be a need to change the entire injection system. The fuel injection pumps will normally be a restrictor for the mass flow, but the pumps can normally deliver water-fuel emulsion with 10 to 20% water added without changes to the pumps (Heinrich Schmid and German Weisser, 2005). If a higher degree of reduction is desired the engine can either be derated or the system has to be redesigned.

The injection nozzle design also has to be adapted to the increased amount of liquid. This modified nozzle design will penalise fuel consumption and component temperatures while operating without water.

Engines with electronically controlled injection and/or valve timing will have an advantage over camshaft controlled engines when running on water-fuel emulsion. This is because of the increased control possibilities for optimal injection timing etc. This can also be an advantage while running the engine without emulsion, but the problems mentioned before will still exist to some degree.

1.4.2.5 Control system

In the water-fuel emulsion system there must be a control system to ensure the right mix at different loads. In Figure 8 there is a water-in-oil measurement instrument after the pre-heater this instrument can measure the water content by capacitive, dielectric or infrared measurements. The data from the measurement is feed to a control system that can control the water mixing to ensure that the mixing is correct.

1.4.3 System efficiency

The efficiency of water-fuel emulsion can be related to different parameters. It can be related to how much NO_X emissions can be reduced pr percent of water in the fuel. Or how much more/less fuel the engine uses pr. kWh also called specific fuel oil consumption. It can also be related to the effect upon other emissions like soot, hydro carbons or PM, but this is more of a side effect than a main function.

Regarding water amount NO_X reduction generally follows the relationship that adding 10% water to the fuel decreases specific NO_X emissions by 10%. When it comes to SFOC when using water-fuel emulsion it is hard to find exact data, but MAN B&W has set the penalty to about 1% for a retrofit installation that gives 20% reduction in NO_X emissions.

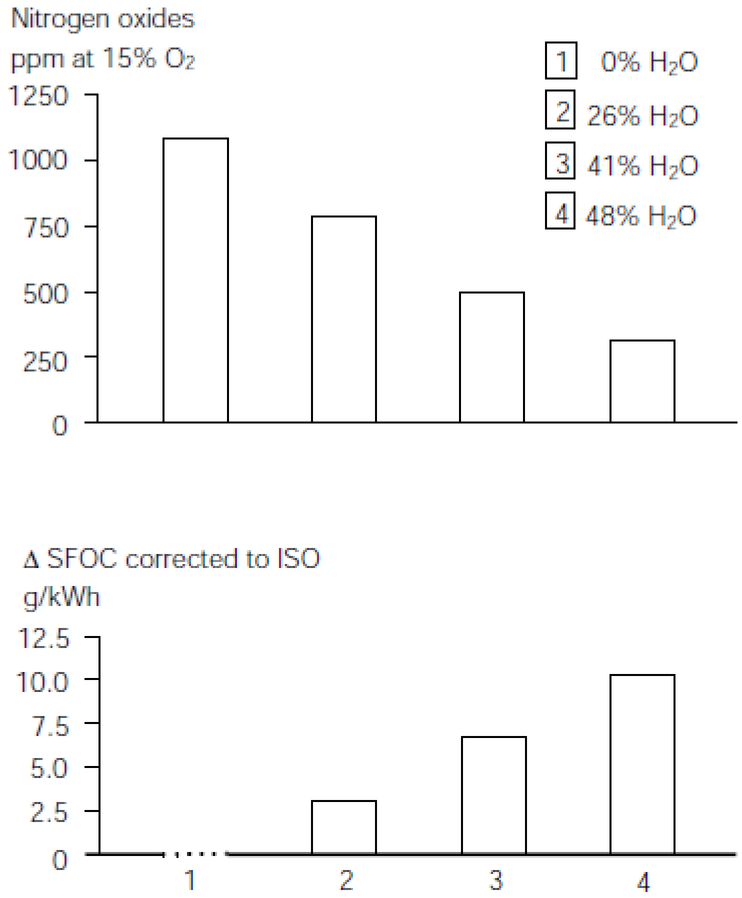

Figure 10: Effects of WFE on NOx emission and SFOC (Man B&W, 2003).

1.4.4 Comparison with other techniques

As Figure 11 suggests the specific fuel oil consumption penalty of WFE is low compared to other techniques with the same reduction. That being said it is clear from the figure that direct water injection will give a reduction of 50% which is large compared to that of water-fuel emulsion with only a slightly larger penalty to fuel consumption. Direct water injection will however require more equipment than WFE.

Compared to other techniques of NO_X WFE in ship diesel engines is generally used for lower range reductions of approximately 20%. This is because adding more water than approximately 20% will generally mean a redesign of the injection system. Figure 11 made by Wärtsilä shows that a reduction of 20% by WFE will give a penalty of about 3 g/kWh in specific fuel consumption.

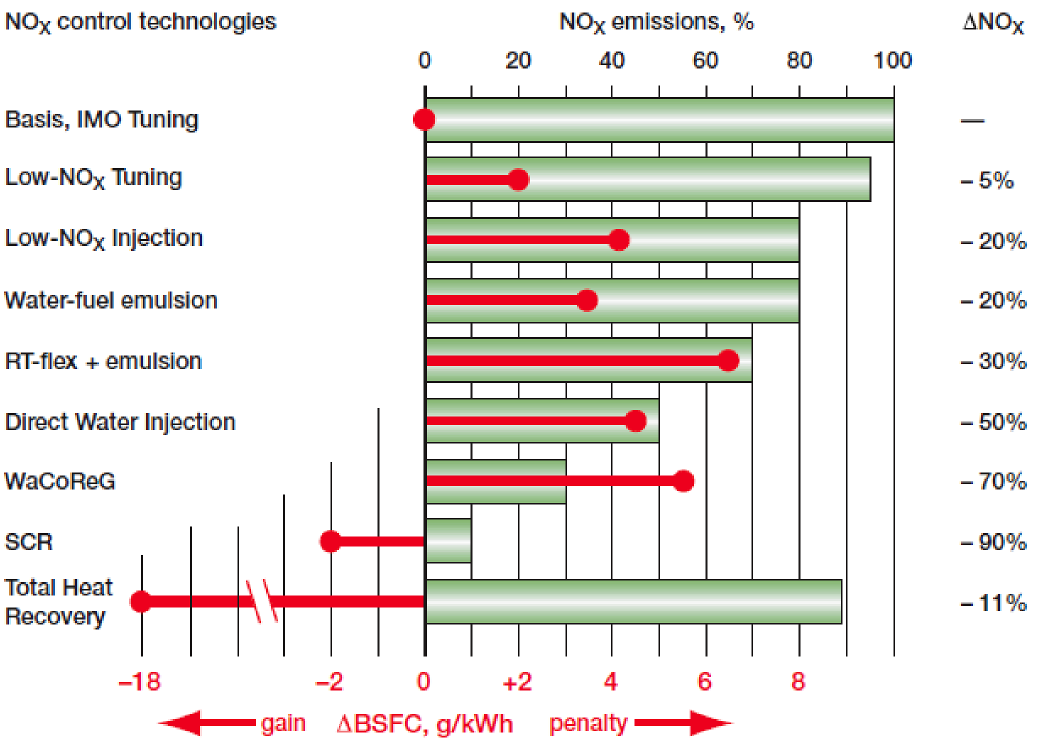

Figure 11: Summary of the changes in NO_X emissions and specific fuel consumption for various emission control technologies (Heinrich Schmid and German Weisser, 2005).

1.4.4.1 Combinations

There is also the possibility of combining WFE with other techniques for NO_X reduction. Figure 12 shows a comparison made by MAN B&W, this comparison shows combinations of water-fuel emulsion, slide valves, EGR (exhaust gas recycling) and reduction of the maximum pressure by injection retardation. Especially interesting is the combination of slide valves, 50% WFE and 20% EGR. This alternative has a large reduction potential about 72% and a small penalty in fuel consumption.

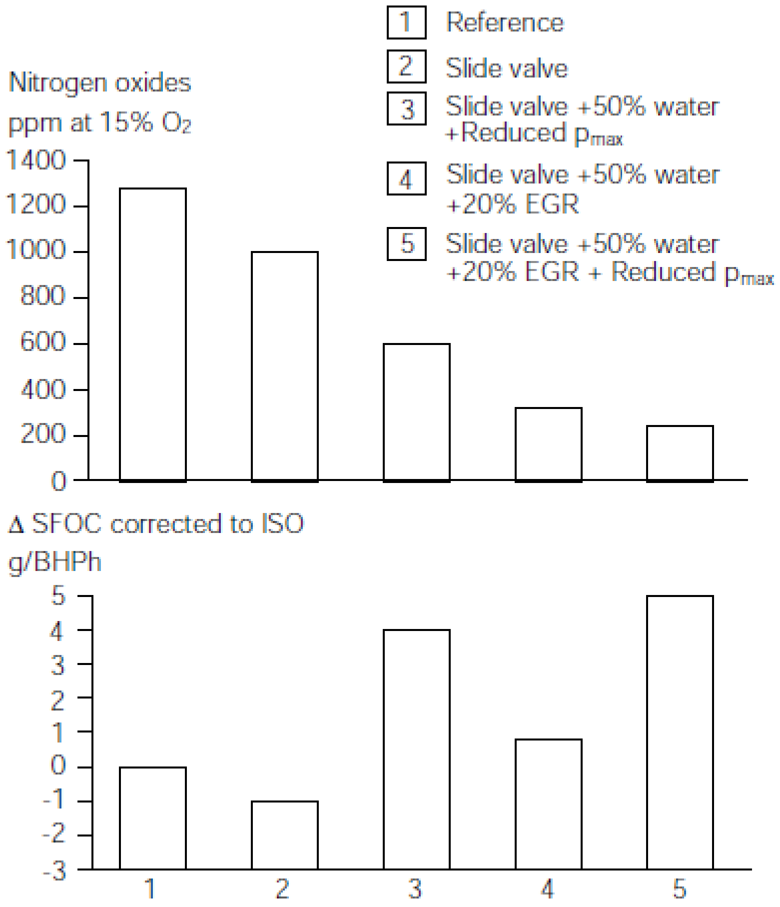

Figure 12: Effect of combining NO_X reduction methods on a 5S70MC engine (Man B&W, 2003).

2 Operational Aspects

There are some specific concerns when running on WFE compared with normal fuel oil. In this chapter I will address some of these concerns.

When water is emulsified into the fuel by a homogeniser it does not create a stable mix. Water and fuel will separate; the time it takes before the mix separates depends upon the temperature and viscosity of the mix. An emulsifying agent can be added if there is a need for a more stable mix. White diesel for instance is a mix of fuel and water in which an emulsifying agent is added to keep the mix stable.

The stability of the mix can be a concern when shutting down the engine. If there is water left in the fuel it will separate and cause corrosion in the fuel system. This corrosion can destroy the injectors. If this happens it is both time consuming and expensive to fix. There is also need for the system to be design from the “fail safe” principle. This means that the emulsion system must be shut down if there is a failure. This needs to be done to avoid wrong dosage and/or water left in the system if there is a failure leading to an engine stop. There should be a system independent of the dosage control system that can shut down the water supply. This should be automated to ensure that the emulsion system gets shut down in time for the engine to keep running. If there is an engine failure it should be able to shutdown at once when the failure is detected.

Operational aspects also include how much NOx-reduction is needed. The fuel injection pumps might not be able to handle all the water mass flow in addition to the fuel that has to be injected. If the water amount at full load are exceeding about 20-25% which is a normal overcapacity of the injectors, the plunger diameter of the injection pumps will have to be increased.

While the water amount at full load is limited by the plunger diameter, the water amount at normal load can be higher. The injection pumps can handle a higher relative water mass at normal load. Since the engine is not running at full load while the ship is sailing en route. The NOx-reduction can be higher en route than it is at full load. If the NOx-reduction at full load is designed to be so high that the plunger diameter has to be changed, this will affect the engine operation. The engine will no longer be able to operate on normal fuel oil because the fuel injected pr. crank angle will be to grate and the pressure rise in the cylinder will be higher than the design pressure. This will wear out the engine and at worst destroy it.

2.1 Motorconsults RedSys system

Motorconsult AS is a Trondheim based company founded by Dr. Ing. Magnus B. Småvik. By email correspondence with Motorconsult I have received some operational and maintenance information about their patented RedSys system.

MOTORCONSULT produces and sells their water in fuel emulsion dosage apparatus. The RedSys system is based on that water and emulsifying agent are pumped into the fuel oil pipe and then exposed to turbulent flow to create an emulsion with water droplets in the fuel. The emulsifying agent used in this system is called Hammerdown. Hammerdown does not only contain an emulsifying agent, but one of the substances also reduce the surface tension of the fuel oil in order to make it break up into smaller droplets. This contributes to complete combustion and reduces the sediments of soot inside the engine cylinder. Hammerdown is mixed with one part to 6500 parts of fuel.

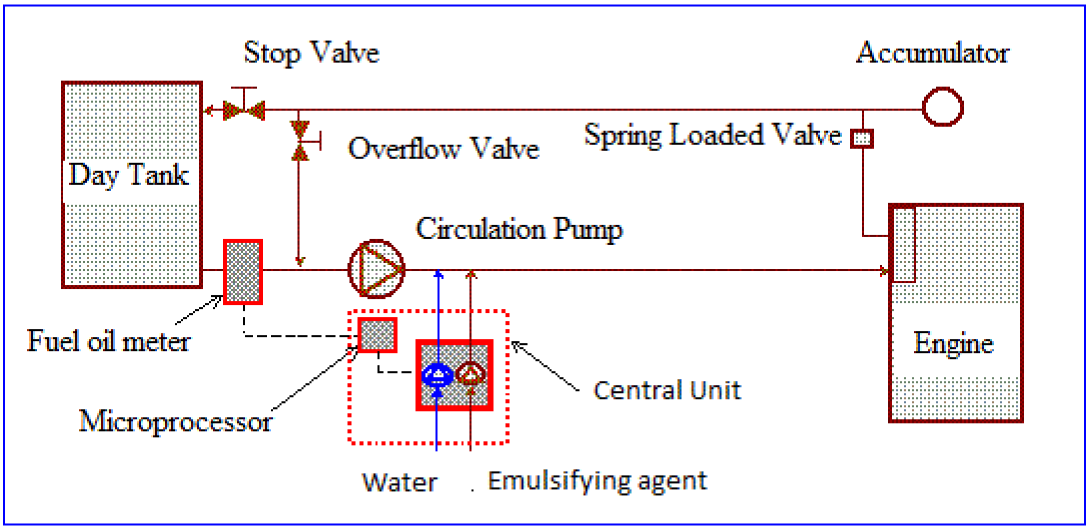

Figure 13: RedSys system drawing (MOTORCONSULT AS).

In Figure 13 a system drawing of the RedSys system for running on distillates is shown. The RedSys system itself consists only of the central unit and the fuel oil meter. The fuel oil meter sends a pulse signal that is converted to a current ranging from 4 to 20 mA. The microprocessor receives this signal and is programmed to give a certain water amount on the basis of the engine power. According to (MOTORCONSULT AS, 2010) the relationship between fuel oil and engine power is not of a more critical nature than that it can be gathered by analyzing the engine test data from the ships sea trial. The water pump delivers the amount of water regulated by the microprocessor and the emulsifying agent pump delivers the right amount of emulsifying agent controlled directly by a signal amplifier.

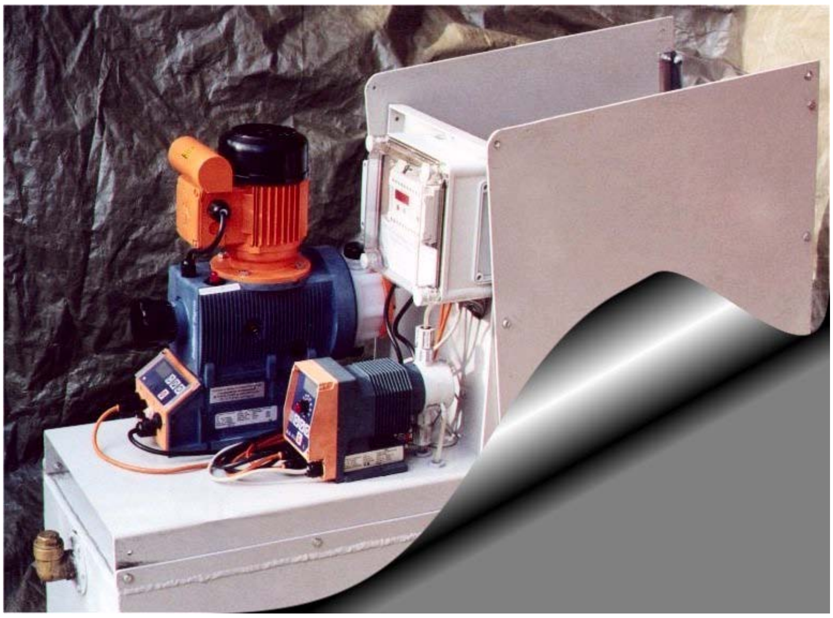
Figure 14: Central unit in the RedSys system (MOTORCONSULT AS).

Figure 14 is a picture of the central unit in the RedSys system. This unit consists of the microprocessor and the pumps for water and emulsifying agent.

For the RedSys system I got some specific information concerning operational and maintenance aspects from (MOTORCONSULT AS, 2010).

When running at part load below a certain limit, the charge air pressure will drop more when running on WFE than when running on normal fuel. To avoid the engine from stalling and ensure proper combustion there has to be a protective system separate from the dosage system to ensure that the water pump shuts down and the engine continues to run on normal fuel.

When it comes to reliability the WFE systems seems to have few problems. Dr. Ing. Småvik from MOTORCONSULT informed me that his company used a couple of years to determine which components to use for their system. They tested different back-pressure valves for both water and Hammerdown. After this period of testing they found components that have good reliability and they have not observed any operation problem with the back-pressure valves.

When it comes to maintenance the lube oil for the dosage pumps have to be changed every year. MOTORCONSULT informed me that one customer had waited six years for the lube oil change. When the central unit finally was sent to MOTORCONSULT for servicing a cleaning of the pump and changing of the lube oil was performed. After a function test the unit was operating normally and could be returned.

Wear and tear in certain parts of the fuel oil meter and the membrane in the dosage pump make it necessary to change them after 5-10 years according to the manufacturers of the parts, but the fuel oil meter and pump itself last much longer. Two microprocessors have been changed within the warranty time, while the rest of them have lasted several years. The signal converter from pulse to current has had satisfactory performance.

Condition monitoring of the system is of such a simple nature that it can be performed by the customers after instructions by MOTORCONSULT.

3 Simulations with GT-Power

GT-Power is an engine simulation program where a model of the engine containing all the different components are defined and linked by a graphic user interface. The modelling includes the crankshaft, cylinders, injectors, turbocharger, charge air cooler and all of the piping. The different components must be defined by their geometry and properties. The program can be used for a variety of purposes and different methods can be utilized depending on what the user wants to investigate.

3.1 Model

The model I have used in the simulations is based on a Rolls Royce KRG-6 engine. The model was made by Rolls Royce and given to Nabil Al Ryati a PH.D student at NTNU department of marine technology. It is a complete model of the engine with turbocharger, charge air cooler and a control system to regulate the injected fuel in order to reach a target brake power. In Table 2 there is some data from the original Rolls Royce KRG6 which the model is based on.

Rolls Royce KRG6 data	Value	Unit
Number of cylinders	6	[-]
Cylinder bore	250	[mm]
Piston stroke	300	[mm]
Rated Power MCR	1165	[kW]
Power 75% load	874	[kW]
Power 50% load	583	[kW]
Power 25% load	291	[kW]
Rated Active Power Gen	1110	[kW]
MEP	22	[bar]
Rated Speed	720	[rpm]
Mean piston speed	7.2	[m/s]
Displacement	88	[l]
SFC	193	[g/kWh]
Epsilon	1:12.5	[-]

Table 2: Rolls Royce KRG6 Engine Data.

Mr. Al Ryati received the model without a defined combustion model, so he had to define a Wiebe function to describe the combustion. The combustion model was made without knowing the pressure trace for an entire cycle so the combustion model was calibrated from performance data and is not an accurate reproduction of the real heat release in the engine cylinder. The performance data used to calibrate the combustion was the data in Table 2 plus Pme, receiver pressure, Pmax and temperature before and after the turbocharger for 50, 75 and 100% load.

The engine that is modelled has cam controlled injectors and this means that the pressure profile of the injectors will be constant with constant engine speed. The engine is running at generator loads and this means that the engine speed is constant for all the loads. This has been a problem when running simulations and will be described in the simulations chapter.

3.2 Adaptations

There were some adaptations that needed to be done in the GT-Power model in order to calculate the NOx emissions. If the correct NOx model is not used the NOx values you get from simulations will only be the equilibrium NOx in the cylinder and not the correct NOx emissions. When I started working with GT-Power I was not aware of this and there were a lot of frustration linked to unrealistic NOx emission values from simulations. Thanks to GT-Power technical support I got help with adding the right modifications to the model in order to achieve realistic NOx emission values.

3.2.1 Combustion model

In GT-Power you can choose between many different combustion models depending on what kind of information you have about the engine. The combustion model chosen by Mr. Al Ryati for this model is “ENGCylCombDIWiebe”. This model is made for direct injection engines and is a Wibe function that consists of three different functions with different parameters controlling the different phases of combustion. The tree functions are respectively to describe the premixed, main and tail phase of combustion. The Wibe function can be changed by changing the duration in crank angle, the relative fraction and exponent for the tree different phases. It is also possible to change the ignition delay.
These Wibe parameters have been adjusted by Mr. Al Ryati to make the model as realistic as possible between 50% to 100% load. He did not have the correct pressure trace measurements so the model has been tuned to comply with the performance data mentioned in chapter 3.1. This means that the combustion model is not 100% right compared with the real engine but an approximation.

Combustion model parameters	Value	Unit
Ignition Delay	5.1	[deg]
Premixed Fraction	0.03	[-]
Tail Fraction	0.45	[-]
Premixed Duration	6	[deg]
Main Duration	20	[deg]
Tail Duration	35	[deg]
Premixed Exponent	1.072	[-]
Main Exponent	1.42	[-]
Tail Exponent	0.63	[-]
Number of Temperature Zones	two-temp	[-]
Entrained Fuel-Air Option	stoich-lean-obsolete	[-]
Fraction of Fuel Burned	def	[-]
Ignition Delay Multiplier	def	[-]
Premix Duration Multiplier	def	[-]
Main Duration Multiplier	def	[-]
Tail Duration Multiplier	def	[-]
NOx Reference Object	NOX	[-]

Table 3: Combustion model parameters.

The combustion characteristics will change when running on WFE, typically longer ignition delay and cleaner burning in the late phase of combustion. Since I did not have any engine run data of the KRG6 engine running on WFE available, the parameters for the Wiebe curve has not been changed in my simulations. In Figure 15 the Wiebe function for the model with normalized burn rate on the vertical axis and crank angle on the horizontal axis can be observed. As seen in the figure the Wiebe function is a product of 3 different curves; the premixed, main and tail function. The apparent burn rate is represented by the black curve.

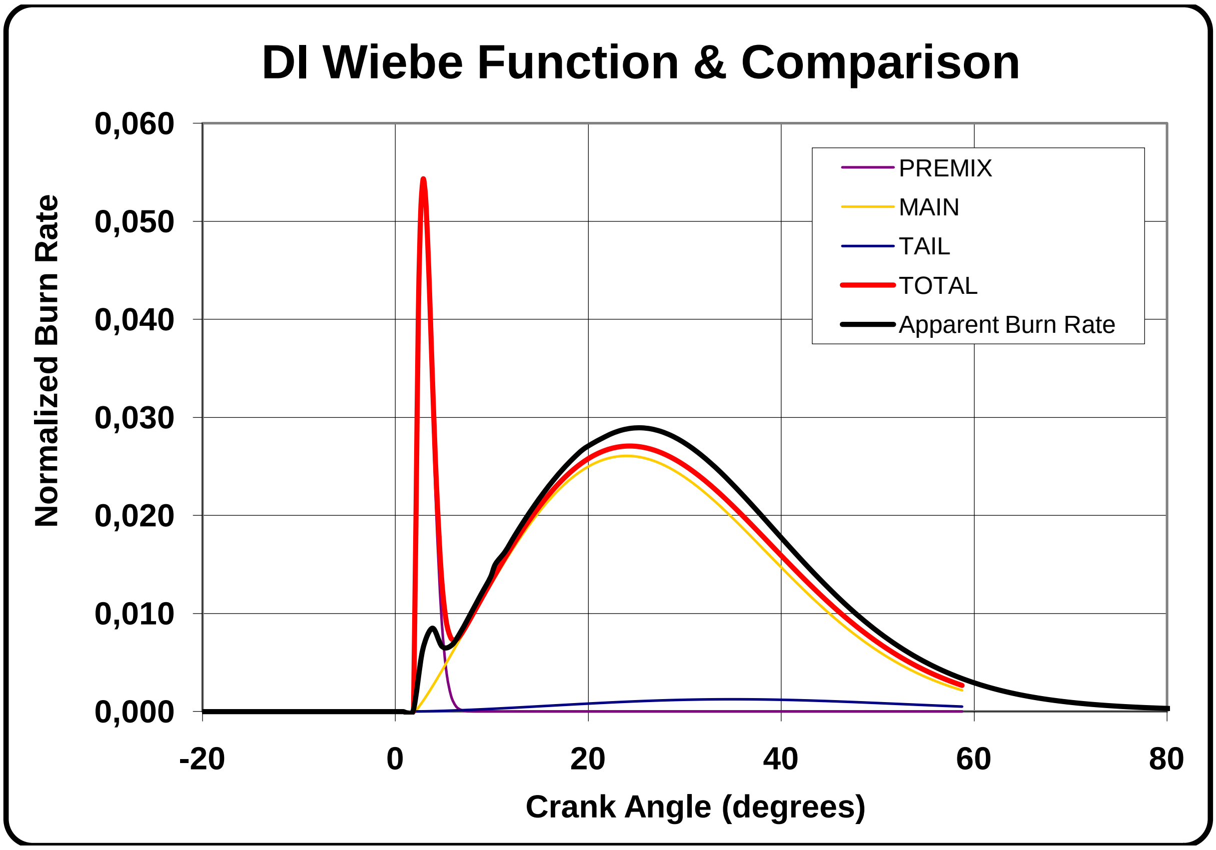

Figure 15: Wiebe function for the combustion model in GT-Power.

3.2.1.1 Two Zone Combustion

In order to get reasonable values for NO formation when running simulations, the cylinder volume has to be divided into two zones. This is done to get an approximation for the local peak cylinder temperatures. In order to do this the option number of temperature zones in the combustion model settings has to be set to two-temp.

When two zone combustion is chosen in GT-Power the combustion occurs in the following manner(Gamma Technologies, 2009):

1. At the start of injection the cylinder is divided into two zones: burned and unburned zone. All of the content in the cylinder is in the unburned zone also including the residual gases from the previous cycle.
2. At each time step, a mixture of fuel and air is transferred from unburned to burned zone. The amount transferred is either prescribed or calculated by the combustion model.

3. When the unburned fuel and associated air has been transferred to the burned zone in the given time step, a chemical equilibrium calculation for the entire “lumped” burned zone is carried out. This calculation takes into account all of the atoms of carbon, hydrogen, oxygen and nitrogen present in the burned zone at that time. The products from the calculation is the equilibrium concentration of N2, O2, H2O, CO2, CO, H2, N, O, H, NO and OH. The equilibrium concentration of these species depends strongly upon the current burned zone temperature and to a lesser degree the pressure.
4. After the composition of the burned zone species have been calculated the internal energy for each species is calculated. Summing up the different species give the sum of burned zone energy. Applying the principal that energy is conserved the new burned and unburned zone temperatures as well as the cylinder pressure is obtained.

The energy equations for respectively unburned and burned zone are solved separately for each time step in each zone. Equation 6 shows the energy equation solved for the unburned zone. Notation u in Equation 6 stands for unburned zone, f for fuel, a for air and the notation f,i for injected fuel. The first term on the right hand side of Equation 6 handle pressure work, the second heat transfer, the third combustion and the final term handle addition of enthalpy from injected fuel. The combustion term consists of the instantaneous rate of fuel consumption or burn rate.

$$\frac{d(m_u e_u)}{dt} = -p\frac{dV_u}{dt} - Q_u - \left(\frac{dm_f}{dt}h_f + \frac{dm_a}{dt}h_a\right) + \frac{dm_{f,i}}{dt}h_{f,i}$$

Equation 6: Unburned zone energy calculation (Gamma Technologies, 2009).

The energy equation for the burned zone is shown in Equation 7 the notation b stands for burned zone, f for fuel and a for air. As for the unburned zone there are also terms for pressure work, heat transfer and combustion in the burned zone energy equation.

$$\frac{d(m_b e_b)}{dt} = -p\frac{dV_b}{dt} - Q_b - \left(\frac{dm_f}{dt}h_f + \frac{dm_a}{dt}h_a\right)$$

Equation 7: Burned zone energy calculation (Gamma Technologies, 2009).

3.2.1.2 NOx model

To calculate the NOx emissions in GT-Power on have to enable the NOx model: "EngCylNOx" in the settings for the combustion model under NOx reference object.

The NOx model in GT-Power is based on the extended Zeldovich mechanism with relations shown in Equation 8. These relations are chemical equilibriums that describe the reactions the molecules in the cylinder undergo when forming NO that is the main part of the NOx emissions from diesel engines. Some of the NO will react with HO2 to create NO2 and OH, but the contribution to NOx emissions from this reaction is only 5-10% of the NOx emissions from a diesel engine.

$$O + N_2 \rightarrow NO + N (1)$$
$$N + O_2 \rightarrow NO + O (2)$$
$$N + OH \rightarrow NO + H (3)$$

Equation 8: Zeldovich mechanism relations (Heywood, 1988).

Each of the relations in Equation 8 has a rate constant. The rate constants are shown in Equation 9. From the rate constants it is easy to see that the temperature is very important. The higher the temperature is the faster the splitting of the nitrogen gas molecule occurs and NO can form.

$$k1 = F_1 * 7.60 * 10^{10} * e^{\frac{-38000 * A_1}{Tb}}$$
$$k2 = F_2 * 6.40 * 10^{6} * T_b * e^{\frac{-3150 * A_2}{T_b}}$$
$$k3 = F_3 * 4.10 * 10^{10}$$

Equation 9: Rate constants (Gamma Technologies, 2009).

In Equation 9 F1, F2, F3, A1 and A2 are changeable multipliers while Tb is the burned zone temperature.

3.2.2 Adding water to the fuel

To implement water in fuel emulsion I have assumed that the fuel consist of both water and fuel as in white diesel. To achieve this it is necessary to use the combine function and make a fluid mixture. In this function I have selected the same diesel as the standard used in the model named "diesel2-combust" and added "h2o-combust" as the second fraction of the fluid mixture. The function then creates two changeable mass fractions parameters, that are changed in the case setup menu in order to make simulations with different fractions of water. Table 4 contains data on the two fractions of the fuel.

Property	Diesel	Water	Unit
Heat of Vaporization at 298 K	250	2442	[kJ/kg]
Density	830.0	1002.5	[kg/m^3]
Enthalpy			
(T-Tref) Coefficient, a1	2050	4192	[-]
(T-Tref)^2 Coefficient, a2	0.00	0.19	[-]

Vapour properties	Diesel	Water	Unit
Carbon Atoms per Molecule	13.5	0	[-]
Hydrogen Atoms per Molecule	23.6	2	[-]
Oxygen Atoms per Molecule	0	1	[-]
Lower Heating Value	43.25	0	[MJ/kg]
Critical Temperature	569	647	[K]
Critical Pressure	24.6	220.9	[bar]
Enthalpy			
(T-Tref) Coefficient, a1	1634	1865	[-]
(T-Tref)^2 Coefficient, a2	1.82	0.19	[-]
(T-Tref)^3 Coefficient, a3	0	1.80E-04	[-]
(T-Tref)^4 Coefficient, a4	0	-8.80E-08	[-]
(T-Tref)^5 Coefficient, a5	0	1.23E-11	[-]

Table 4: Fuel properties.

3.2.3 Evaporation model

For evaporation of the fuel mixture I have used “EngCylEvap” an evaporation model in GT-Power. This model calculates the evaporation rate based on Equation 10.

$$EvaporationRate = \frac{4.16 * RPM_{ref}}{CA50} \left(\frac{T}{T_{ref}}\right)^{TMPEXP} \left(\frac{RPM}{RPM_{ref}}\right)^{RPMEXP}$$

Equation 10: Evaporation Model (Gamma Technologies, 2009).

In Equation 10 CA50 is a time constant for evaporation. It represents the crank angle duration required to evaporate 50% of the fuel at the reference temperature and engine speed which is: 600K and 4000 RPM. TMPEXP and RPMEXP are the exponents for temperature and RPM dependence. Tref and RPMref are the reference temperature and speed (= 600K and 4000 RPM). T and RPM are the actual temperature and engine speed.

During my simulations the CA50 value were set to 10 and the TMPEXP and RPMEXP values were set to 1.

3.3 Simulations

To check the effects of implementing water in the fuel I have been running simulations. Since the combustion model has been tested in the area 50% - 100% load, I chose to run simulations at 50%, 75% and 100% load. The simulations have been run at different water content from 0% water added up to 40% water added. There is a problem related to changing the injected mass with such a magnitude. The problem is that the injector is set up to inject at a given pressure profile. And if the pressure is not sufficient for injecting the needed mass in the time available there will be an inconsistence. This inconsistence is the relationship between the injector pressure profile, injected mass and the geometry of the injectors. So in order to be able to inject the needed mass of fuel and water to reach the target brake power either the pressure needs to be higher, the number of holes needs to be increased or the diameter of the holes needs to be larger.

In Table 5 the settings for the injectors are given. The injected mass is calculated by a control system in the model. The control system has sensors which combined give the power. When the target power is reached the injected mass is correct. When using a set pressure profile in the simulations the nozzle discharge coefficient is calculated. So the input consists of the nozzle geometry that I have chosen to be constant, the target brake power given by the current load and the pressure profile.

Injector properties	Value	Unit
Injected Mass	Calculated	mg
Start of Injection	-12.5	deg
Profile Type	presprof	[-]
Injected Fluid Temperature	343	K
Fluid Object	WaterInFuelEmulssion	[-]
Nozzle Hole Diameter	0.33	mm
Number of Holes per Nozzle	10	
Nozzle Discharge Coefficient	def	

Table 5: Injector properties.

According to (Gamma Technologies, 2009), the RLT variable “cnozz” which is the nozzle discharge coefficient should be monitored to verify that a realistic value has been calculated. This value should be within 0.60-0.75. If the value of “cnozz” is outside of this area there is a inconsistency between the nozzle geometry, the injected mass and the pressure profile.

In order to have a nozzle discharge coefficient within the recommended limits I have changed the injection profile multiplier in the case setup menu. This multiplier will change the original pressure profile in Table 6 by the factor chosen. If the “cnozz” value is too high there is a need for more pressure to inject the needed mass and vice versa.

Injector Pressure Profile					
Crank angle	0	2	10	18	[deg]
Pressure	225	1300	1300	1300	[bar]

Table 6: Injector pressure profile.

Since the engine model is of a cam controlled engine it is not really realistic to change the pressure profile as the injector pump will give a constant pressure with constant engine speed. But since the goal of this part of my master thesis is to investigate the possibility of using GT-Power to measure NOx emissions and fuel consumption while running on WFE I have chosen to disregard this fact and assume that the injector pressure can be regulated without.

The simulations have been done with 50%, 75% and 100% load. For each of these loads I first ran a simulation with diesel then 10%, 20%, 30% and 40% water added.

3.3.1 Input data

When running the simulations the pressure profile multiplier was changed. The value of the multiplier ranged all the way from 0.7 for the lowest value to 3.7 for the highest value. The change in pressure profile multiplier was done to keep the nozzle discharge coefficient within reasonable values. Such large changes in the pressure profile would be unlikely in a real engine. I also don't think that GT-Powers injector simulation in this case can really cover what happens in a real injector.

However adding large amounts of water at full load is not reasonable in a real engine so the trend in which the pressure profile multiplier is changing is realistic.

3.3.1.1 100% load

The input data from the simulations at full load can be seen in Table 7. The profile multiplier is 2 already at 0% water added and is almost doubled at 40% water added. The required nozzle discharge coefficient has been found by changing the pressure multiplier in steps until the "cnozz" value was within the 0.65-0.75 limit.

Input 100% load	0%	10%	20%	30%	40%	Unit
Water Added	0	10	20	30	40	[%]
Xwater	0.000	0.091	0.167	0.231	0.286	[-]
Xfuel	1.000	0.909	0.833	0.769	0.714	[-]
Brake Power	1165	1165	1165	1165	1165	[kW]
Engine Speed	720	720	720	720	720	[RPM]
Injected mass	calculated	calculated	calculated	calculated	calculated	[mg]
Profile multiplier	2	2.4	2.8	3.3	3.7	[-]
injection start	-12.5	-12.5	-12.5	-12.5	-12.5	[deg]
Ambient temp	283.15	283.15	283.15	283.15	283.15	[K]
Ambient pressure	1	1	1	1	1	[bar]
Inlet temp	333	333	333	333	333	[K]

Table 7: Input data 100% load.

3.3.1.2 75% load

Input data for 75% load is represented in Table 8. Here the pressure profile varies from 1.2 at 0% water added to 2.2 at 40% water added. The procedure for finding the pressure profile values is the same as with 100% load.

Input 75% load	0%	10%	20%	30%	40%	Unit
Water Added	0	10	20	30	40	[%]
Xwater	0.000	0.091	0.167	0.231	0.286	[-]
Xfuel	1.000	0.909	0.833	0.769	0.714	[-]
Brake Power target	874	874	874	874	874	[kW]
Engine Speed	720	720	720	720	720	[RPM]
Injected mass	calculated	calculated	calculated	calculated	calculated	[mg]
Profile multiplier	1.2	1.4	1.7	1.9	2.2	[-]
injection start	-12.5	-12.5	-12.5	-12.5	-12.5	[deg]
Ambient temperature	283.15	283.15	283.15	283.15	283.15	[K]
Ambient pressure	1	1	1	1	1	[bar]
Inlet temprature	333	333	333	333	333	[K]

Table 8: Input data 75% load.

3.3.1.3 50% load

Input data for 50% load is represented in Table 9. Here the pressure profile varies from 0.7 at 0% water added to 1.3 at 40% water added. The procedure for finding the pressure profile values is the same as with 100% load.

Input 50% load	0%	10%	20%	30%	40%	Unit
Water Added	0	10	20	30	40	[%]
Xwater	0.000	0.091	0.167	0.231	0.286	[-]
Xfuel	1.000	0.909	0.833	0.769	0.714	[-]
Brake Power target	583	583	583	583	583	[kW]
Engine Speed	720	720	720	720	720	[RPM]
Injected mass	calculated	calculated	calculated	calculated	calculated	[mg]
Profile multiplier	0.7	0.8	1	1.1	1.3	[-]
injection start	-12.5	-12.5	-12.5	-12.5	-12.5	[deg]
Ambient temperature	283.15	283.15	283.15	283.15	283.15	[K]
Ambient pressure	1	1	1	1	1	[bar]
Inlet temprature	333	333	333	333	333	[K]

Table 9: Input data 50% load.

3.3.2 Results

In this chapter the most relevant results for the simulations at 100%, 75% and 50% load is covered. If one wants to have a look at more of the results I have attached the complete tables for engine performance predictions and key cylinder predictions for all the loads and water contents in AppendixA: to AppendixD: at the end of the report. In the appendix tables the different water contents are denoted case 1 to case 5, where case 1 is no water added and case5 is 40% water added.

3.3.2.1 100% Load

The most important results for the simulations at full load are covered in Table 10. The results I have looked into are the NOx in ppm and specific emission, brake specific fuel consumption and temperature.

100% Load - Power: 1165 [kW]						
Water added	0	10	20	30	40	[%]
Nox	2425	2204	1992	1794	1605	[ppm]
Indicated Specific NOx	23.4	21.4	19.5	17.7	16.0	[g/kW-h]
Reduction in NOx	0.0	2.0	3.9	5.6	7.4	[g/kW-h]
Relative red in NOx	0.0	8.4	16.5	24.2	31.6	[%]
Maximum Cylinder temp	1725	1714	1703	1693	1682	[K]
BSAC	8032	8057	8082	8107	8130	[g/kW-h]
BSFC	190.3	209.8	229.3	248.7	268.4	[g/kW-h]
BSFC corrected for water	190.3	190.7	191.1	191.3	191.7	[g/kW-h]
Change in BSFC	0.00	0.43	0.78	1.01	1.41	[g/kW-h]
Relative change in BSFC	0.00	0.22	0.41	0.53	0.74	[%]

Table 10: Simulation results 100% load.

The results give very high NOx emission, with specific emission values over 20 g/kWh. I don’t know how large the NOx emissions from a real KRG-6 engine are, but I suspect that they will be lower than what my simulation results were. The reason for these high values might be that none of the changeable parameters in the NOx model has been changed in any way and the combustion model is not completely correct. So it might be wrong input data that makes the NOx emissions higher than they are supposed to be. With measurements of the cylinder pressure curve and NOx emissions from a real KRG-6 engine, it would be possible to change the parameters in combustion and NOx model to comply with the real engine characteristics.

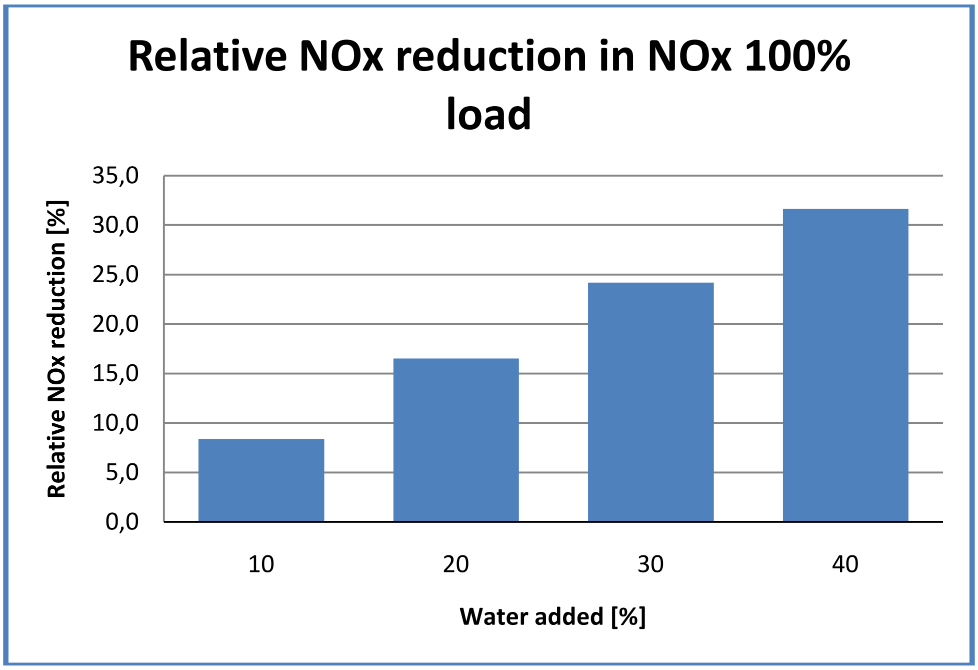

Figure 16: Relative reduction in NOx emission 100% load.

Figure 16 shows the relative reduction in specific NOx emission. The reduction is about 8% pr. 10% water added and is quite close till the results given by the engine manufacturers. They say that 10% water added will give about 10% reduction in NOx emission. It is also important to keep in mind that the engine manufactures are company representatives and typically take the results for the best possible conditions when presenting their findings.

The change in brake specific fuel consumption in Figure 17 shows a slight increase in fuel oil consumption with increasing water content. The increase is lower than what the engine manufacturers claims is the penalty, but still the trend is the same with higher brake specific fuel consumption with higher water content.

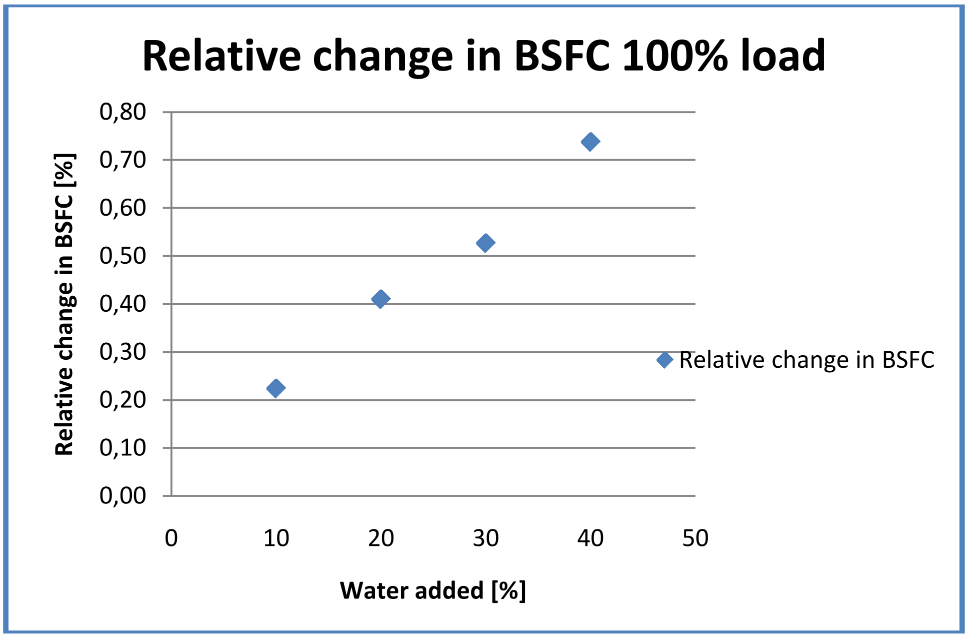

Figure 17: Relative change in BSFC 100% load.

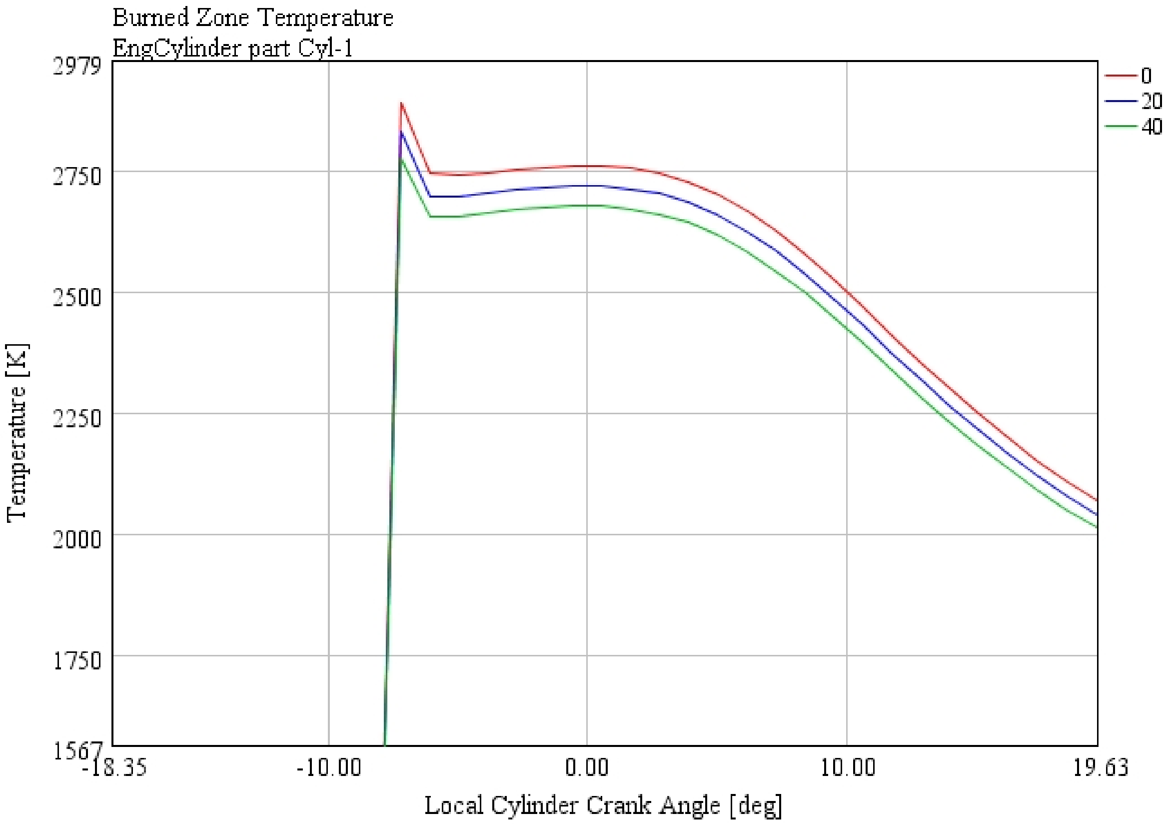

Figure 18: Burned Zone Temperature 100% load.

Looking at Figure 18 one can see that the burned zone temperature is lowered with increasing water content. This is in accordance with the before discussed reduction of NOx as the formation of NO is speeded up in an exponential manner with temperatures above 2000°C (Stapersma, 2009).

3.3.2.2 75% Load

The most important results for the simulations at 75% load are covered in Table 11. As with full load the results I have looked into are the NOx in ppm and specific emission, brake specific fuel consumption and temperature.

75% Load - Power: 874 [kW]						
Water added	0	10	20	30	40	[%]
Nox	1898	1706	1546	1377	1217	[ppm]
Indicated Specific NOx	19.2	17.4	15.8	14.2	12.6	[g/kW-h]
Reduction in NOx	0.0	1.8	3.4	5.0	6.5	[g/kW-h]
Relative red in NOx	0.0	9.4	17.6	26.0	34.1	[%]
Maximum Cylinder temp	1675	1664	1658	1648	1638	[K]
BSAC	8492	8521	8508	8528	8558	[g/kW-h]
BSFC	193.5	213.4	233.4	253.5	273.6	[g/kW-h]
BSFC corrected for water	193.5	194.0	194.5	195.0	195.4	[g/kW-h]
Change in BSFC	0.00	0.50	1.00	1.50	1.93	[g/kW-h]
Relative change in BSFC	0.00	0.26	0.52	0.78	1.00	[%]

Table 11: Simulation results 75% load.

As with full load the emission of NOx seams to high, I give the same grounds for the results here as with the full load results.

The relative reductions in specific NOx emission from Figure 19 are larger than for the full load case. I will not speculate on why this happens. But as discussed earlier in this report, the combustion parameters are kept the same for all loads and water contents. This might be some of the reason for these results.

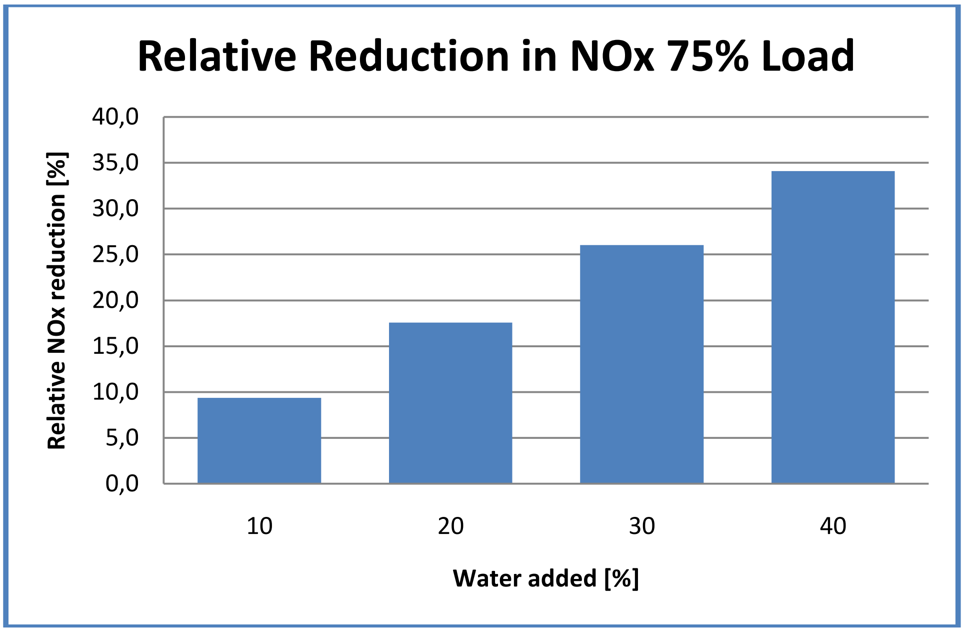

Figure 19: Relative reduction in NOx emission 75% load.

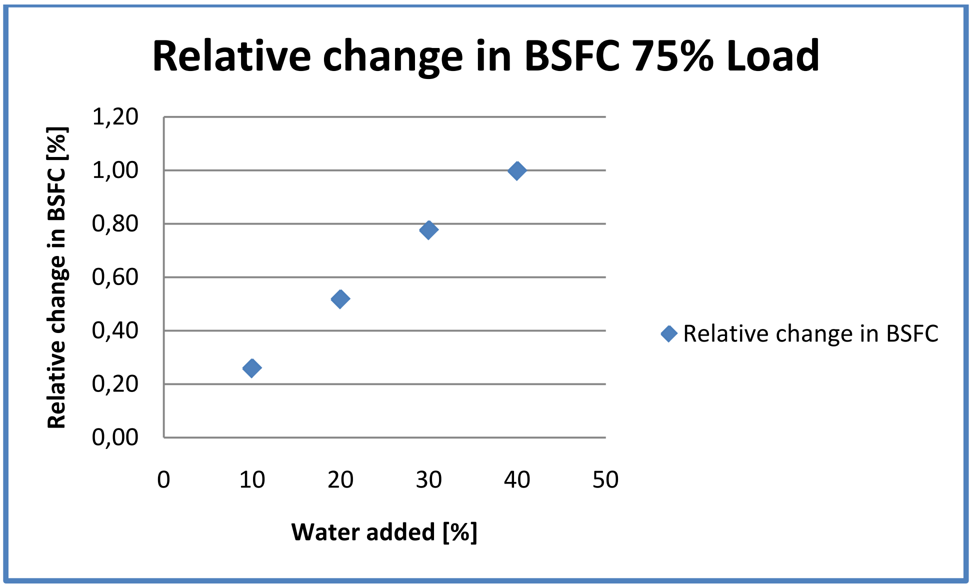

Figure 20: Relative change in BSFC 75% load.

The relative brake specific fuel consumption penalty is slightly higher than at full load.

The burned zone temperature displayed in Figure 21 has a different course for no water added than it has for 20 and 40% added. This is something I would look into if I had more time available, but for the time being I have no explanation for this phenomenon.

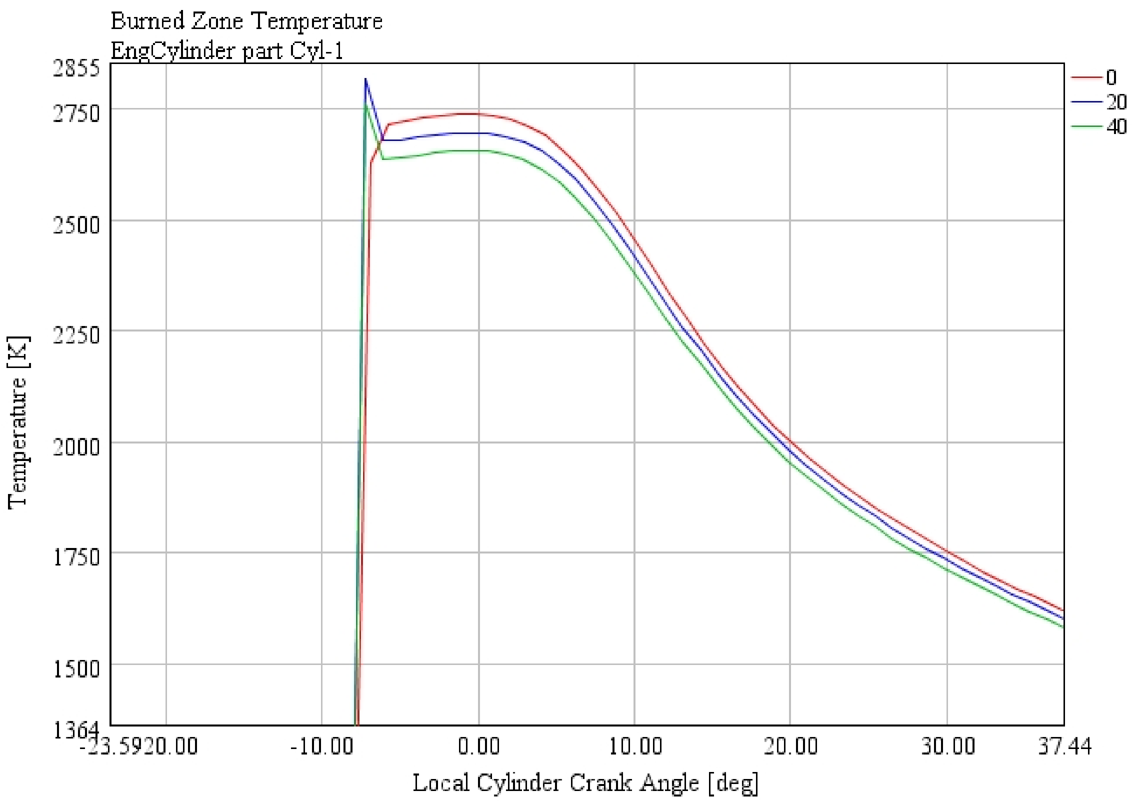

Figure 21: Burned Zone Temperature 75% load.

3.3.2.3 50% Load

The results for simulations at 50% load can be seen in Table 12. The NOx emissions have been lowered a great deal from the two other load cases.

50% Load - Power: 583 [kW]						
Water added	0	10	20	30	40	[%]
Nox	1168	1041	921	810	707	[ppm]
Indicated Specific NOx	13.4	12.0	10.7	9.5	8.4	[g/kW-h]
Reduction in NOx	0.0	1.4	2.7	3.9	5.0	[g/kW-h]
Relative red in NOx	0.0	10.2	19.9	29.0	37.6	[%]
Maximum Cylinder temp	1565	1556	1547	1538	1529	[K]
BSAC	9815	9850	9888	9923	9956	[g/kW-h]
BSFC	198.5	218.9	239.4	260	280.6	[g/kW-h]
BSFC corrected for water	198.5	199.0	199.5	200.0	200.4	[g/kW-h]
Change in BSFC	0.00	0.50	1.00	1.50	1.93	[g/kW-h]
Relative change in BSFC	0.00	0.25	0.50	0.76	0.97	[%]

Table 12: Simulation results 50% load.

The relative NOx reduction in Figure 22 is more or less in accordance with what was found in the literature study. There is about 10% reduction of specific NOx emission per 10% water added.

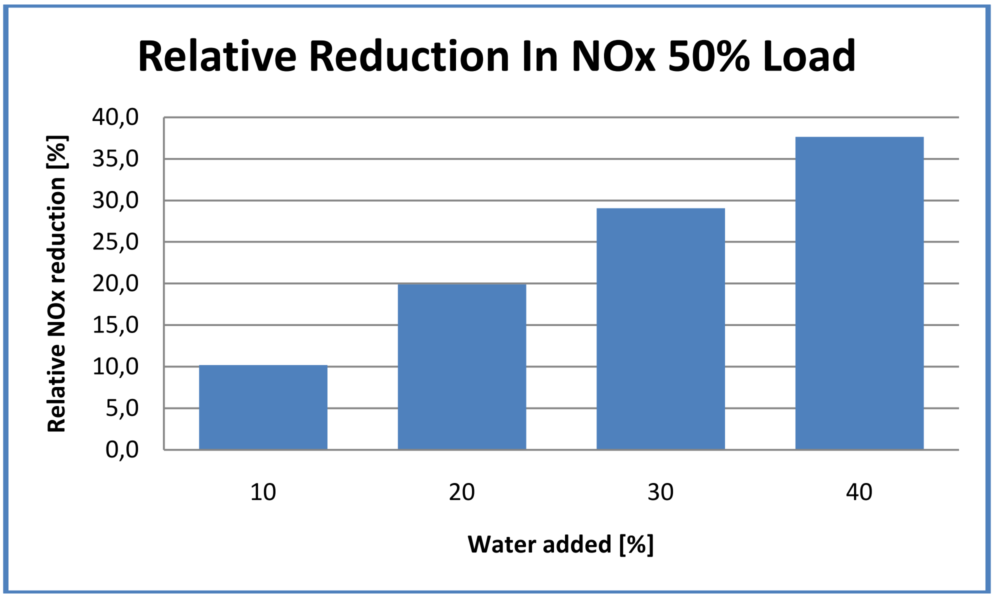

Figure 22: Relative reduction in NOx emission 50% load.

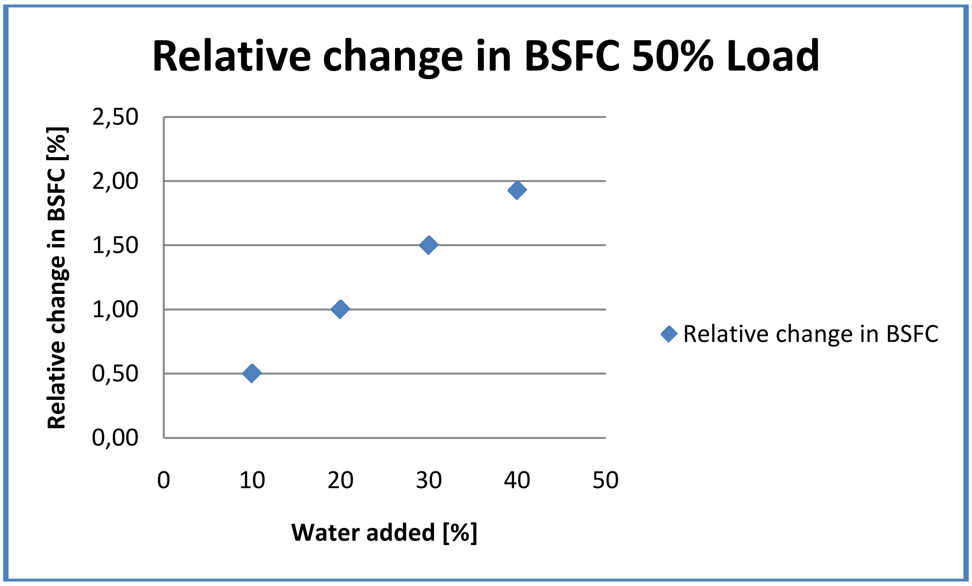

Figure 23: Relative change in BSFC 50% load.

Figure 23 displays the penalty in break specific fuel consumption. The penalty is increased compared to two other load cases.

The burned zone temperature curve in Figure 24 shows reduction in burned zone temperature with increasing water content. The burned zone temperature is lower than what is seen for the other load cases, this can explain the much lower specific NOx emission for this load case.

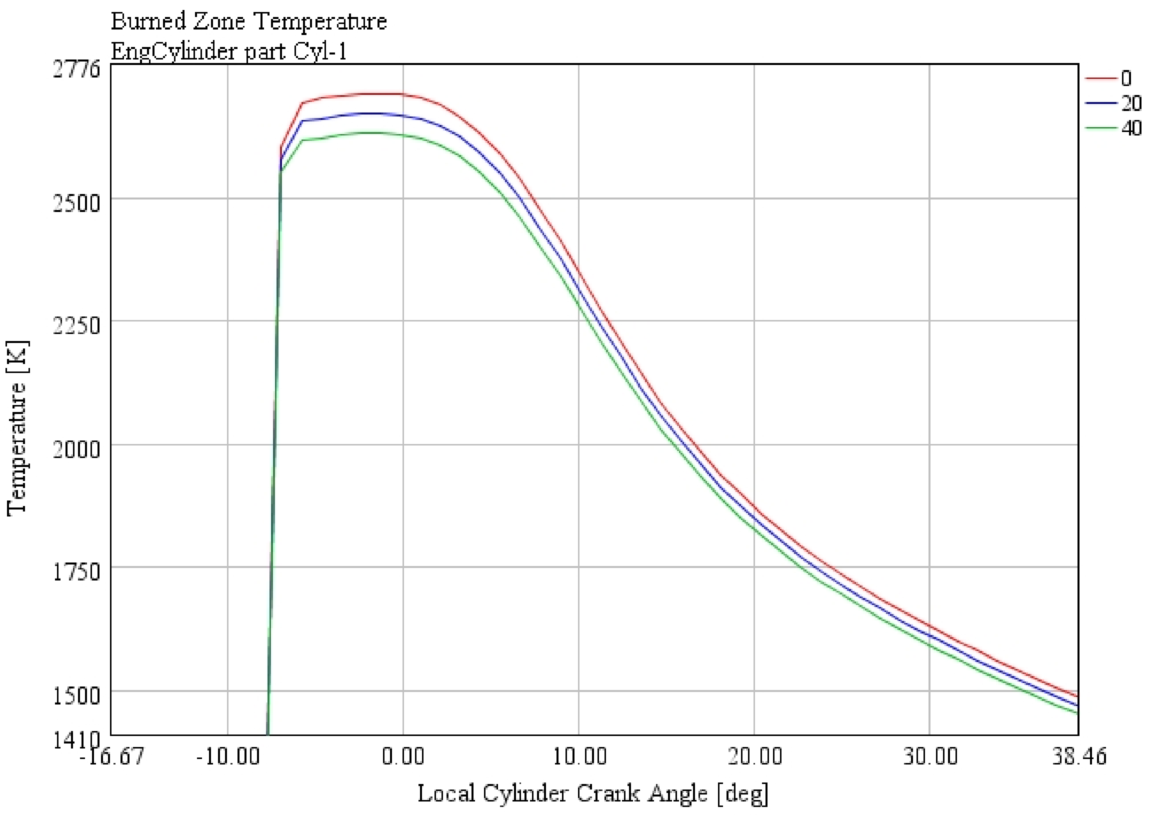

Figure 24: Burned Zone Temperature 50% load.

4 Lab

In order to investigate the effects of introducing water to the injected mass there has been preformed engine runs at MARINTEKs KR3 engine. Some data for the engine is represented in Table 13. Since this engine doesn't have an emulsion system the tests were ran using white diesel.

KR-3 Data	Value	Unit
Cylinder Bore	250	[mm]
Stroke length	300	[mm]
Stroke Volume	14.7	[l]
Number of cylinders	3	[-]
Total Stroke Volume	44.2	[l]
Compression Ratio	12.5	[-]
100% MCR Power 750 RPM	500	[kW]
100% MCR Power 900 RPM	660	[kW]

Table 13: KR3 Engine Data.

White diesel is a mix of diesel and water that has an emulsifying agent added. The emulsifying agent keeps the water and diesel from separating. The white diesel is a mix between auto diesel containing 7% biodiesel and cleaned water. The mix has also been added with an emulsifying agent in order to keep the water mixed in the diesel and a cetane booster to maintain good ignition properties. The delivered fuel totalled 1010 litres where 720 litres were diesel with specific gravity 0.83 kg/m^3. The content by mass and mass fraction can be seen in Table 14.

Content	Mass	Mass frac
	[kg]	[-]
Diesel	598	0.673
Water	266	0.299
Emulsifying agent	22	0.025
Cetane booster	3	0.003
Sum	888	1.000

Table 14: White diesel mass content.

To be able to compare the measurement with MGO, the engine was first run on MGO. Reference measurements were done at 75%, 100% and 50% load. Then there was done measurements running with white diesel. These were done at 50% and 75%. Results were obtained from the log sheet, while the particle measurements and gas emission measurements after the SCR reactor were obtained separately.

4.1 Smoke number measurement

The measurement of the smoke number is made by an AVL variable sampling smoke meter. The smoke meter is designed for automatic measurement.

The manner of operation for the smoke meter is that a defined flow rate is sampled from the engine's exhaust pipe and sucked through clean filter paper in the instrument (AVL). The filtered soot will cause blackening on the filter paper which is detected by a photoelectric measuring head and evaluated in a microprocessor to produce a result. The result is given in FSN (filter smoke number). This number is an indication on the soot content in the exhaust gas.

The blackening of the filter paper primarily depends on the soot concentration and the "effective filter length". The value 0 is assigned to clean paper while the maximum value of 10 is assigned to the absolutely black paper. Figure 25 shows the relation between "effective length, sample volume, dead volume, leak volume and filter area. The effective length is standardized by ISO DP 10054 to 405mm, the sample volume to 330cm^3 and the paper is to have a diameter of 8.15 cm^2. The dead volume is the gas way from the sampling point up to the filter paper.

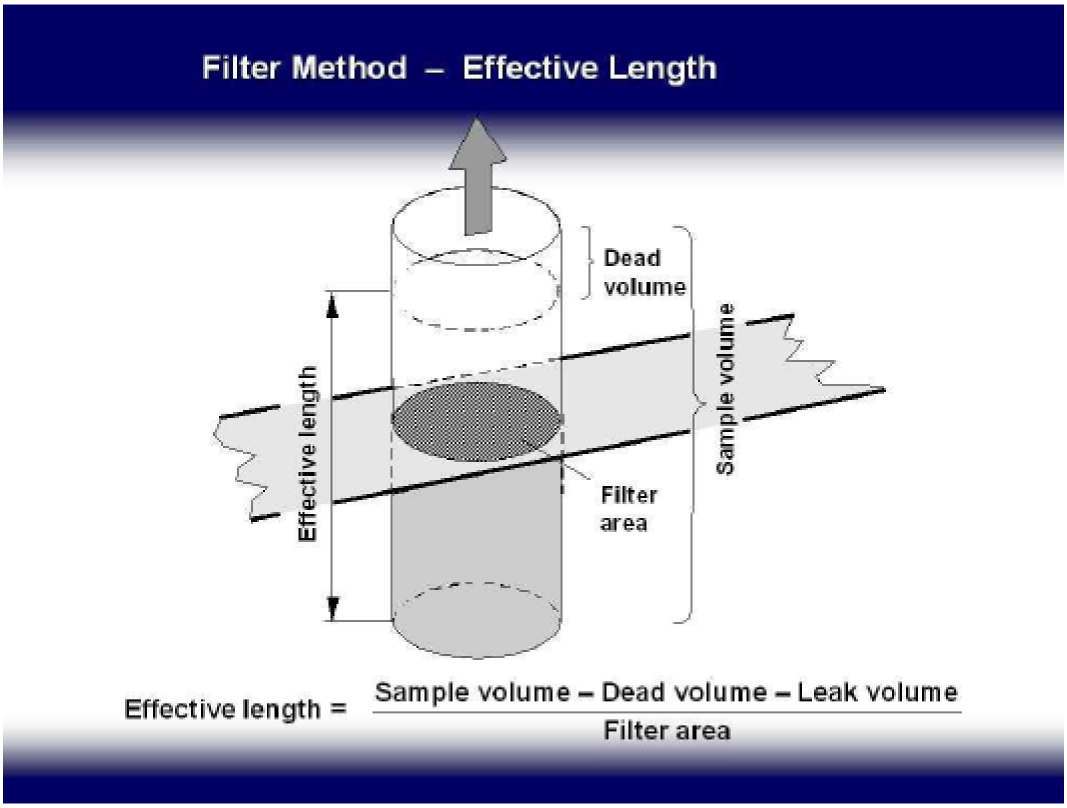

Figure 25: AVL 415S smoke meter filter method (AVL).

4.2 Particulate matter measurement

MARINTEK has two different systems for particulate measurement one DEKATI ELPI and one TSI SMPS. In this chapter the two systems, their sampling and their differences will be described.

In Figure 26 a schematic overview of the sampling is given. The exhaust is extracted from the exhaust pipe by a nozzle. The sample flows through the sample line where it is heated to a temperature of 400°C, to avoid nucleation. It arrives at the FPS 4001 dilutor where the primary dilution is done with dilution air at 400°C. The system is a two stage dilution system. The exhaust gas is first diluted through a porous tube (primary dilution) and then through an ejector dilutor (secondary dilution)(Nielsen J. B., 2009). The secondary dilutor cools the sample down to ambient temperature.

The total dilution ratio is around 50. The dilution ratio is measured by two Horiba NOx analyzers. One of them measures NOx from the diluted air and the other one measure NOx from a second sample of raw exhaust.

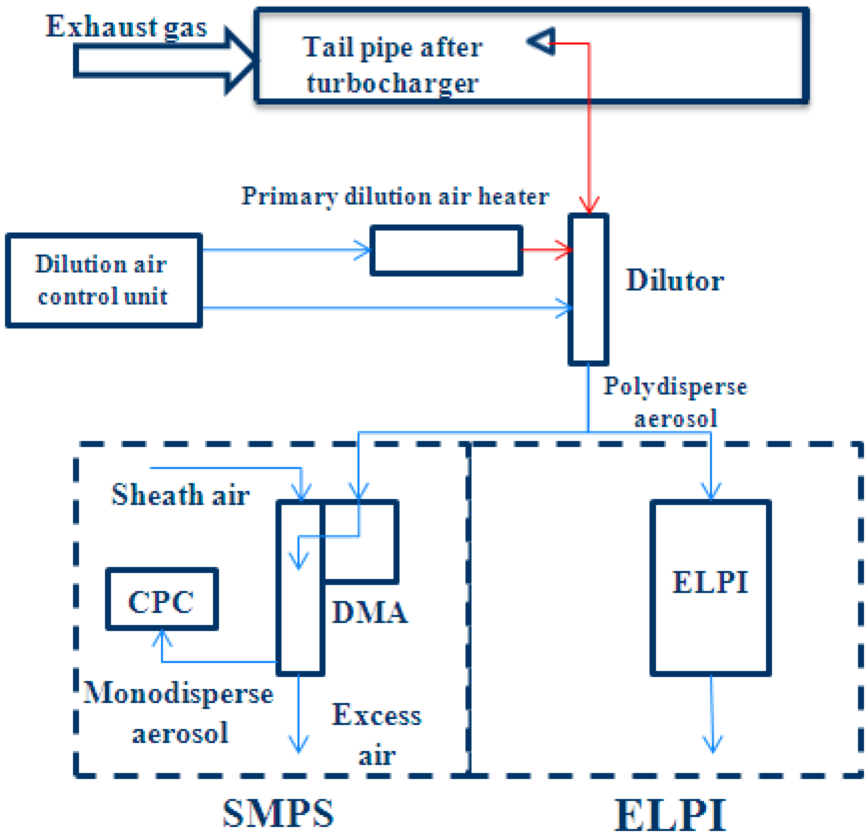

Figure 26: Sampling setup for particle measurements (Nielsen J. B., 2010).

4.2.1 DEKATI ELPI

The DEKAPI ELPI is an electrical low pressure impactor. Its operating principle can be divided into three major parts: particle charging, size classification in a cascade impactor and electrical detection with electrometers.

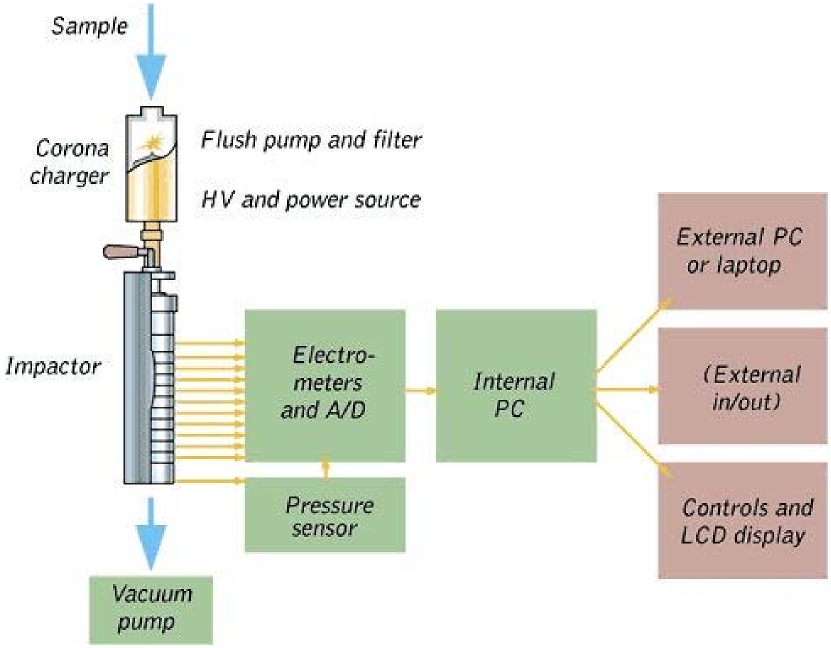

Figure 27: Schematic drawing of the ELPI (Dekati).

A unipolar corona charger gives the particles a known charge (Dekati). Later the particles enter a cascade low pressure impactor with electrically insulated collection stages. Here the particles are collected in the different impactor stages according to their aerodynamic diameter. The electrical charge of the particles at each stage is measured in real time by sensitive multichannel electrometers. The current signal from each stage is directly proportional to the particle number; this will be used to get the particle size distribution. The particles are categorized in thirteen different stages by their aerodynamic diameter. The minimum detection limit is 7 nm with a filter stage installed and the maximum detection limit is 10 µm.

The particle collection into each impactor stage is dependent on the aerodynamic size of the particles. Measured current signals are converted to (aerodynamic) size distribution using particle size dependent relations describing the properties of the charger and the impactor stages. The result is particle number concentration and size distribution in real-time.

Due to the rough resolution of the ELPI local peaks in particle size are hard to see from the results. The ELPI has a time resolution of ca. one second which makes it possible to look at transient operation.

The ELPI has a standard RS232 port that is used for communication with a computer using ELPIVI software.

4.2.2 TSI SMPS

The SMPS or Scanning Mobility Particle Sizer consists of an electrostatic classifier and a condensation particle counter (CPC). The electrostatic classifier uses a differential mobility analyser (DMA) to classify particles by their electrical mobility or mobility diameter. The CPC counts the number of particles in a monodisperse flow, a flow with particles of the same size, from the DMA.

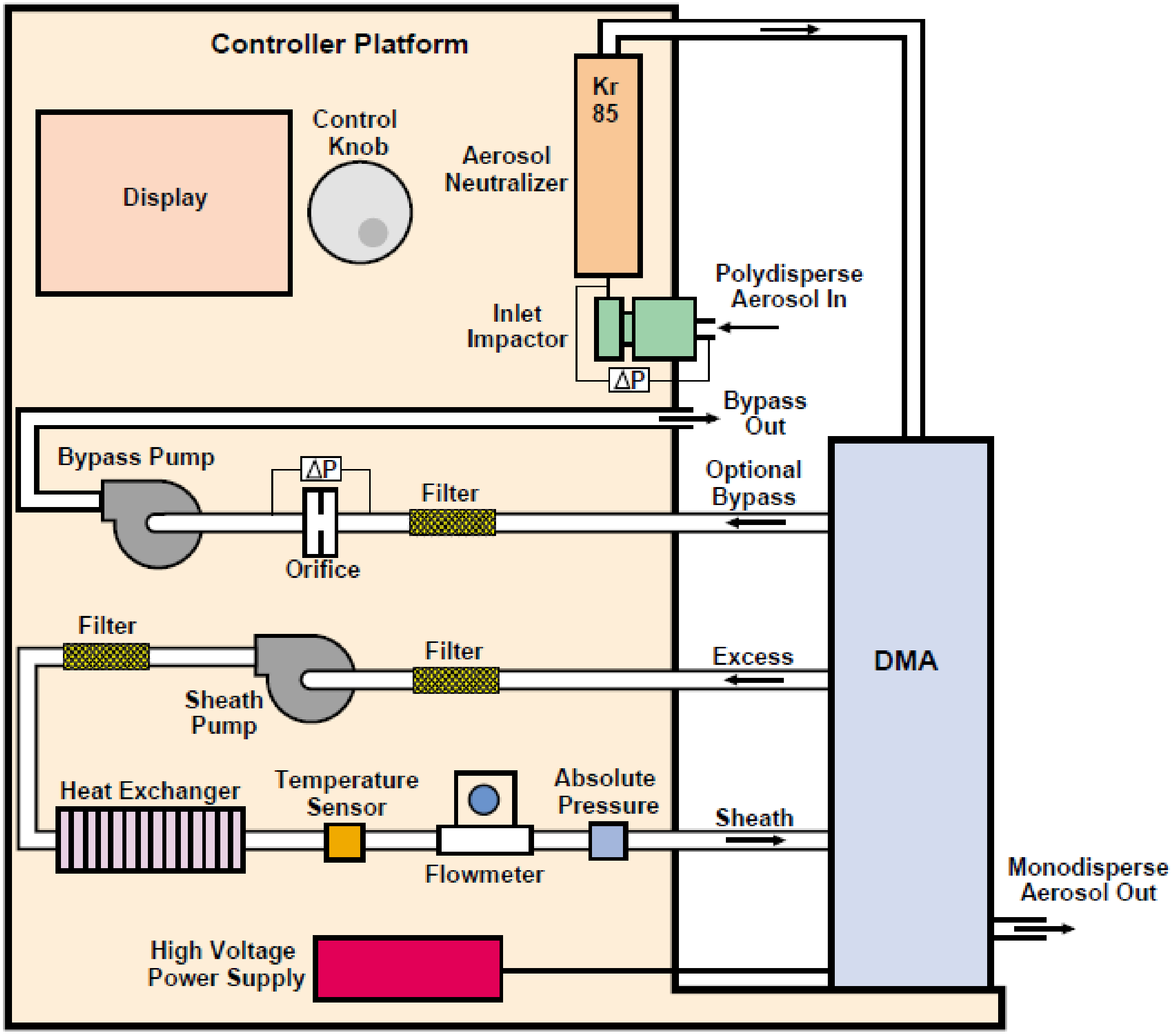

Figure 28: Schematic drawing of the 3080-series electrostatic classifier (TSI Inc.).

Before the exhaust gas sample can enter the electrostatic classifier it has been diluted and conditioned to acceptable levels for the system. Particles larger than the detection limit are removed by an impactor at the inlet see Figure 28.

The diluted exhaust flow is passed through a radioactive source, an aerosol neutralizer, to create a monodisperse flow. Particles in the exhaust flow are often charged from the generation process or from contact electrification. The radioactive source neutralizes the charged particles by bombarding them with bipolar ions. After passing the radioactive source the particles will have a known bipolar charge distribution which is used to determine the particle size distribution. Next the exhaust sample is feed into the DMA. The DMA is a long vertical cylinder where the exhaust is feed at the top.

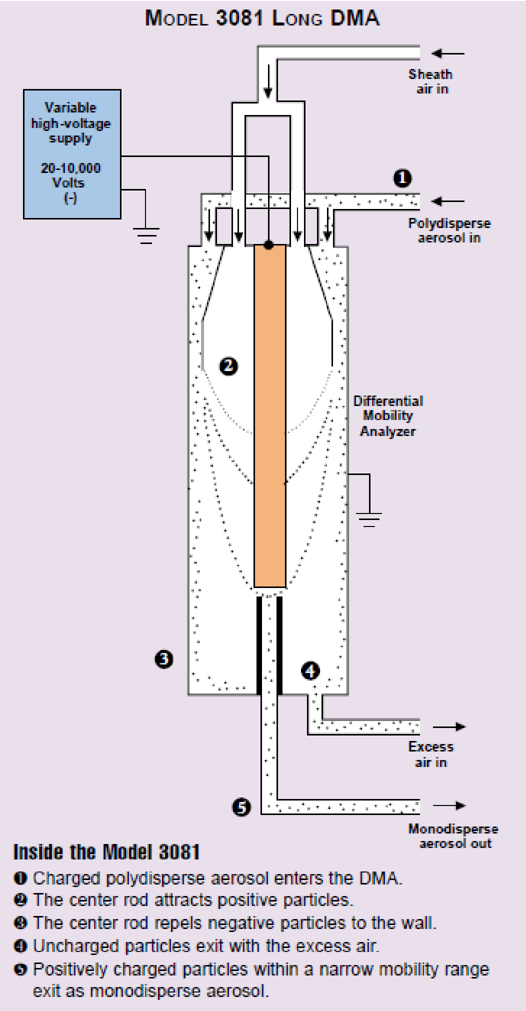

Figure 29: TSI DMA (TSI Inc.).

Figure 29 shows that the sheet air forces the exhaust to flow downward in a thin layer on the outer wall of the DMA. The sheet air and aerosol must merge without mixing. In the middle of the DMA there is a high voltage rod that sets up an electrical field. The electrical field causes the bipolar charged particles to drift inwards through the clean sheet air. The particles with higher electric mobility will deposit on the rod upstream while particles with lower mobility will exit with the excess air.

The electrostatic classifier and DMA can deliver a monodisperse with a size range from 10 to 1000 nm. In order to change the particle size of the mondisperse flow the voltage of the rod is changed. Size range is also dependent on the combination of sheath flow rate and sample flow rate. The sample flow rate is controlled by the CPC.

The CPC is a particle counter. By itself it only gives the total particle consentration. It uses an optic sensor and laser light to detect the particles. By them self particles are too small to be detected by the optics so the sample aerosol is passed through a heated saturator. In the saturator a cylindrical wick evaporate buthanol in to the sample flow. Then the sample enter a condenser witch cools the sample down to 14°C. With dropping temperature the buthanol vapour condenses on the particles, this makes the particles grow into a detectable size.

Depending on the number of particles in the flow, the CPC will either count particles individually or by light scattering to calculate the total number of particles.

In order to get the size distribution the electrostatic classifier and the CPC works together. This instrument setup is called SMPS spectrometer by TSI. The DMA has to scan through the complete size range and the CPC has to count the particles for the complete range. Scanning through the complete size range takes about 3 minutes. This means that the size distribution is updated every 3 minutes.

The SMPS can typically give size distribution results in a range from 10 to 700 nm. The number of channels is typically over 100.

4.2.3 Comparison between the DEKATI ELPI and the TSI SMPS

The ELPI and the SMPS are two very different instruments for measuring the particle emission. The ELPI is a simpler instrument that gives a much rougher distribution than the SMPS. On the other hand the SMPS gives a very limit distribution in terms of range.

The resolution of the SMPS depends on the size range it operates in. It will typically have more than 100 channels in a size range of 10 to 700nm. The resolution is much more detailed than the DEKATI ELPI which has 13 channels over a size range from 7nm to 10μm, but the range is more limited.

When it comes the time it takes to get a scan of the complete size distribution, the ELPI is quick and uses about one second to scan the complete distribution. This is due to the relatively simple method it uses. The SMPS on the other hand has to go through the different channels one by one. This process takes about 3 minutes.

This means that if looking at transient operation is of importance the ELPI is the instrument of choice. If more detailed resolution is needed the SMPS is a better choice, but one has to keep in mind that the range of the scanning is more limited than the ELPI.

4.3 Lab Results

The results I have looked into are the results for specific NOx emissions, the smoke number and the particulate matter size distribution, concentration and mass. These results have been compared to see the difference between running on MGO and white diesel.

4.3.1 NOx

In Table 15 the result for NOx measurements is given. The measurements were received from the engine test data log sheet. It is unclear whether the results from these measurements are correct considering the extra water that is added to the combustion chamber. It is possible that the ISO calculations of exhaust flow do not take into account the extra water that is introduced to the combustion chamber.

NOx Reduction			
Load		50%	75%
MGO	[g/kWh]	13.7	12.8
White diesel	[g/kWh]	11.3	9.7
Reduction	[g/kWh]	2.4	3.1
Reduction	[%]	17.5	24.2

Table 15: NOx reduction.

The results show a reduction of 24% when the engine is running at 75% load and 17.5% at 50% load. The mass fraction of water in the diesel is quite high at 29.9% and the results for NOx-reduction seams quite low when compared to the results from the literature study.

4.3.2 Smoke number

From the measurement result in AppendixF: we can extract the data in Table 16. Here the FSN, filter smoke number, shows a high reduction when running on white diesel. The smoke number is an indication on the soot content of the exhaust gas. So these results will mean there is a reduction of soot particles in the exhaust gas. This is accordance to the literature (Ki Hoon Song, 2000). But a reduction as high as 80-88% seams quite high.

Smoke Number			
Load		50%	75%
MGO	[-]	0.10	0.169
White diesel	[-]	0.02	0.020
Reduction	[-]	0.08	0.149
Reduction	[%]	0.80	0.88

Table 16: Smoke Number.

4.3.3 Particulate matter size distribution

In Equation 11 the unit for all the particulate matter size distribution graphs is given. This is the derivative of the cumulative distribution with respect to diameter. The unit is one over cubic centimetres.

$$\frac{dN}{d\log d_p}[\frac{1}{cm^3}]$$

Equation 11: Unit for PM distribution.

4.3.3.1 50% load particle matter size distribution

The results from the SMPS measurements in Figure 30 for 50% load shows a very high peak in particulate matter when running on white diesel compared to running on MGO. The peak is in the size range 10-40nm with maximum value around 20nm. This is the range of nucleation particles.

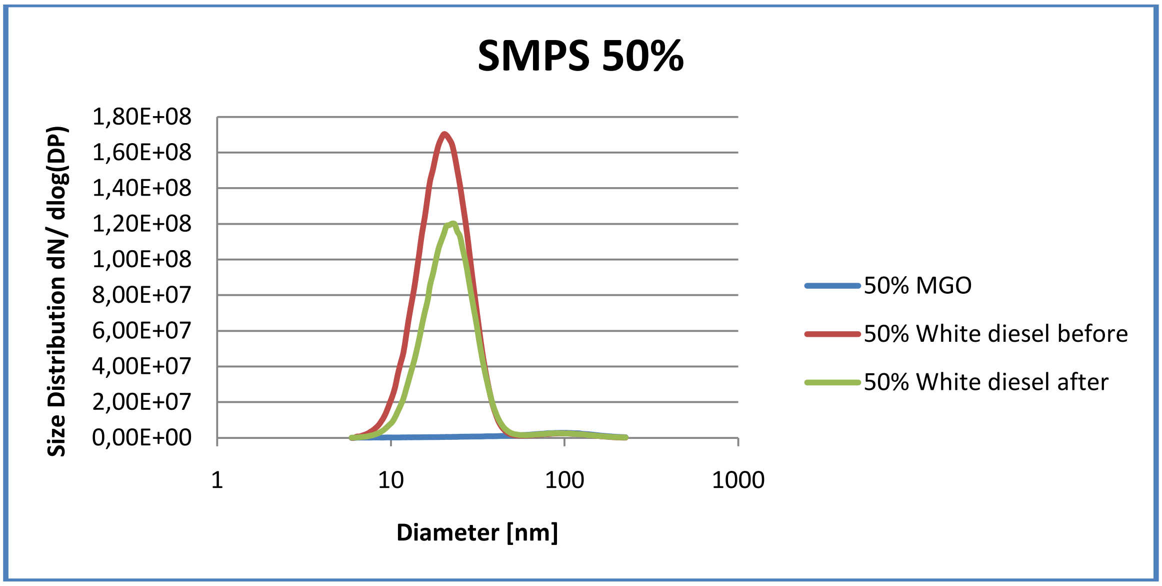

Figure 30: SMPS 50% load.

The ELPI results for 50% load in Figure 31 confirm the results from the SMPS. There is the same peak in particles with diameter around 20nm.

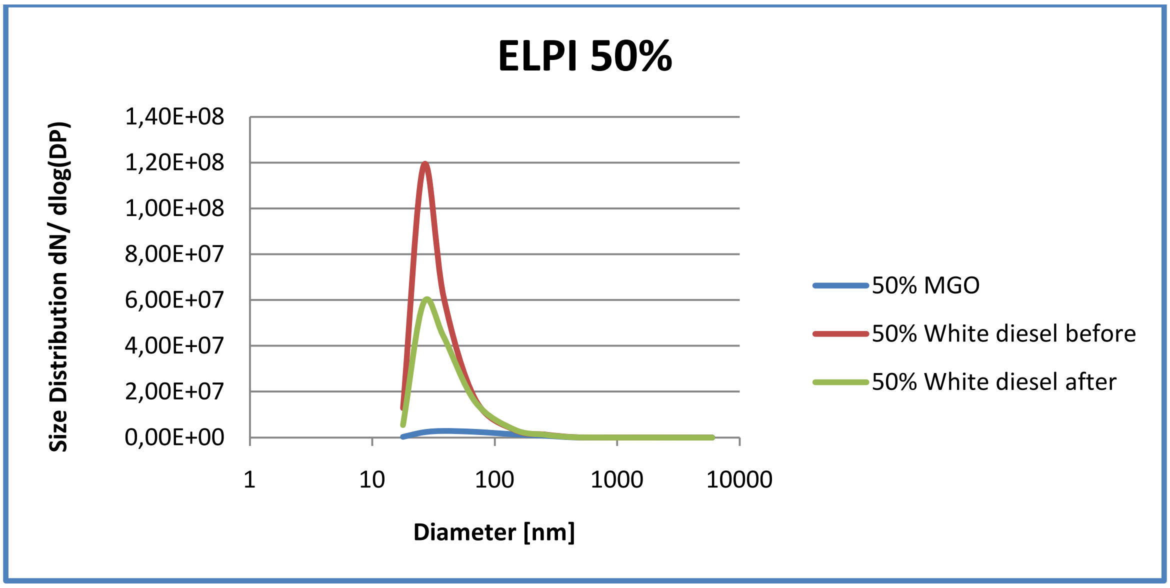

Figure 31: ELPI 50% load.

Figure 32 shows the results from Figure 30, but zoomed in to show the peak in accumulation particles with diameter around 80nm. The figure shows an almost unchanged amount of particles in the accumulation range when running on white diesel. This is a strange result when comparing it to the smoke number as the smoke number shows a reduction in soot, while the accumulation particles with peak value around 80nm show almost only a small reduction.

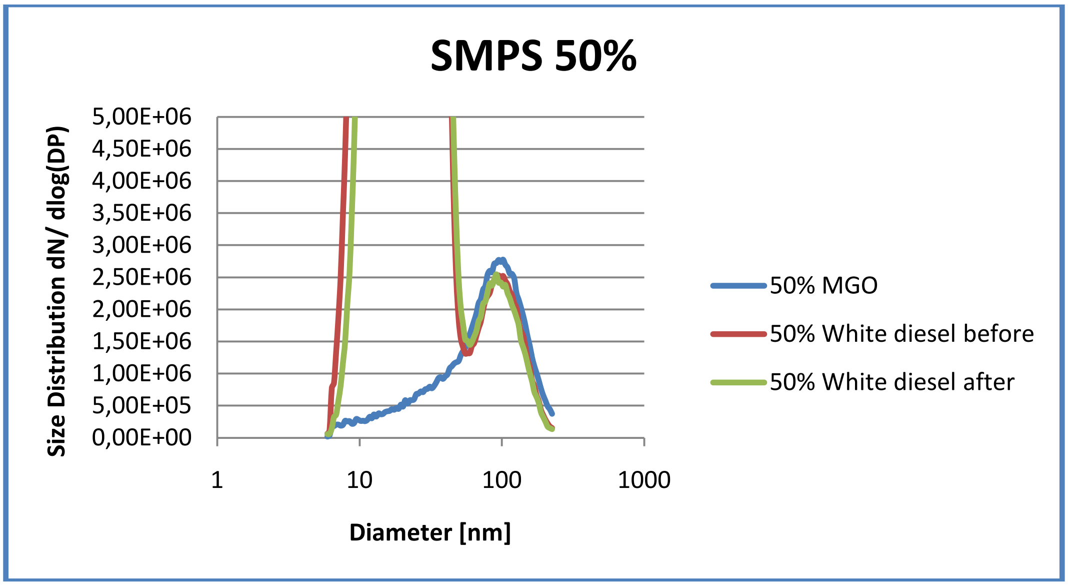

Figure 32: SMPS 50% load zoomed.

Figure 33 shows the results from the SMPS measurements at 75% load. As with the 50% load measurements there is a large peak in nucleation mode particles. The peak in the size distribution is here also around 20nm.

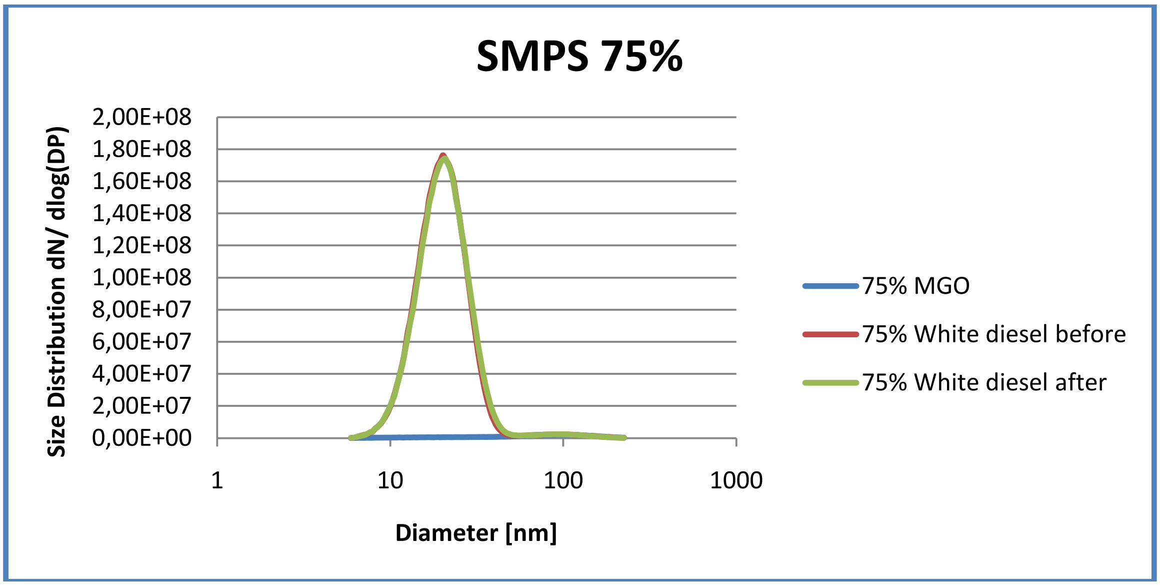

Figure 33: SMPS 75% load.

The results from the ELPI in Figure 34 can confirm the results from the SMPS there is here also a peak around 20nm.

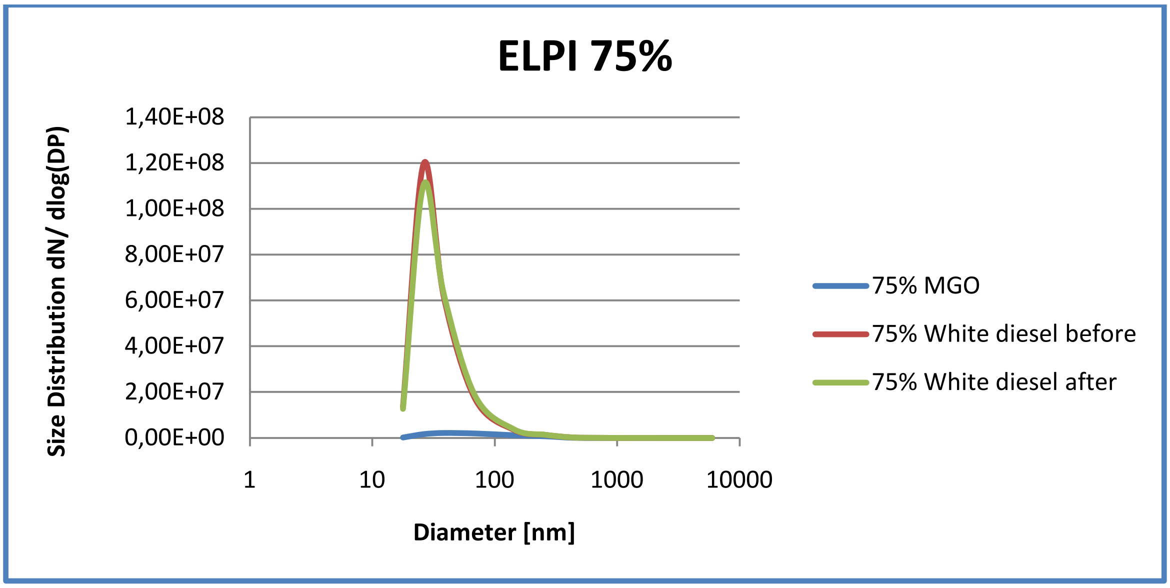

Figure 34: ELPI 75% load

Figure 35 shows Figure 33 zoomed in to look at the difference in accumulation mode particles. As with the 50% load measurements, we find a small change in the size distribution around 80nm diameter. The accumulation mode particles actually have higher values in size distribution in this size range.

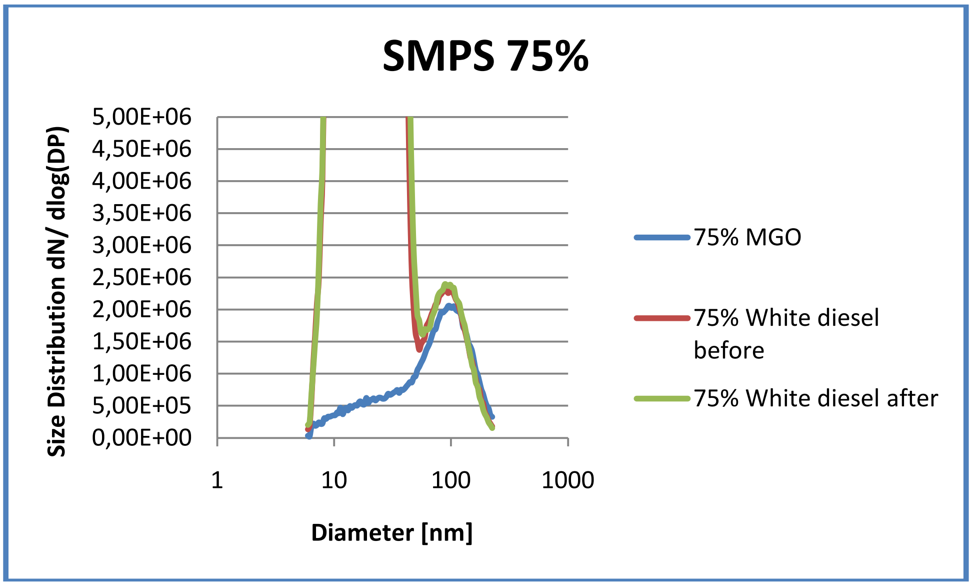

Figure 35: SMPS 75% load zoomed.

In Table 17 the size distribution values for the ELPI are given. This table is interesting for looking at larger particles that are outside of the range of the SMPS.

By studying the data it is possible to see that the size distribution values for white diesel actually is higher for all of the different levels in the ELPI measurement.

DEKATI ELPI Size Distribution Measurement								
Load	Fuel	Diameter versus size distribution						
		17.8	26.3	38.3	69.3	144.1	260.1	nm
50%	MGO	2.84E+05	2.27E+06	2.87E+06	2.49E+06	1.41E+06	7.52E+05	1/cm^3
50%	WD	1.29E+07	1.19E+08	6.12E+07	1.68E+07	3.41E+06	1.33E+06	1/cm^3
75%	MGO	2.06E+05	1.68E+06	2.20E+06	1.99E+06	1.20E+06	7.37E+05	1/cm^3
75%	WD	1.36E+07	1.20E+08	6.18E+07	1.70E+07	3.49E+06	1.36E+06	1/cm^3
		430.5	722.3	1166.4	1852.4	2928.0	5962.9	nm
50%	MGO	7.23E+04	1.06E+04	2.18E+03	9.38E+02	3.08E+02	9.30E+01	1/cm^3
50%	WD	2.43E+05	8.17E+04	3.26E+04	2.12E+04	6.94E+03	1.92E+03	1/cm^3
75%	MGO	1.10E+05	2.37E+04	3.69E+03	1.05E+03	3.14E+02	8.61E+01	1/cm^3
75%	WD	2.51E+05	8.58E+04	3.40E+04	2.18E+04	7.13E+03	2.00E+03	1/cm^3

Table 17: DEKATI ELPI diameter versus size distribution.

4.3.4 Particulate matter concentration and mass

Fuel	SCR	Load	PM Concentration	PM mass
			[1/cm^3]	[mg/m^3]
MGO	Before	50%	2.39E+06	9.2393
MGO	Before	75%	1.93E+06	10.0371
White Diesel	Before	50%	3.99E+07	134.1119
White Diesel	Before	75%	4.07E+07	139.2611
White Diesel	After	50%	2.52E+07	101.5676
White Diesel	After	75%	3.98E+07	142.4830

Table 18: PM concentration and mass.

In Table 18 the total PM concentration and mass from the engine particulate measurements are represented. From this data is it clear that the particulate matter has increased both in concentration and mass compared to running on MGO.

5 Conclusion

When it comes to operational aspects of water in fuel emulsion there are some concerns that needs to be addressed. For one there is a need for a “fail safe” system to makes sure that water in fuel emulsion is not left in the system when failures occur. Water left in the fuel system will gradually separate from the fuel and can cause corrosion. Such a system needs to be separate from the dosage system, because if the dosage system fails it is important to have this system intact. It should also be able to shut down if the dosage system fails and gives wrong dosage of water into the fuel.

GT- Power can be used for simulations with water in the fuel. There are some modifications that need to be done in order to get results. One needs to create a combine item to combine water and fuel, one need to choose to calculate the temperature by a two zone model and one need to activate the NOx plug-in. One cannot expect to get good results without having enough data to create good models. Such data was not available during my work so my results cannot be seen as realistic simulation results. Sill reasonable results were found for relative reduction of NOx emissions.

The engine runs from running MARINTEKS KR-3 engine on white diesel have resulted in NOx-reductions that are lower than what was expected. The results with regards to smoke number show a high degree of reduction in soot. While the particulate matter measurements does not back this up.

The particulate matter measurement results on the size distribution show very high peaks in nucleation mode particles. The peak of the size distribution curve is at around 20nm in diameter. There are also peaks around the size where we find accumulation mode particles. This peak is about the same size both when running on MGO and white diesel. When looking at even bigger particles from the ELPI size distribution measurements we observe that running on white diesel gives the largest number of particles in this size range as well.

These results are somewhat contradictory as the smoke number is lowered while the size distribution, total concentration and mass pr. cubic meter exhaust is higher when running on white diesel.

It is not clear if these results would be the same for water in fuel emulsion. But these results are not good news for those who see white diesel as a solution for reducing NOx. The adverse health effects of particulate emissions have been studied and it is clear that especially the smaller particulates can be dangerous; it is therefore bad news that running on white diesel creates such a high peak in the size distribution around 20nm.

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Table of Appendix

AppendixA: GT-Power engine performance predictions

Engine performance predictions for 100% load					
	Case 1	Case 2	Case 3	Case 4	Case 5
Brake Power [kW]	1165	1165	1165	1165	1165
Brake Power [HP]	1563	1563	1563	1563	1563
Brake Torque [N-m]	15454	15454	15453	15454	15453
IMEP [bar]	23.54	23.54	23.54	23.54	23.54
FMEP [bar]	1.56	1.56	1.56	1.56	1.56
PMEP [bar]	0.00	0.00	0.00	-0.01	-0.01
Air Flow Rate [kg/hr]	9359	9388	9417	9446	9473
BSAC [g/kW-h]	8032	8057	8082	8107	8130
Fuel Flow Rate [kg/hr]	222	244	267	290	313
BSFC [g/kW-h]	190	210	229	249	268
Volumetric Efficiency [%]	399	401	402	403	404
Volumetric Efficiency (M) [%]	131	131	131	131	131
Trapping Ratio	0.78	0.79	0.79	0.79	0.79
A/F Ratio	42.2	38.4	35.2	32.6	30.3
Brake Efficiency [%]	44.0	44.2	44.3	44.5	44.7

Engine performance predictions for 75% load					
	Case 1	Case 2	Case 3	Case 4	Case 5
Brake Power [kW]	874	874	874	874	874
Brake Power [HP]	1172	1172	1172	1172	1172
Brake Torque [N-m]	11591	11591	11594	11594	11593
IMEP [bar]	17.89	17.89	17.89	17.89	17.89
FMEP [bar]	1.41	1.41	1.40	1.40	1.41
PMEP [bar]	-0.01	-0.02	-0.02	-0.02	-0.03
Air Flow Rate [kg/hr]	7422	7446	7437	7455	7480
BSAC [g/kW-h]	8492	8521	8508	8528	8558
Fuel Flow Rate [kg/hr]	169	187	204	222	239
BSFC [g/kW-h]	194	213	233	254	274
Volumetric Efficiency [%]	317	318	317	318	319
Volumetric Efficiency (M) [%]	129	129	129	129	129
Trapping Ratio	0.79	0.79	0.79	0.79	0.79
A/F Ratio	43.9	39.9	36.5	33.6	31.3
Brake Efficiency [%]	43.3	43.4	43.6	43.7	43.9

Engine performance predictions for 50% load					
	Case 1	Case 2	Case 3	Case 4	Case 5
Brake Power [kW]	583	583	583	583	583
Brake Power [HP]	782	782	782	782	782
Brake Torque [N-m]	7732	7732	7732	7732	7732
IMEP [bar]	12.26	12.26	12.26	12.26	12.26
FMEP [bar]	1.26	1.26	1.26	1.26	1.26
PMEP [bar]	-0.02	-0.02	-0.02	-0.02	-0.02
Air Flow Rate [kg/hr]	5722	5742	5764	5785	5804
BSAC [g/kW-h]	9815	9850	9888	9923	9956
Fuel Flow Rate [kg/hr]	116	128	140	152	164
BSFC [g/kW-h]	199	219	239	260	281
Volumetric Efficiency [%]	244	245	246	247	248
Volumetric Efficiency (M) [%]	127	127	127	127	127
Trapping Ratio	0.80	0.80	0.80	0.80	0.80
A/F Ratio	49.5	45.0	41.3	38.2	35.5
Brake Efficiency [%]	42.2	42.3	42.5	42.6	42.8

AppendixB: GT-Power key cylinder predictions for 100% load

Key Cylinder Predictions for 100% Load					
Case #	Case 1	Case 2	Case 3	Case 4	Case 5
Part Name	Cylinder Average	Cylinder Average	Cylinder Average	Cylinder Average	Cylinder Average
Volumetric Efficiency [%]	399	401	402	403	404
Volumetric Efficiency (m) [%]	131	131	131	131	131
Trapping Ratio	0.785	0.785	0.786	0.786	0.786
Burned Residuals Mass (SOC) [%]	0.10	0.10	0.10	0.10	0.10
EGR [%]	0	0	0	0	0
F/A Ratio (trapped)	0.030	0.033	0.036	0.039	0.042
Lambda, Effective	2.310	2.280	2.252	2.226	2.200
IMEP [bar]	23.54	23.54	23.54	23.54	23.54
PMEP [bar]	0.00	0.00	0.00	-0.01	-0.01
ISFC [g/kW-h]	178	196	214	232	251
Indicated Efficiency [%]	47	47	47	48	48
Fuel Mass [mg]	1711	1886	2062	2236	2413
Maximum Pressure [bar]	172	172	172	172	172
CA at Max. Pressure [deg]	12.98	12.98	12.98	12.98	12.98
dPmx/DCA [bar/deg]	4.15	4.13	4.11	4.11	4.09
Maximum Temperature [K]	1725	1714	1703	1693	1682
Intake Pressure [bar]	3.52	3.53	3.54	3.56	3.57
Intake Temperature [K]	330	330	330	330	330
Exhaust Pressure [bar]	3.18	3.20	3.21	3.23	3.24
Exhaust Temperature [K]	653	650	648	645	643
Heat.Tr. (frac. of F.E) [%]	13.32	13.25	13.17	13.10	13.02
Swirl at TDC	0	0	0	0	0
Swirl at BDC	0	0	0	0	0
NOx in ppm	2425	2204	1992	1794	1605
Indicated Specific NO2 [g/kW-h]	23.35	21.40	19.50	17.71	15.97
Soot Concentration @ STP [g/m^3]	0	0	0	0	0
Indicated Specific Soot [g/kW-h]	0	0	0	0	0
HC in ppm	0.463	0.935	1.405	1.865	2.323
Indicated Specific HC [g/kW-h]	0.020	0.020	0.020	0.020	0.030
CO in ppm	0.930	0.922	0.915	0.910	0.905
Indicated Specific CO [g/kW-h]	0.010	0.010	0.010	0.010	0.010
CO2 in ppm	61686	61291	60897	60473	60104
Indicated Specific CO2 [g/kW-h]	568	569	570	570	572
Knocking zones	0	0	0	0	0

AppendixC: GT-Power key cylinder predictions for 75% load

Key Cylinder Predictions for 75% Load					
Case #	Case 1	Case 2	Case 3	Case 4	Case 5
Part Name	Cylinder Average	Cylinder Average	Cylinder Average	Cylinder Average	Cylinder Average
Volumetric Efficiency [%]	317	318	317	318	319
Volumetric Efficiency (m) [%]	129	129	129	129	129
Trapping Ratio	0.790	0.790	0.791	0.791	0.792
Burned Residuals Mass (SOC) [%]	0.10	0.10	0.12	0.12	0.12
EGR [%]	0	0	0	0	0
F/A Ratio (trapped)	0.029	0.032	0.035	0.038	0.040
Lambda, Effective	2.420	2.387	2.343	2.311	2.284
IMEP [bar]	17.89	17.89	17.89	17.89	17.89
PMEP [bar]	-0.02	-0.02	-0.02	-0.03	-0.03
ISFC [g/kW-h]	178	197	215	234	252
Indicated Efficiency [%]	47	47	47	47	48
Fuel Mass [mg]	1305	1439	1575	1710	1845
Maximum Pressure [bar]	132	132	132	132	132
CA at Max. Pressure [deg]	12.85	12.90	12.95	13.10	13.13
dPmx/DCA [bar/deg]	3.13	3.12	3.11	3.09	3.08
Maximum Temperature [K]	1675	1664	1658	1648	1638
Intake Pressure [bar]	2.76	2.77	2.77	2.78	2.79
Intake Temperature [K]	323	323	323	323	323
Exhaust Pressure [bar]	2.51	2.52	2.53	2.54	2.55
Exhaust Temperature [K]	636	634	633	631	629
Heat.Tr. (frac. of F.E) [%]	13.65	13.55	13.47	13.37	13.30
Swirl at TDC	0	0	0	0	0
Swirl at BDC	0	0	0	0	0
NOx in ppm	1898	1706	1546	1377	1217
Indicated Specific NO2 [g/kW-h]	19.19	17.39	15.82	14.19	12.65
Soot Concentration @ STP [g/m^3]	0	0	0	0	0
Indicated Specific Soot [g/kW-h]	0	0	0	0	0
HC in ppm	0.445	0.898	1.357	1.808	2.252
Indicated Specific HC [g/kW-h]	0.020	0.020	0.020	0.020	0.030
CO in ppm	0.993	0.990	0.990	0.985	0.980
Indicated Specific CO [g/kW-h]	0.010	0.010	0.010	0.010	0.010
CO2 in ppm	58946	58612	58549	58273	57902
Indicated Specific CO2 [g/kW-h]	570	571	572	574	575
Knocking zones	0	0	0	0	0

AppendixD: GT-Power key cylinder predictions for 50% load

Key Cylinder Predictions for 50% Load					
Cylinder #	Case 1	Case 2	Case 3	Case 4	Case 5
Part Name	Cylinder Average	Cylinder Average	Cylinder Average	Cylinder Average	Cylinder Average
Volumetric Efficiency [%]	244	245	246	247	248
Volumetric Efficiency (m) [%]	127	127	127	127	127
Trapping Ratio	0.799	0.799	0.799	0.799	0.800
Burned Residuals Mass (SOC) [%]	0.13	0.13	0.13	0.13	0.13
EGR [%]	0	0	0	0	0
F/A Ratio (trapped)	0.025	0.028	0.030	0.033	0.035
Lambda, Effective	2.758	2.715	2.676	2.637	2.602
IMEP [bar]	12.26	12.26	12.26	12.26	12.26
PMEP [bar]	-0.02	-0.02	-0.02	-0.02	-0.02
ISFC [g/kW-h]	178	196	215	233	252
Indicated Efficiency [%]	47	47	47	48	48
Fuel Mass [mg]	893	985	1077	1170	1262
Maximum Pressure [bar]	97	97	97	97	97
CA at Max. Pressure [deg]	12.33	12.48	12.57	12.62	12.62
dPmx/DCA [bar/deg]	2.16	2.15	2.14	2.13	2.12
Maximum Temperature [K]	1565	1556	1547	1538	1529
Intake Pressure [bar]	2.12	2.13	2.14	2.14	2.15
Intake Temperature [K]	317	317	317	317	317
Exhaust Pressure [bar]	1.94	1.94	1.95	1.96	1.97
Exhaust Temperature [K]	601	599	597	595	593
Heat.Tr. (frac. of F.E) [%]	13.95	13.83	13.73	13.62	13.53
Swirl at TDC	0	0	0	0	0
Swirl at BDC	0	0	0	0	0
NOx in ppm	1168	1041	921	810	707
Indicated Specific NO2 [g/kW-h]	13.39	12.02	10.73	9.50	8.35
Soot Concentration @ STP [g/m^3]	0	0	0	0	0
Indicated Specific Soot [g/kW-h]	0	0	0	0	0
HC in ppm	0.390	0.790	1.185	1.580	1.968
Indicated Specific HC [g/kW-h]	0.020	0.020	0.020	0.020	0.030
CO in ppm	0.992	0.987	0.982	0.978	0.973
Indicated Specific CO [g/kW-h]	0.010	0.010	0.010	0.010	0.010
CO2 in ppm	51904	51649	51353	51100	50827
Indicated Specific CO2 [g/kW-h]	569	570	571	573	574
Knocking zones	0	0	0	0	0

AppendixF: Lab results from running on white diesel

In this table all the results from running the engine first on MGO for reference measurements and then on white diesel is represented. This excludes particulate measurements and emission gas measurements after the SCR. The measurements 1-3 are respectively from running on MGO and 4 and 5 are from running on white diesel. There were done two measurements with white diesel, respectively 50% and 75% load.

Measurement nr.		1	2	3	4	5
Date		05/05/2010	05/05/2010	05/05/2010	05/05/2010	05/05/2010
Time		12:17	12:49	13:17	15:07	15:40
Fuel		MGO			White Diesel	
x_water	[-]	0	0	0	0.299	0.299
x_diesel	[-]	1	1	1	0.673	0.673
Engine Performance						
Load		75%	100%	50%	50%	75%
Engine Speed	[rpm]	748	750	753	750	750
Torque	[Nm]	4795	6419	3202	3207	4790
Power	[kW]	375.8	504	252.5	251.9	376.2
Power iso	[kW]	372.7	497.9	249.7	247.8	368.6
BMEP	[bar]	13.7	18.3	9.1	9.1	13.6
Effective eff	[%]	42.7	42.6	41.4	29.5	29.6
Vol eff	[%]	130.4	129.9	127.2	126.4	130.3
Adiabatic eff comp	[%]	73.6	75	69.9	69.3	73.4
Turbine Speed	[rpm]	49221	57220	39445	38550	48794
Fuel consumpt	[g/s]	20.5	27.5	14.2	19.9	29.5
SFC	[g/kWh]	196.1	196.7	202.2	284	282.6
Energy consumption	[MJ/kWh]	8.43	8.46	8.7	12.21	12.15
Energy consumption iso	[MJ/kWh]	8.36	8.36	8.6	12.01	11.91
Hn diesel	[MJ/kg]	43	43	43	43	43
Air – Exhaust						
Temp b Compressor	[C]	26.1	27.1	27.1	28	28.7
Temp a Compressor	[C]	127.8	163.6	91.9	89.9	128.3
Temp Airreceiver	[C]	42.6	48.1	37.3	37.3	42.1
Exhaust Temp Cyl 1	[C]	336	372	309	296	327
Exhaust Temp Cyl 2	[C]	344	378	318	308	334
Exhaust Temp Cyl 3	[C]	340	377	309	306	336
Exhaust Temp b Turbine	[C]	430	472	394	380	420
Exhaust Temp a Turbine	[C]	324	331	320	310	317
DP Air Trottle	[mbar]	15.05	24.82	7.95	7.43	14.29
Press Airreceiver	[barg]	1.13	1.74	0.59	0.55	1.09
Press a Compr	[barg]	1.19	1.81	0.64	0.6	1.14
Press Exhaustreceiver	[barg]	0.89	1.36	0.52	0.5	0.87
Air Cons	[kg/s]	0.89	1.14	0.65	0.63	0.87

Air Cons_s	[kg/kWh]	8.56	8.14	9.28	8.98	8.3
Exhaust Flow_s	[kg/kWh]	8.76	8.34	9.48	9.26	8.58
Exhaust Flow	[kg/s]	0.91	1.17	0.67	0.65	0.9
Humidity	[%]	23	22.9	21.5	20.5	23.5
Ambient Press	[mbar]	1007.4	1007.3	1007.1	1006.7	1006.6
Ambient Temp	[C]	25.8	26.8	27	27.8	28.5
Emission						
O2	[%]	14.48	14.13	14.48	14.26	14.09
CO	[ppm]	39.5	46.5	38	38	28
CO2	[%]	4.71	4.96	4.71	4.88	5.01
HC	[ppm]	8	12	11	14	16
Nox	[ppm]	982	976	975	822	756
CO 5% O2	[mg/nm3]	121.625	135.825	117.5	113.4	80.8
HC 5% O2	[mg/nm3]	37	51.3	52.4	63.2	68.3
Nox 5% O2	[mg/nm3]	4963.6	4682.6	4922.8	4015.7	3601.7
AVL FSN	[-]	0.169	0.28	0.1	0.02	0.02
O2_s	[g/kWh]	1330.2	1232.2	1439.5	1383.2	1261.8
CO_s	[g/kWh]	0.3175	0.355	0.33	0.32	0.22
CO2_s	[g/kWh]	594.9	595	644.4	651	616.5
HC_s	[g/kWh]	0.1	0.14	0.16	0.19	0.2
Nox_s	[g/kWh]	13	12.2	13.9	11.5	9.7
Nox_s_corr	[g/kWh]	12.8	12.1	13.7	11.3	9.7
Nox Correction	[-]	0.98	0.99	0.99	0.99	1
Dry2WetCorr	[-]	0.95	0.95	0.95	0.95	0.94
Lambda_tot	[-]	3.06	2.91	3.06	2.96	2.89
Lambda_Mflow	[-]	2.97	2.82	3.12	2.15	2
Cooling water/Lub oil						
Water temp b Engine	[C]	80.8	81.4	80.2	80.6	80.7
Water temp a Engine	[C]	82.4	83.5	81.3	81.8	82.4
Sea Water Temp b Engine	[C]	27.9	27.8	27.2	27.6	27.9
Sea Water Temp a Engine	[C]	34.8	37.8	31.8	32.1	35
Lub Oil Temp b Engine	[C]	56.2	56.4	55.3	55.8	56.3
Lub Oil Temp a Engine	[C]	62.6	63.9	61.2	61.7	63

AppendixG: **CD**

The CD attached to this report contains all of the simulation data from GT-Power, as well as the excel sheet containing all of the results from particualare matter measurements and a copy of this report.

The folder GT-Power simulation folder contains 3 folders named 50, 75 and 100. These are the files for respectively 50%, 75% and 100% load. In the GT-Power models the case 1 is the first run without any water added, case 2 is 10% water added, case 3 is 20% water added, case 4 is 30% water added and case 5 is 40% water added.