Introduction of water to reduce NOx emissions

Martin Axelsson

KTH Industrial Engineering and Management

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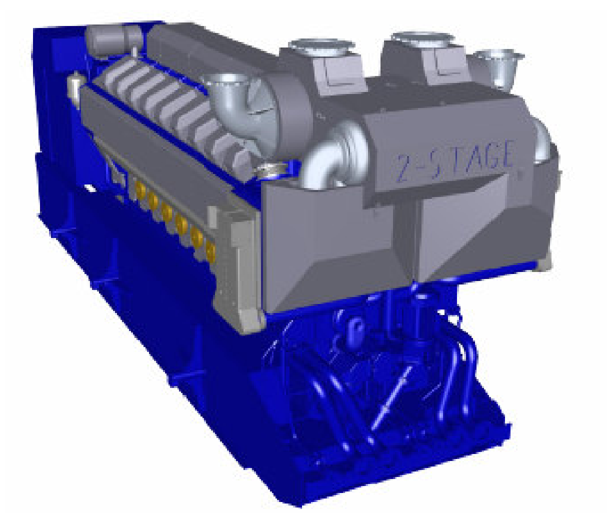

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	Introduction of water to reduce NOx emissions
	Martin Axelsson

Approved	Examiner	Supervisor
2008-11-05	Hans-Erik Ångström	Gustav Ericsson
	Commissioner Wärtsilä Finland OY	Contact person Göran Hellén

Abstract

The background for this master thesis is the need for lower emissions, especially NOx emissions. Different NOx reduction methods are described and evaluated in this study. The focus is on the "wet NOx reduction technologies" and especially the ones where the inlet air is humidified. The evaluation consists of a literature study, test investigation and simulations.

The NOx reduction effect with the introduction of water (or vapour) is achieved by reducing the local maximum combustion temperatures in the combustion chamber and also by reducing the concentration of oxygen by the addition of inert media with high specific heat (vapour). NOx formation is dependent of mainly the combustion temperature but also the availability of oxygen and by reducing the temperature and/or the oxygen concentration the NOx emissions will therefore be reduced. The reason for the reduced temperature is the increased heat capacity and the increased mass from the added water.

When water is introduced to the inlet air it is important to have sufficient evaporation to get high absolute humidity and to avoid corrosion. The evaporation is dependent on mainly the nozzle design, temperature and pressure. It is challenging to get good evaporation of the injected water and to reach high absolute humidity on a highly boosted engine like the investigated 2-staged turbo charged Wärtsilä 20V32C engine. The high pressure and the relatively low temperature due to the low pressure ratio over the second compressor are limiting the maximum possible humidity and makes the evaporation slow. According to results obtained from simulations and calculations it can be possible to reach the desired humidity but there will be very high demands on the water injection system and the design of it. The receiver pressure increases with increased water to fuel ratio which can make turbo charger re-matching necessary. This

and other performance parameters affected by the humidification was simulated in GT-power in preparation for an engine test in 2009.

A practical investigation of the Humid Air Motor (HAM) system and Wärtsilä's inlet air humidification system was carried out. The investigation of the HAM system on the Mariella ship showed that NOx emissions are reduced close to 70% in the IMO E3 cycle. The "mysteriously good" results reported from the HAM system are explained and the explanations can mainly be found in the temperature and pressure used. From the investigation of Wärtsilä's inlet air humidification system the limitation in performance can mainly be found in the evaporation and the charge air temperature.

In a comparison between all the wet-technologies evaluated in the study, the Emulsion technology and has the best overall results. It is quite equal between the other wet-technologies but "humidification with steam" is the wet-technology with the worst over all properties.

Interesting findings from the literature about new or different ways of using the wet-technologies are presented like instantaneous mixing of water and fuel or late direct water injection. Suggestions for further work, improvements and interesting things to test are also presented in this thesis work.

KTH Industriell teknik och management	**Examensarbete MMK 2009:25 MFM118** **Introduktion av vatten för reduktion av NOx emissioner** Martin Axelsson	
Godkänt 2008-11-05	Examinator Hans-Erik Ångström	Handledare Gustav Ericsson
	Uppdragsgivare Wärtsilä Finland OY	Kontaktperson Göran Hellén

Sammanfattning

Bakgrunden till detta examens arbete är behovet av att få lägre emissioner och då speciellt NOx emissioner. I detta arbete är olika NOx reduktions metoder beskrivna och utvärderade. Tyngdpunkten har lagts på de "våta NOx reduktions teknologierna" och då speciellt teknologier för att befukta insugsluften. Detta arbete består av en litteratur studie, genomgång av test resultat från motor tester med dessa teknologier samt simuleringar.

NOx reduktionen genom at introducera vatten eller vatten ånga till förbränningen åstadkoms genom att sänka den maximala temperaturen i förbränningsrummet och genom att tillsätta inert media med hög värmekapacitet vilket också reducerar syrekoncentrationen. NOx formationen är främst beroende av förbränningstemperaturen men också syretillgången och genom att reducera dessa kan NOx emissionerna minskas. Orsaken till den minskade temperaturen är den ökade massan och den ökade värmekapaciteten från det tillsatta vattnet eller ångan.

När vatten introduceras till insugsluften är det viktigt att ha bra och effektiv förångning för att nå en hög absolut luftfuktighet och undvika korrosion. Vatten förångningen är beroende av vatteninsprutnings munstyckenas utformning samt den rådande temperaturen och trycket. Det är utmanande att uppnå bra förångning samt hög absolut luftfuktighet på en motor med mycket högt laddtryck som tillexempel på den undersökta 2-stegs överladdade Wärtsilä 20V32C motorn. Det höga laddtrycket och den relativt låga temperaturen på grund av det låga tryckförhållandet över den andra kompressorn begränsar den maximala absoluta luftfuktigheten samt gör förångningen av vattnet långsam. Utifrån simulerings resultat och luftfuktighetsberäkningar bör det vara möjligt att uppnå den önskade luftfuktigheten men det ställer stora krav på vatten insprutnings systemet och designen av det. Laddtrycket ökar med ökad mängd vatten

vilket kan göra det nödvändigt att ändra på turbo laddar specifikationen. Det här och andra prestanda parametrar som påverkas av befuktningen av luften simulerades i GT-Power i förberedelse för ett motor test som kommer att ske 2009.

En undersökning av Humid Air Motor systemet (HAM) och Wärtsiläs luftbefuktnings system har också utförts. Undersökningen av HAM systemet på fartyget Mariella visade att en NOx reduktion på närmare 70 % i IMO E3 cykeln kan uppnås. De ”osannolikt bra” resultaten rapporterade från HAM systemet är förklarade och förklaringen ligger huvudsakligen i de temperaturer och tyck som används. Från undersökningen av Wärtsiläs luftbefuktnings system kan begränsningen i NOx reduktion huvudsakligen förklaras av förångningen samt laddtrycks temperaturen som används.

I en jämförelse mellan alla ”våta NOx reduktions teknologier” utvärderade i studien uppvisar Emulsions teknologin de bästa helhets egenskaperna. Bland de övriga är det ganska jämnt men ”ång-befuktning” har de sammantaget sämsta egenskaperna.

Intressanta nya ”våta NOx reduktions teknologier” och nya sätt att tillämpa dessa teknologier påträffades under litteraturstudien som tillexempel ”omedelbar blandning av vatten och bränsle” och sen direkt insprutning av vatten. Förslag på fortsatt arbete, förbättringar och intressanta förslag på nya tester är också presenterade i detta arbete.

Nomenclature

ABP	Air ByPass
BSFC	Brake Specific Fuel Consumption
CAC	Charge Air Cooler
CASS	Combustion Air Saturation System
CFD	Computational Fluid Dynamics
CO	Carbon monoxide
CO_2	Carbon Dioxides
Cp	Specific heat capacity
cSt	CentiStokes (kinematic viscosity)
DWI	Direct Water Injection
EGB	Exhaust Gas Boiler
EGR	Exhaust Gas Recirculation
EGC	Exhaust Gas Cleaner
g/kg	g H_2O/kg dry air
HAM	Humid Air Motor
HC	Hydrocarbons
HFO	Heavy Fuel Oil
IMO	International Maritime Organisation
MCR	Maximum Continuous Rating
MDO	Marine Diesel Oil
MEPC	Marine Environment Protection Committee
NOx	Nitric Oxides: NO, NO2, NO3
PLC	Programmable Logic Controller
PM	Particulate matter (particulate emissions)
RH	Relative Humidity
SCR	Selective Catalytic Reduction
SDWI	Stratified Fuel Water Injection
SFOC	Specific Fuel Oil Consumption
SLOC	Specific Lube Oil Consumption
SOI	Start Of Injection
SOx	Sulphur oxides
TBN	Total Base Number
TBO	Time Between Overhaul
TC	Turbo Charger
THC	Total Hydro Carbon
WPPP	Wasa Pilot Power Plant

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1 Introduction

The background for this study is the development of new Nitric Oxides (NOx) emission regulations for mainly seagoing ships. To meet the proposed NOx regulations requires development of today's new heavy marine diesel engines and development of reduction system to retrofit on old engines that are already in use. The focus is more on NOx than on other emissions but of course some attention must be given to them as well. This study will describe and evaluate different NOx reduction methods. The evaluation consists of a literature study, test investigation and simulations in GT-power. Evaluation and planning of a CFD (Computational Fluid Dynamics) simulation made by the company involved in this study is also carried out with inputs from the GT-power simulation. There are many different ways to reduce NOx but this study will focus on reducing NOx by introducing water to meet the regulations.

1.1 Development of pollutant emission regulations

The development of pollutant emissions regulations for seagoing ships is mainly made by International Maritime Organization, IMO, which was established by the United Nations to promote cooperation among governments and the shipping industry to improve maritime safety and to prevent marine pollution.
In this section the pollutant emissions regulations for seagoing ships will be described and what the likely development will be. Methods to reduce the pollutant emission will also be briefly described. There are regulations for NOx and Sulphur emissions but for particles the only regulation (in some places) is that they should not be visible. In some regions, the proposed reduction will not only include new engines but also engines that are already in use. For power plant engines World Bank has made globally recommendations for emissions limits but there are often local emissions regulations (different in different countries) which can be very strict in some cases.

1.2 Particulate matter (PM) smoke and sulphur

Today only local smoke restrictions are in force in some areas, like in Alaska, Florida, California, Hawaii and in some ports like port of Rotterdam and Hamburg. There will be more restrictions concerning PM but due to the fact that measurement results differ a lot for different methods the results are not comparable and the establishment of correlations between results is often impossible. This is maybe the reason why it has taken some time for IMO to do regulations for PM. There are two different proposals for the future regulations. One approach is to introduce PM measurement system with smoke number according to some standards. A second approach would not involve defining specific emission limits for PM, since PM emissions are reduced as a function of reducing sulphur emissions. Regulations could be based on the sulphur emission regulation instead which is easier to measure. A sulphur restriction is the likely approach according to the latest IMO/MEPC meetings but in the future it is also likely that the further PM restrictions will come up. The approach to restrict the sulphur instead of PM is possible due to the fact that PM from heavy fuel oil operating engines heavily depend on the sulphur and ash content. Fuels with high sulphur content will generate sulphate aerosols which can contribute a significant fraction of the total PM contribution from shipping, up to 80%. It is more effective to reduce the sulphur and ash content in the fuel (since this is the major part) than to reduce other parts caused by combustion like, partly burned fuel and lube oil, deposits peeling from the combustion chamber etc. A typical distribution of particulate for a heavy marine engine running on Heavy Fuel Oil, HFO can be seen in Figure 1.1. But of course the PM emission from the diesel engine will depend on how the engine itself performs and diesel particulate matter is a complex mixture of solid and carbonaceous material, unburned hydrocarbons and inorganic compounds. The amount of absorbed and condensed matter strongly depends on the cooling conditions like temperature, cooling rate etc. The development of the combustion process with use of common rail and different piston top shapes have led to an improvement of the PM emissions, especially at low loads.

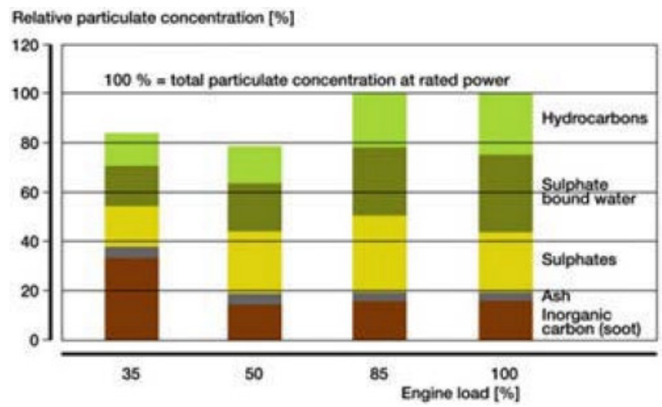


Figure 1.1 Relative particulate concentrations (mass) versus engine load according to MAN [1]

Sulphur Restrictions

Sulphur oxides (SOx) regulations in force and future limits in some areas are shown in table 1 and in Figure 1.2. One can see that the regulations will be much stricter in the future and the regulations are not uniform and differ from area to area. To meet the regulations there are two ways to go, either use fuel oil with low sulphur content (there will be no restrictions in the use of heavy fuel oil) or apply an exhaust gas cleaning (EGC) system to reduce the total emission of SOx since abatement technologies including scrubbers are allowed as alternatives. However, the consequence of the 0.1% and 0.5% sulphur caps will be distillate fuel in SECA areas from 2015 and most probably globally from 2020. IMO will do a review in 2018 for the availability of required fuel quality in 2020 and if necessary do a postponement to 2025. A likely consequence of the new regulations is that we will see more of fuel quality switching on board the ships when entering the harder restricted areas. Operation on "automotive quality" of distillate fuel (Ultra Low Sulphur Fuel) is not always straight forward due to low fuel viscosity challenges since the fuel system has to work with HFO (which has higher viscosity) as well. In this case the fuel injection system needs to be design with more narrow tolerances otherwise leaking inside the system can cause for example lower fuel pressure. Another problem can be the wear of both the valve and the valve seats when very low sulphur fuels are used. This problem can be solved by changing the materials (or alloys). But by changing the materials the life time of the valves will be shorter if HFO will be used as well.

TABEL 1.1: Sulphur regulations in force and future limits in some areas. The IMO limit on 1,5 % is comparable to SOx emissions of 6 g/kWh (as SO_2 mass) for a typical heavy marine engine.

Sulphur regulation:	IMO in SOx Emission Control Area, SECA	EU (passenger ships)	California	IMO Global
Where:	in SECA areas i.e. Baltic Sea, North Sea and English Channel	Ports and inland vessels	Within 24 nautical miles of the California coastline, only diesel electric propulsion & auxiliary engines	Everywhere except in areas were other stricter limits are used
Limit in force:	1.5% (15000 ppm)	1.5% (15000 ppm) in force	0.5% (5000 ppm) in force	4.5% (45000 ppm)
Future limit:	1.0% (10000 ppm) from 2010	0.1% (1000 ppm) from 2010	0.1% (1000 ppm) from 2012	3.5% (35000 ppm) from 2012

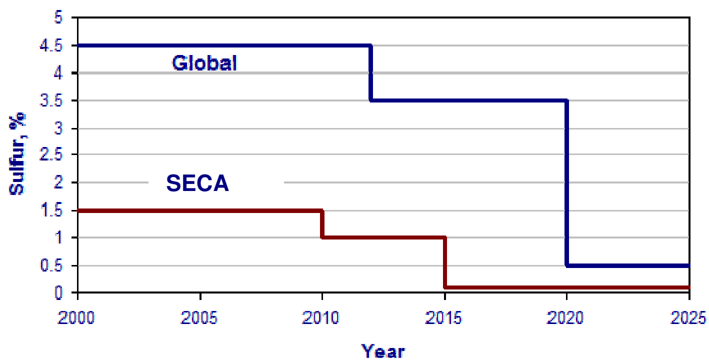


Figure 1.2: sulphur legislation from today until year 2025 globally and for the SECA areas [2], note though that this can still be changed

Since HFO is still allowed to use it can be cheaper (at least globally until 2020) to use HFO with high sulphur content together with an exhaust gas cleaner, EGC. There are two kinds of EGC, sea water and caustic soda scrubber. The sea water scrubber reduces SOx by dissolving it and neutralizing it in sea water. Sea water scrubber exploits the natural alkalinity of water to neutralize the dissolved sulphur dioxide from the exhaust gas. Caustic soda scrubber reduces SOx in a closed loop system with fresh water, to which Sodium Hydroxide (NaOH) is added for the neutralization of SOx. The SOx reduction potential is 80-90% with this system. The main drawback for sea water scrubbing is additional operating costs for the required parasitic power (pumping) for the system which is 2% of the engines maximum continuous rating, MCR. The soda scrubber requires less power (1%) but instead needs NaOH and fresh water.
All sulphur entering the engine combustion chamber will oxidize and form SOx which is emitted to the atmosphere with the exhaust gases, as the sulphur absorbed by the alkaline lubricant is negligible in this respect. Because of that the SOx emissions from the engine will practically be directly proportional to the fuel sulphur content and fuel consumption.

1.3 NOx Regulations and NOx formation

1.3.1 NOx regulation

The NOx reduction from current IMO-level, Tier I to Tier II will be introduced in 2011 and the NOx reduction from current level will be 20-25%. The next step after that, Tier III will be introduced in 2015/2016. The proposed NOx reduction is about 80% from current level, see Figure 1.2. IMO Tier III will be applied for all ships but only when they operate in certain areas and in other areas will Tier II be the level to follow. The Tier III proposed areas are the SECA areas, Tokyo Bay, USA and Canada coastal areas. The later ones are not decided yet. Observe that these regulations “Regulations for the Prevention of Air Pollution from Ships” are still under negotiations and subject for further discussions within the IMO organisation. Final adoption will be taken by IMO/MEPC 58 in October 2008. Today the IMO Tier regulations do not include HC and CO emissions since they are generally very low for heavy marine diesel engines.

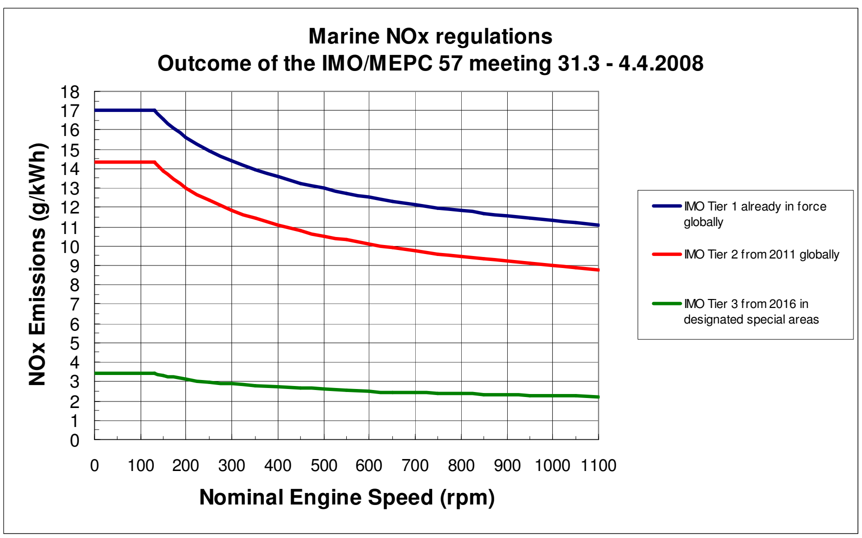


Figure 1.2: IMO regulations levels with specific NOx emissions verses rated engine speed for all marine engines globally [3]

The emission regulations are valid for the following engines:

- Each marine engine with more than 130 kW power output
- The vessel constructed (keel laid) 1 January 2000 or later
- Or a major conversion of the engine is done

The pre-2000 engines will also be affected and the proposal for these engines is that they have to comply with current Tier 1 NOx level under the following circumstances (the final outcome of this is still very open) [3]:

- Built between year 1990 and 2000
- Cylinder displacement > 90 litres/cylinder and power output > 5000kW
- Application only to those engines where an update kit is commercially available (there are proposal for the requirements of these kits concerning price, affect on fuel consumption and power output etc)

The scenario for the power plant engines concerning the NOx emissions are pretty much the same but the reduction steps are smaller in some cases, see Figure 1.3. And as mentioned earlier, the World Bank has made recommendations but for example in USA the legislated levels are really low and in that case a gas engine can be an interesting alternative to use instead of diesel engines.

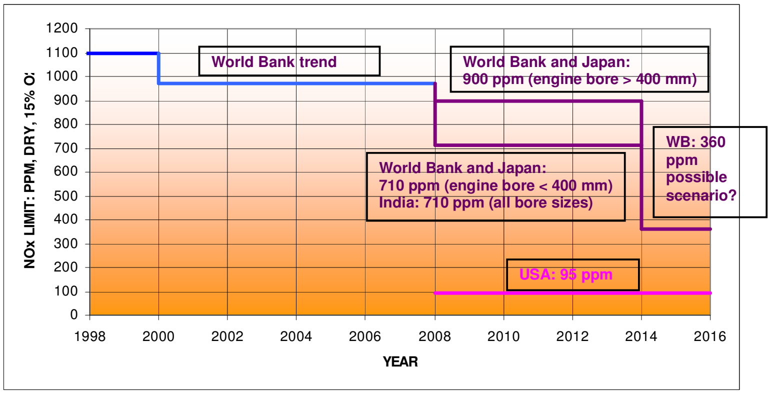


Figure 1.3: the NOx emission recommendations and legislation for power plant engines

1.3.2 NOx formation

The NOx formation process is extremely complex including hundreds of different chemical reactions and also depends on the stochastic differences in air movement, air-fuel mixture and temperature etc. This makes it hard to predict exact how the NOx formation will be and this is also subject for a lot of research. However, the main principles are thought to be relatively known. NOx emissions are relatively high from diesel engines due to high local combustion temperatures since the NOx formation is thermal and has a strong exponential temperature influence. The formation rate is primarily a function of temperature and the residence time of nitrogen at that temperature. At high temperatures, molecular nitrogen (N_2) and oxygen (O_2) in the combustion air disassociate into their atomic states and participate in the reactions described by well known extended Zeldovich mechanism. Note that the reactions take place in both directions i.e. formation and deformation.

The Zeldovich mechanism:

- $N_2 + O \leftrightarrow NO + N$
- $N + O_2 \leftrightarrow NO + O$
- $N + OH \leftrightarrow NO + H$

There are three primary sources of NOx in combustion processes namely thermal NOx, fuel NOx and prompt NOx. Thermal NOx is explained above due to the high temperature but the fuel NOx formation is depending on the nitrogen content of the fuel (which is quite low in HFO but varies between 0 and 1%). During combustion, the nitrogen bound in the fuel is released as a free radical and can ultimately form free N_2, or NO. The third source, called prompt NOx is not fully explained and can not be explained by the Zeldovich mechanism either but on the other hand the contribution of prompt NOx is normally considered relatively negligible. This mechanism takes place at the fuel rich conditions at relatively low temperatures and has short residence time compared to the thermal NO-mechanism. The prompt NOx is attributed to the reaction of atmospheric nitrogen, N_2, with radicals such as C, CH, and CH_2 fragments derived from fuel. The reactions discussed are of the following types:

- $N_2 + CH \rightarrow HCN + N$
- $N_2 + CH_2 \rightarrow HCN + NH$

The compromise between NOx, PM and fuel consumption (CO_2)

A dilemma is how to avoid increased fuel consumption which will give an equal increase of carbon dioxide, CO_2 when reducing NOx. When running engines on normal fuel (not bio fuel) it will be a contributor to the ‘greenhouse effect’. Even though the diesel engine is the most efficient power plant, and therefore presents a low CO_2 emission, reducing NOx further will increase CO_2 and also other emission components (mainly soot) again, see Figure 1.4. Therefore a number of alternative solutions must be investigated for the future, beyond Tier II.

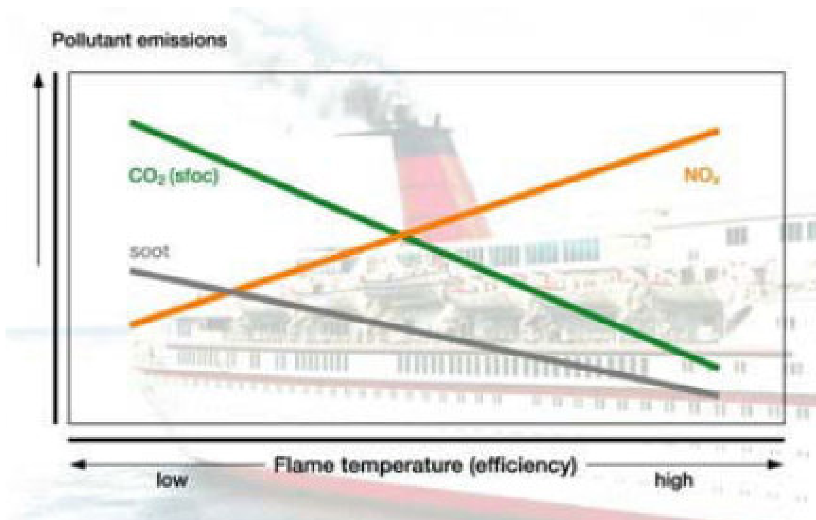


Figure 1.4: NOx-PM and fuel economy trade off, a reduction of the combustion temperature level in order to decrease NOx emission will lead to an increase in fuel consumption (i.e. CO_2) and soot [1]

1.4 NOx reduction methods overview

1.4.1 Dry Low NOx Technologies:

The dry low NOx technologies are a combination of one or several of following elements: Late Fuel Injection Timing, High Compression Ratio, Combustion Chamber optimization, Miller Cycle, Variable Valve Timing, Common Rail with improved control systems, Turbocharger improvement, 2-stage turbo charging, etc. The NOx reduction potential depends on technology or combinations of them. 2-stage turbo charging and Miller timing have showed promising NOx reduction potential for heavy marine engines, up to 40% but the reduction status today is 15-20% below IMO Tier I level. These technologies have in general low impact on fuel consumption and the fuel quality requirements are often unchanged. The cost for some parts of the system is not so high but to achieve high NOx reduction advances control system, high efficiency turbochargers with high pressure ratio is required and variable valve timing may also be required which makes the engine at least 10% more expensive. Another drawback is the difficulties to retrofit the system to existing engines.

1.4.2 SCR - Selective Catalytic Reduction

The SCR is an after treatment of the combustion gases and does not reduce the NOx formed in the combustion chamber. The SCR reduce NOx to N_2 and water with aid of a catalyst and addition of some kind of ammonia solution like a Urea solution which typically contain water with 40% urea. The NOx reduction potential is up to 80-90% and has small impact on the fuel consumption. Hydrocarbons are also reduced. The restrictions on fuel quality are not changed but it is recommended to use fuel with lower sulphur content than 1.5%. The drawbacks with this system are that it is expensive to install and the additional operating costs is highly dependent on reagent price which can be expensive. Today this makes about 10% additional operating costs. Other drawbacks are that the system takes a lot of space which makes it hard to retrofit to existing engines and it is also hard to adjust the amount of the reagent accurately to transient engine operations. Logistical problem with the urea supply has to be considered when installing a SCR (storage and transportations of large quantities of Urea can be expensive or not even allowed depending on location). Durability problem with the SCR system is common in heavy marine engine application when HFO is used due to corrosion, cracks and clogging in the system etc [4]. The system also requires a certain exhaust temperature which can be a problem.

1.4.3 EGR - Exhaust Gas Recirculation

EGR is an efficient NOx reduction method used in automotive engines but cannot be used together with current marine fuel qualities. In an EGR system a part of the exhaust gases are taken away and mixed in to the engine intake air. The NOx reduction appears due to lower peak temperatures. The temperature reduction is mainly a result of increased mass. Additional NOx reduction appears due to lower oxygen concentration. The NOx reduction potential is great, about 50-70% but the main drawback is that the fuel quality restriction is strict. Distillate fuel quality with very low sulphur (maximum 0.05% or 500 ppm) and ash content can only be used. With today’s anticipated residual and emulsified marine fuel qualities this technology can not be used due to extremely fouling and corrosion issues. The MAN B&W experience with testing EGR technology on marine engines has been characterized as “disastrous” [5] despite the installation of filters and water scrubbers to clean the recirculated exhaust gas, massive fouling and corrosion was observed after only 20 hours of operation. Wärtsilä and Pielstick have tested and dismissed EGR according to similar test result. Tests where EGR have been used together with a sulphur-scrubber have though been more promising in terms of corrosion issues especially with new improved scrubber technology than test without cleaning of the exhaust gases.

1.4.4 Wet Low NOx Technologies

Wet low NOx technology reduces NOx by introduction of water into the combustion process and the NOx reduction is mainly a result of lower peak combustion temperatures. Water can be introduced into the engine in several ways: by humidification of inlet air, by adding water into the fuel (emulsion) or directly inject water into the cylinder. Depending on which wet-technology is used, the effect on combustion and thereby the emissions are slightly different. This thesis will focus on these wet-technologies and they will therefore be described in detail later. In many cases the wet-technologies is used as a substitute for EGR when fuel with high sulphur content is used. There are other technologies that may reduce NOx like non thermal plasma but they are mainly intended to reduce other emission and are also not usable for marine engine running on heavy fuel.

1.5 Purpose

The purpose is to investigate NOx reduction technologies which use water in some way to reduce NOx emissions (more focus on NOx then on other emissions) and compare them with pros and cons. Another purpose is to develop a model in GT-Power to simulate several “wet technologies” to find out the most promising ones and the needed changes to the engine specification compared to a non humidified engine. This is done in preparation for an engine test that will be carried out in 2009, but the test is not a part of the thesis work.

The focus is more on engine specification and how to get the water evaporated rather then estimating the actual NOx level. The engine test will show what the actual NOx level will be and important data will also be available as a base for further development and improvement of the models.

1.6 Task, method description and delimitations

The task is to evaluate NOx reduction methods by introducing water by focusing on performance i.e. focus on NOx, smoke, specific fuel consumption, power output, turbocharger issues, corrosion, water quality requirements and experiences achieved during tests.

To limit the scope of the project, the focus of the work will be on the wet technologies and not all NOx and PM reduction technologies. The “dry” is just briefly discussed in some occasions. A reason to focus on the wet technologies is that they in comparison with other technologies are relatively easy to retrofit to old engines already in use which is important for the commissioner. In some regions, the proposed or required reduction from the engine in use potentially represent an even grater challenge than the emission limits proposed for new engines. For example some internal modifications are not broadly and practically applicable to in-use fleets. Some of the wet technologies don't follow the NOx-PM and fuel economy trade offs associated with most emission reduction strategies. This is also an important factor for the study.

To evaluate NOx reduction by introducing water a simulation model in GT-Power will be developed for a two stage turbocharged four-stroke marine diesel engine running on heavy fuel. The company already have a model of the two stage turbocharged four-stroke engine but not with humidification. From the simulation result the engine specification will be derived as well as suggestion for interesting things to test on the real engine.

The project starts with a literature study and then the simulations will be done. Finally the evaporation issue and nozzle configuration will be investigated.

2 NOx reduction by introducing water, findings from literature

In this chapter the different wet NOx reduction technologies will be described and compared with each other. Depending on which wet-technology is used the effect on combustion and thereby the emissions are slightly different. The investigated technologies are:

- Humidification of the inlet air, water respective steam
- Emulsion
- Direct Water Injection (DWI)
- Stratified Fuel Water Injection (SFWI)
- Other interesting technologies
    - Instantaneous mixing of water and fuel
    - Late DWI

2.1 Humidification, humidification of the inlet air

2.1.1 Description of humidification of inlet air in general

In a system like this the charge air is humidified by injecting water in fine spray into the inlet air with the aim to saturate it. Normally this is done close to the compressor outlet where the temperature is high which is preferably for the water evaporation. Tests have been done with injection of water on other places like before the compressor inlet which can reduce the compression work due to the cooling effect of evaporation and improve the dispersion of the water/vapour but in these cases big durability problems of the compressor have occurred (according to Wärtsilä experience).
The humidification of the inlet air is a relatively simple solution with good potential to reduce NOx emissions and no engine modifications except for corrosion protection of the inlet which makes this technology to an attractive one. But more advanced systems exist in this category like the HAM system. The HAM system will be described later. Another benefit is that the vapour tends to give a cleaner combustion chamber and turbine which can reduce engine wear and enable longer service interval [4], [6].

The possible water injection amount is limited of temperature and pressure of the charge air. Increased temperature will allow higher absolute humidity since the saturation point will increase with the higher temperature while increased pressure will have the opposite affect and reduce the possible absolute humidity. When the saturation point is reached or if the charge air is cooled to the dew point the vapour will condensate and either way the excess water will have to be drained out. The saturation lines at different temperatures and pressures can be seen in Figure 2.1 and the evaporation and saturation issues will be described more in detail lately in chapter 6.

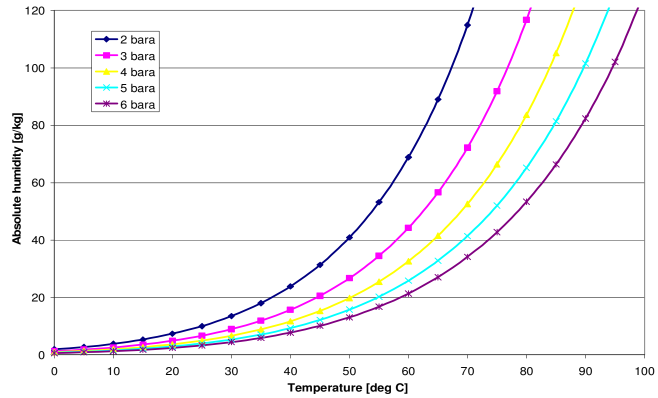


Figure 2.1: Saturation lines at different pressures versus temperature, the air can contain less vapour with increased pressure

Normally the maximum allowed charge air temperature is regulated by a Charge Air Cooler (CAC) and if the same temperature is used as in normal operation (about 40-50 ºC) without humidification, the NOx reduction potential is limited to a maximum of 20-30% depending on the charge air pressure used. With increased temperature higher reduction is possible but it will increase the thermal load and an increased receiver temperature will increase the NOx again but not as much as the increased water amount reduces it. The water consumption is very high when water is introduced to the inlet air since all the air into the combustion chamber is humidified and not only the air close to the flame as in the other wet-technologies. The temperature reduction in the cylinder is also less since the evaporation takes place in the inlet system before entering the combustion chamber where the CAC sets the temperature any way. High water consumption can of course be a big problem depending on the availability of suitable water. Humidification of the inlet air requires at least twice the amount of water to get the given NOx reduction as Emulsion or DWI would do and often more depending on evaporation efficiency and other optimisations.

NOx mechanism

The evaporated water in the charge air changes both the composition and properties of the working media. When the water evaporates the charge air pressure increases due to the increased flow which will increase the mass trapped in the engine cylinders. This affects the temperatures in the cylinders since the increased mass needs more energy to be heated up which will reduce the temperature given that the amount of fuel is constant. Another reason for the decreased temperature is the fact that the water vapour has almost twice the specific heat capacity of dry air (1.86 versus 1.0 kJ/(kg*K)) which will increase the overall specific heat of the mixture and therefore reduce the temperature globally in the combustion chamber. The increased mass and the dilution effect is the reason why exhaust gas recirculation (EGR) also reduces NOx emissions but in the case when only EGR is used (no addition of water) the specific heat capacity for the exhaust gases is about the same as for dry air.

The NOx reduction effect with the introduction of water vapour is achieved by reducing the maximum combustion temperatures in the combustion chamber and also reducing the concentration of oxygen by the addition of inert media with high specific heat. NOx

formation is dependent on the availability of oxygen and by reducing the oxygen concentration the NOx emissions will therefore be reduced [7].
A disadvantage with this technology is that the temperature is not only reduced at the flame but also at the periphery of the combustion chamber closer to the cylinder walls which can quench the flame in this region and increase other emissions like hydro carbons as well as the fuel consumption. This is though only likely to happen when the amount of vapour into the combustion is really large. Normally other emissions are relative little affected but other emissions like PM/smoke and HC tend to be slightly increased with increased water amount [1].
It is more effective in NOx reduction perspective to inject the water into the flame than to add water to the whole cylinder charge since NOx is formed on the lean side of the diffusion flame and in the hot postflame gasses, even after fuel-injection is completed. But as the diffusion flame is very thin, most of the NOx is formed in the post flame gasses [5] and by reducing the temperature in the diffusion flame only will cause the temperature in post flame gasses to be reduced and suppressing NOx formation. This is one of the reasons why water injection into the flame is more effective than to add water to the whole cylinder charge. The water or vapour has an effect on the timing and duration of the combustion as well and therefore the amount and the water injection timing is important which can't be adjusted when the charge air is humidified. The change in combustion and heat release will be described more lately in the simulation results.

Challenges and problems
The main challenge with humidification of the inlet air is in many cases the evaporation of the injected water. The evaporation is a difficult issue which is dependent on many things like droplet size, pressures, flow field distribution and the allowed receiver temperature etc. This is investigated and explained in chapter 6. Further, corrosion in the inlet system/receiver and clogging of CAC and water mist catchers is a common problem with this technology. The clogging and deposits is often a consequence of too little bled off from the recirculated drain water or too bad water quality while the corrosion problem often is result of too late or insufficient evaporation. Another reason can be too cold walls in the inlet system and that is why insulation might be necessary to avoid condensation in some cases.

Note that there are several names for this technology, for example Continues Water Injection (CWI), Fumigation and just Humidification etc so it can be a bit confusing when reading the literature concerning this subject.

2.1.2. The HAM system

The HAM system is capable to give very high absolute humidity of the charge air by injecting a huge amount of water which enables a NOx reduction of 60-80% depending on load and boost pressure according to the result presented in chapter 3. The unique with the HAM system is the ability to use sea water or even grey water for the humidification of the air. The main principle of The HAM system is illustrated in Figure 2.2.

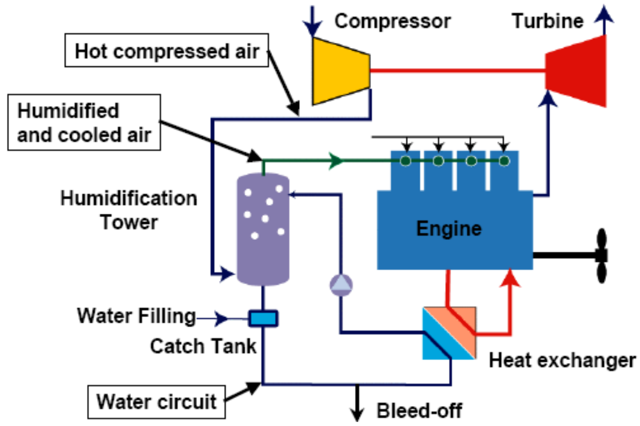


Figure 2.2: the main principle of the HAM system in a simplified sketch

First sea water is pumped into a catch tank (where also the hot drain water is gathered), then the water is heated by a heat exchanger from exhaust gases and /or engine cooling water on its way to the humidifier. In the humidifier the hot water is sprayed against the flow of hot compressed air in three steps, with a surface increasing element between each step which helps to evaporate the water, see Figure 2.3.

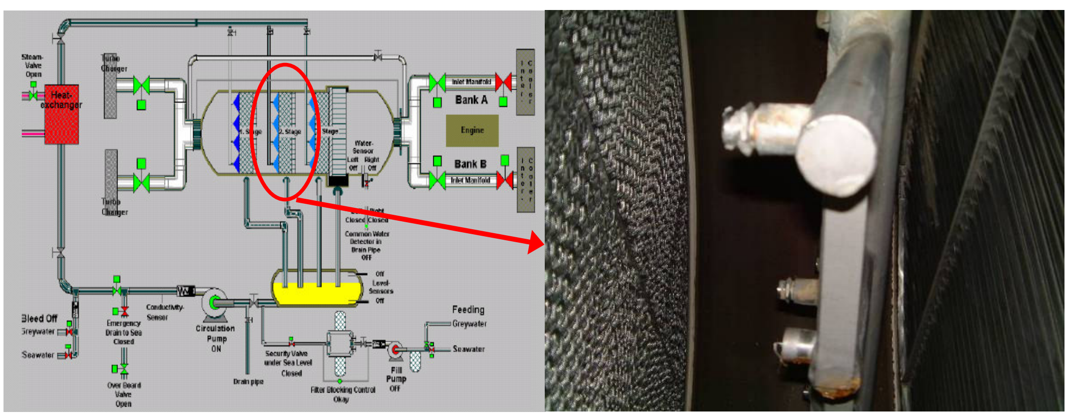


Figure 2.3: the picture show one of three humidification stages. To the right in the enlarge Figure can the droplet separator be seen, in the middle is the nozzle arrangement and to the left is the perforated steel plate or "surface increasing element" which increases the contact area between air and water a lot.

The phase conversion of the water caused by the heat of the compressed air results on one hand in saturating air and on the other hand it decreases the temperature of the mixture through energy transfer. But only 5 to 10 percent of the water flow is evaporated and the rest is drained back to the water catch tank. To avoid higher and higher salt content in the circulating water about 1 to 25 percent (depending on salt content in the sea water) of the water has to be bled off back to the sea. The water consumption is about three times the fuel consumption (waste water not included, only the amount reaching the cylinders). New cold sea water is finally pumped into the water catch tank to replace the consumed water (evaporated and bleed of water). The energy required for heating the new sea water (the water temperature into the humidifier is

around 85 ºC) and recirculating drain water is very large. External heat can be added from for example auxiliary engines to increase the amount of heat. Anti-scaling or "HAM Treatment" is added to the sea water to avoid deposits to build up in the system.
On low and medium load the water helps to heat the charge air (increases the possible amount of vapour due to increased dew point temperature) and it also helps to increase the evaporation. The huge amount of vapour increases the energy in the exhaust gases and it also increases the flow over the cylinders which make TC matching necessary.
The whole system is automatically controlled by an advanced control system with input from salt content sensors, water level and temperature sensors etc. The system has also a lot of safety features if something happens like emergency drain, disconnection of the humidifier back to normal CAC operation and other safety valves.

The humidification unit has to be sized depending on the engine size and the reason for that is that that the maximum flow velocity in the humidifier is crucial for the droplet separator located inside the humidifier. If velocity of the air is too high the droplet separator will not work properly. To keep down the flow velocities, the humidifier has to be relatively wide but the unit also needs to be long enough. The reason for the length is the time needed for evaporation. This makes the humidifier large and therefore requires a lot of space (about 1.2 m in diameter and 3.9 m long for the engines on Mariella).
Due to the fact that the system uses sea water corrosion protection is necessary and therefore the humidifier and air receiver are made of exclusive materials (like acid proof steel, reinforced plastic from the aircraft industry etc) to withstand in this corrosive environment.
Normally the salt and other minerals in the sea water will newer reach the cylinder because either the water will evaporate (distillate) and the salt stays in the unevaporated drain water or either the water will be caught by the droplet catcher. Some water droplets may even though reach cylinder but in that case they are so small that it will not affect the performance of the cylinders according to the chief engineer at Mariella and the reported longer Time Between overhaul (TBO) underlines this.
The air receiver temperature in HAM operation is increased about 20 ºC and the charge air coolers are bypassed since the humidifier cools the air instead.

The main technically changes necessary to consider when installing a HAM system

- One must ensure that there is enough "waste energy" available for heating the water for the humidification unit. The energy can be taken from both exhaust gases and engine cooling water
- Space available for the system, mainly the big humidification unit
- Corrosion protection of receivers/scavenging area
- HAM gives increased boost pressure and TC speed due to increased flow over the cylinders which make turbo matching necessary compared to a standard engine specification. Air bypass or waste-gate might also be needed.

2.1.3 Steam injection

Another way of humidify the inlet air is to inject steam. This technology solves the problem to get all the injected water to evaporate before it reaches the cylinders since all water have already been evaporated externally which also may enable the use of salt sea water depending on the steam generator. With steam injection one can have the injection before the compressor without any risks of destroying the compressor blades but the life time will probably be reduced due to erosion or corrosion. If higher temperature is used which is normally the case it will also have a negative affect on the life time. The main disadvantages with steam injection are the increased complexity and since more equipment is needed the price for a system like this is much higher than for water injection into the inlet air as well as the requirement of waste heat for the water evaporation in the steam generators. Normally this energy is taken from exhaust gas (also jacket water heat for pre heating) but depending on other waste heat consumers this energy might not be enough and will therefore limit the maximum NOx reduction. In case excess steam is available at sight (not that common) this can be a very good solution.

2.2 Emulsion

There are primarily two different types of emulsion used in large diesel engine operation, namely:

- Water-in-fuel, W/F
- Fuel-in-water, F/W

An emulsion is a mixture of two unblendable substances. One substance is in dispersed phase and the other in continuous phase. The dispersed substance (which normally can't be blended with the other) is very well mixed into the other substance in continuous Phase. This means that in the water-in-fuel (W/F) emulsion above, the water in dispersed phase is mixed into the fuel while the F/W emulsion is mixed in the opposite way. Water-in-fuel emulsion is the more common used emulsion of the two and is used to reduce NOx emissions while fuel-in-water emulsion is mainly used to be able to make use of HFO with very high viscosity but in this case it also has the benefit of reducing NOx.

Challenges and limitations with emulsion

The main limitations when using emulsion in practice when aiming for high NOx reduction is the fuel pump capacity and the fuel nozzle sizes. With a high water to fuel ratio the pump capacity will be a limiting factor at high load and to have a larger pump needs more energy to be driven which will increase the fuel consumption and the price (larger and stronger parts). Normally the engine is designed to have good performance with and without a system like this. Increasing the size of the fuel nozzles will increase the emissions (at low load) when the system is switched off and if the standard nozzles is used the fuel consumption will increase when the system is in use due to a longer opening duration to inject the larger mass (at high load) according to Wärtslä's experience. Extra heating and water production capacity is needed to be able to use emulsion properly. A disadvantage with W/F emulsion is that a undesired temperature decrease can occur and cause an ignition delay since the water comes with the fuel as the fuel injection starts but when the combustion starts it tends to be faster with a higher heat release [8] and the injection timing probably has to be adjusted to compensate for the ignition delay [9].

When the water-in-fuel emulsion is used, the viscosity increases with the water amount, see Figure 2.4. To avoid too high viscosity the temperature of the emulsion must be increased but how much depends naturally on what the HFO viscosity is from the beginning. Most heavy fuels used today are of 180 or 380 cSt viscosities (at 50 °C) and the share of 500 cSt heavy fuel is increasing. But it has to be noted that fuel viscosity – temperature relationship depends on crude oil quality and fuel refining process. It can be said that the temperatures can vary ± 5 °C from the values shown in the table. A rule of thumb in order to keep heavy fuel injection viscosity at the same level as in normal operation (20 cSt), the fuel temperature needs to be increased by 0.5 - 1 °C per each percent of added water.

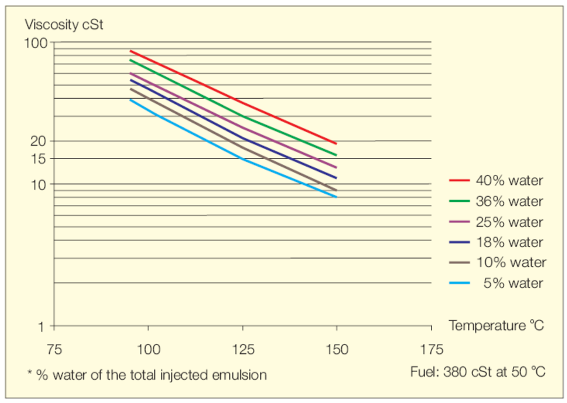


Figure 2.4: Water-in-fuel emulsion viscosity versus temperature, increased water content gives higher viscosity at a given temperature [10]

It's desired that heavy fuel injection temperature do not exceed 140 °C, since at a higher temperature the o-rings of the fuel injection pumps start to suffer according to Wärtsilä experience. On the other hand there are more expensive sealing materials available, which could be an alternative, if the 140 °C temperature can not be maintained. To operate with these temperatures the emulsion system needs to be pressurized to avoid water evaporation and boiling. Increasing the temperature even further will make the system more complicated and expensive. When we are talking about fuel-in-water emulsion this issue is not a problem, since in that case the viscosity is decreasing as a function of added water amount.

The mixing of the water and fuel or fuel and water is another challenge, they need to be properly mixed together and depending on where and how they are mixed the emulsion also needs to be more or less stable. If the emulsion is mixed in a tank long time before injection it needs to be more stable and the tank also needs to be corrosion protected. It is therefore beneficial if the mixing takes place as late and as close as possible to the injection point in the cylinder. MAN recommends a maximum water droplet size of 5µm but it might not be necessary to mix fuel and water in micrometer size droplets because other tests have shown a small impact on the emissions reduction with less advanced mixing [9], [11], [12].
Another issue is the water quality which affects the emulsion itself and the corrosion in the cylinder as well as the wear of the engine parts. A high water quality is required and

according to MAN distillated water is needed for good durability while other including Wärtsilä means that tap water is good enough.

NOx, PM and soot reduction mechanism

Water emulsion has the great benefit of reducing both NOx and also PM emissions which is not possible with humidification of the inlet air. Normally the reduced temperature due to the cooling effect from evaporation and the increased heat capacity will reduce NOx emissions but also increase PM emissions in the same way as retarded SOI would do. Low temperature and low oxygen concentration will give low NOx emissions but all these parameters will at the same time increase the soot and PM emissions. The soot formation is thought to be dependent on the forward reaction of especially acetylene (C_2H_2) as a pre-soot molecule for larger soot molecule formation [13]. But the formation of soot by acetylene is also thought to be balanced by the destruction (oxidation) of soot by the hydroxyl radical, OH [14]. It is this hydroxyl radical that is thought to be the reason for the PM reduction when emulsion is used because when water is introduced to the combustion flame (as in emulsion) it will cool the flame which normally increase the soot formation but since the test have shown a decrease in soot formation a kinetic chemical reaction with the water must occur and have an effect on the formation. It is shown that reduction of soot is caused by the increased concentration of hydrogen and oxygen since water at temperature typical of the flame zone dissociates into O, H, $H_2$and OH (and not acetylene, C_2H_2) [15].
But tests have also shown that the PM reduction tends to be smaller and smaller with increased water amount and at a certain level when the amount of water is too high, the flame temperature will be too low and cause a lot of harm to the combustion by quenching. This phenomenon occur for all the wet technologies but the influence on the PM formation is larger for emulsion since the water is well mixed with the fuel which makes the reaction in and near the flame more affected for emulsion than in the other technologies. The global reduction of the temperature in the combustion chamber for the other wet-technologies affects the PM more negatively than the positive affect of the increase in hydroxyl radical and increases with increased amount of water.

NOx reduction mechanism

The NOx reduction mechanism resulting from the introduction of water via emulsion into the flame and the combustion space is a combination of the water reducing the maximum peak temperatures in the combustion process because of its evaporation as mentioned earlier but also when the water inside the fuel droplet vaporises, the fuel droplets are divided into smaller droplets. These smaller droplets burn more effectively then integral fuel droplets thus reducing the combustion by-products [16] which thereby will reduce the NOx emissions as well as PM emissions. The introduction of water increases the mass injected to the cylinder which gives more kinetic energy to the cylinder which also benefits the homogenisation of air/fuel mixture and thereby improves the combustion and reduces the emissions further. The effectiveness of water in lowering temperature depends on injection timing and is decreased as timing is advanced [17] and the reduction capability of water emulsification varies with the engine type, but generally one per cent of water reduces NOx by one per cent according to Wärtsilä's and MAN's experience [10]. According to Transport Canada the maximum achievable reduction is up to 50 percent [5]. Normally the output engine is kept unchanged.

2.3 Direct water injection

As the name indicates, the water is injected directly into the cylinder with this technology. The water injection nozzle is separated from the fuel nozzle, as one unit with two different nozzles or two or more separate nozzles. The main advantage with this technology is that the water injection can be timed to any point during the cycle and this enables adjustment of duration as well as the injected amount to get the best performance. An example of a DWI system can be seen in Figure 2.5. Another advantage is that the water injection spray can be directed to hit the fuel injection spray which will cool the flame better (than if the water is sprayed everywhere in the cylinder) and also improve the mixing process of fuel and air in the cylinder [22]. This should be done carefully because this can also quench the flame and cause great smoke. If the water is injected before the fuel injection during the compression, the compression work can be slightly reduced (lower temperature gives lower pressure). The expansion work is also reduced with the same magnitude but the heat losses are reduced and an efficiency benefit can occur (at least in theory) but this is hard to achieve in reality. The reduced temperature can increase the ignition delay depending on water amount, droplet size and temperature. This may though enable increased compression ratio or earlier fuel injection timing depending on how much the boost pressure increases with the water injection. If instead the water is injected during the fuel injection and even after the fuel injection ends the expansion work can be reduced (but not the expansion work) with a fuel consumption penalty as a result. In this case, ignition delay is not likely to occur. In general an early injection gives a maximum NOx reduction but a late water injection can give a smoke reduction if the water is injected towards the fuel spray and therefore a compromise has to be chosen. All this can be optimised depending on the requirements and a higher absolute humidity can be reached with this technology compared to the emulsion technology.

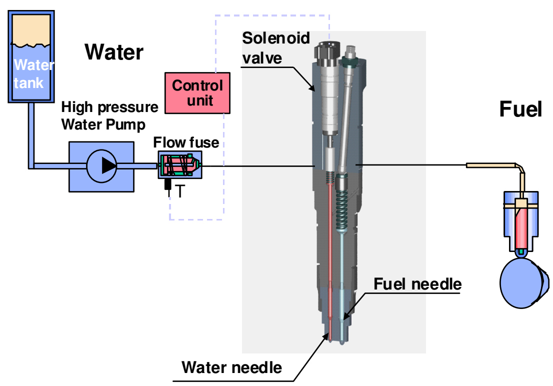


Figure 2.5: an example of a DWI system where the fuel and water nozzles are combined in one unit

Availability of water with sufficient quality can be a limitation and if the water needs to be stored in a tank this adds weight and space requirement but on the other hand the water consumption is much less than for humidification of inlet air.
The NOx reduction mechanism is the same as in the other humidification technologies but since the water evaporates inside the cylinder, the cooling effect from the evaporation will reduce the temperature more than if it would be injected to the charge

air and thereby reduce the NOx with less amount of water but the water quality requirements are higher. A NOx reduction potential of 50-60 percent is possible to achieve without suffering too much in other performance aspects (increased fuel consumption and increase in other emissions) according to test results from Wärtsilä and MAN.

Challenges
When the water is introduced directly into the cylinder during the later part of the compression the evaporation is fast due to the relatively high temperature but if the water is injected earlier there is a risk that the water hits the piston and the walls. This makes it important to get small droplets. It can be challenging to make sure that the droplets are small enough so that the water does not hit the cylinder walls. The oil film can otherwise be damaged or water can contaminate the lube oil with heavily increased engine wear as a result.
A challenge that has been reported from DWI installations is to get sufficient life time of the nozzles since the water injection nozzle is exposed inside the combustion chamber with high pressure and temperature and therefore is it hard to get good life time of them.
A DWI system is a quite advanced system which requires more or less modification of the cylinder head depending on if the water injection nozzle is integrated with the fuel nozzle or not and since the water is injected to the combustion chamber, high water pressure is needed. This makes a system like this quite expensive and a retrofit can be challenging.

2.4 Stratified Fuel Water Injection, SFWI

Stratified Fuel Water Injection is in a way a combination of DWI and emulsion because the water is injected directly to the combustion chamber, not as an emulsion but stratified together with the fuel. The water is mixed in layer with the fuel in the fuel-water nozzle, it is desirable to have some fuel (about 30 % according to [18]) first to avoid an ignition delay and then water and fuel again. One example of a system like this can be seen in Figure 2.6 but more advanced system exist where the stratification between water and fuel can be mixed and timed more independently.

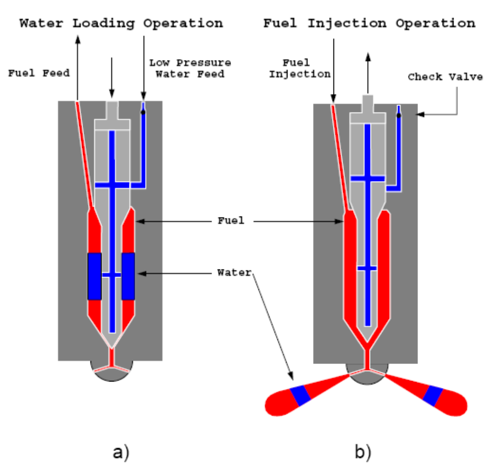


Figure 2.6 a) and b): Water loading operation a) and fuel injection operation b) for a SFWI nozzle (dual feed injection nozzle)

The water loading event (Figure 2.6a) shows the water being pushed first through a one-way valve, then through the hollow passage in the injector body and eventually into the sac region surrounding the central needle. The fuel displaced by the water is returned to the fuel injection pump. At the start of the injection event, the pressure in the fuel line is increased which closes the one-way valve so that no fuel contaminates the water line (Figure 2.6b) and the fuel-water mixture is forced out of the injector by the high pressure fuel. Note that the water supply system does not need to support as high pressures as in the fuel injection system if the fuel system is a pump injection system and not a common rail system. The pressure in the nozzle is not that high after needle closure in a pump injection system.

Advantages, limitations and challenges

The main advantage with this technology is that the water is injected to the flame where it is most effective and an increase in PM can be avoided in a similar way as with the use of emulsion (but maybe not as effective as emulsion). The cooling effect is better in the flame region compared to normal DWI due to better penetration/mixing with the fuel. Unlike emulsion it is possible to adjust the water amount as well as the timing for different operation conditions and the fuel pump capacity is not a limitation but the nozzle hole problem is still there as in emulsion. It is though difficult to know exactly how the mixing will be and it is important to have some fuel first because if too much water will end up towards the front of the injection it can cause a significant ignition delay. Another challenge with Stratified Fuel Water Injection is corrosion in the fuel injection system and wear of the injection nozzle. The increased costs due to higher complexity and the use of more exclusive materials is a challenge to overcome to make this technology competitive. A limitation compared to DWI is that a "pre water injection" can't take place before the fuel injection during the compression (in a few more advanced and expensive systems it can though be possible). The water quality requirements are the same as for Emulsion.

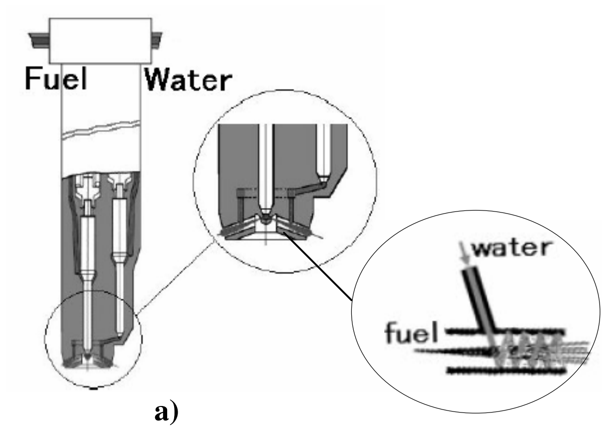
The NOx reduction mechanism is similar to the one in emulsion but the reported NOx reduction potential reported for SFWI is a bit higher, depending on load [19].

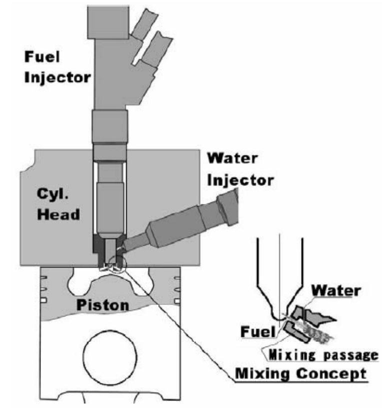
2.5 Other interesting findings from the literature

2.5.1 Instantaneous mixing of fuel and water

This is an interesting further development of the emulsion technology and combines the advantages from both emulsion and DWI. In "instantaneous mixing of fuel and water", water is injected in a mixing passage located in the periphery of the fuel spray. The fuel spray is then wrapped and penetrated by a water spray which gives the emulsion effect in the hot cylinder charge just before it starts to burn, see Figure 2.7a. Another layout of a system like this is shown in Figure 2.7b which has a separate water injection nozzle but this solution require a larger modification of the cylinder head and is therefore probably less attractive.

The mixture is not perfectly emulsified but tests have shown that a mixture like this can give almost the same NOx reduction as a proper emulsion with micrometer sized droplets would do [20]. But the length of the mixing passage is important to get good emulsion effect and test with 4 and 12 mm length showed that 12 mm gives a proper mixing in the test made by Bedford et al. The test was carried out on a single cylinder engine with bore of 125 mm and a stroke of 150 mm.





a) **b)**

Figure 2.7: an "instantaneous mixing of fuel and water" solution with integrated fuel and water nozzles in one unit a) respective a solution with separate fuel and water nozzles b)

This is a smart solution which gives all the advantages of the emulsion technology but almost none of the disadvantages. The water injection can easily be timed to get the best performance. The amount of water injected is not limited by the fuel pump capacity or the nozzle hole sizes. The emulsion unit normally used in emulsion is not necessary any more and due to this there will be cost reductions and the corrosion risk in the fuel system is also avoided. Instead the mixing passages inside the combustion chamber are very exposed to high temperatures and pressures and also wear from the water spray. Another disadvantage is the increased price for the fitting of the new water injection nozzle and mixing passage which has to be made of exclusive materials. Depending on if the new nozzle can be fitted to the cylinder head without any changes or not, some costs for modification of the cylinder head can make retrofit challenging.

2.5.2 Late Direct Water Injection (LDWI) for PM emissions reduction

Normally DWI is mainly used to reduce NOx (but also some PM reduction depending on the water amount) and in that case the water injection starts before or during the fuel injection. By changing the water injection timing from that used for NOx reduction it is shown that PM and soot reduction can be the main issue instead [21]. In an application like that the water injection starts towards the end of the fuel injection and is therefore named late DWI.

When considering normal fuel injection without DWI a possible explanation for the long after-burning and consequent soot formation at the end of the fuel injection duration is the difficulty to avoid low oxygen concentration in the region for this last injected fuel. The fuel injected at the beginning of the injection duration is first located at the tip of the fuel spray and it is then slowed down by the air drag. As a result, the later injected fuel penetrates the early injected fuel with a higher velocity and pushes it forward and to the side continuously by the following spray where a sufficient amount of air is available for the combustion. However, the fuel injected at the end remains in the center of the spray where the air supply is not sufficient because there is no following spray to push it to the front and side. But if instead water is injected at the end of the fuel injection the water droplets injected following the end of fuel injection will push all of the fuel to the front and side of the spray to regions with more oxygen and only the water/steam will remain in the center. The difference in soot formation with (b) and without (a) LDWI can be seen in Figure 2.8. The test is made by Mitsubishi Heavy Industries/Kyushu University [21]. The visual data together with the heat release data obtained from a test engine reveals that the LDWI made the main combustion more active and that the afterburning duration shorter. The test was carried out on a 2-stroke single cylinder engine (at 400 rpm) with a bore of 190 mm and a stroke of 350 mm.

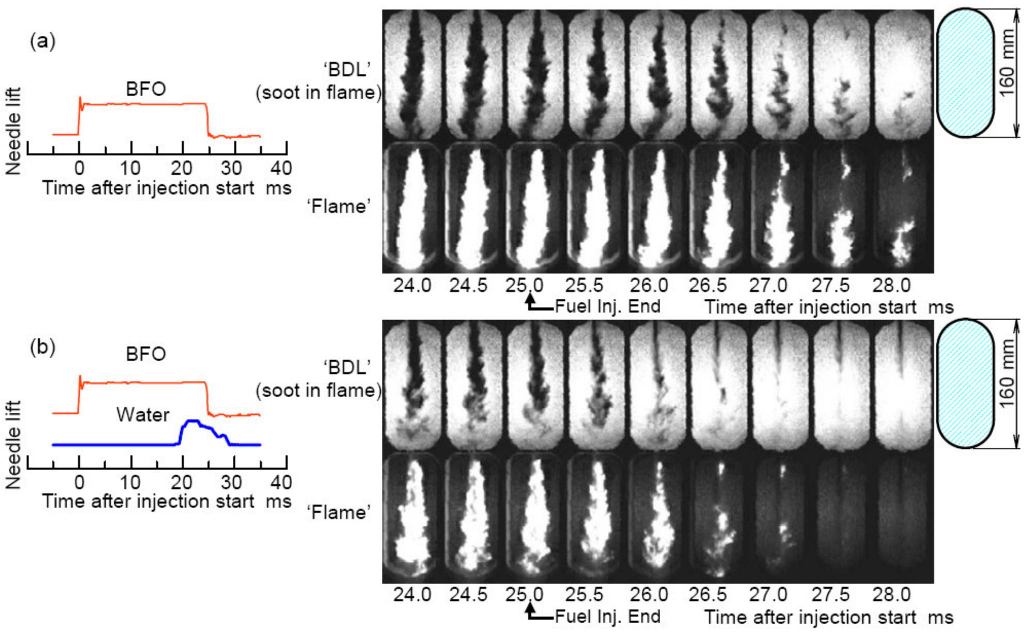


Figure 2.8: the LDWI improves the late combustion and thereby reduces the PM and soot emissions, the pictures are taken with a back diffused laser (BDL) to be able to see the smoke or liquid and a high speed camera to be able to see the flame

3 Practical investigation and comparison of test results

3.1 The Humid Air Motor (HAM) system on Mariella

The HAM system is one of the most promising NOx reduction methods for heavy marine diesel engines running on heavy fuel and especially if we are talking about "wet technologies". Very good or "mysteriously good" results have been reported both concerning emissions and fuel consumption (from limited number of engines). The HAM system and the promising results will be evaluated and explained in this chapter.
A part of the thesis work is to investigate the HAM system and compare it with Wetpac H. To get more information about the "mysteriously good" results with the system and to see it in reality a visit to M/S Mariella was arranged the 11th of March 2008 in Helsinki Harbour. Viking Line is operating Mariella and the route is Helsinki-Stockholm in the Baltic Sea (which has a low salt content). Mariella was the first ship in the world to have the HAM system installed in full scale. The first engine was converted with HAM system in 1999 and the 3 other engines in 2001. HAM is used more or less continuously, from start to stop and at all loads which means that the charge air coolers are very seldom used – they are by-passed. Because the system is in use all the time the accumulated running corresponds to about 3500 hours per year and per engine. The total number of operating hours for the HAM systems (all systems on board) is now more than 103 000 hours with very few faults or problems. In this chapter the findings about the HAM system on Mariella will be explained.

3.1.1 Technical data for the engines

Main engines are four SEMT-Pielstick 12 PC2.6V with the MCR of 5750 kW at 500 rpm on each engine, 12 cylinder V engine with bore/stroke: 400/460mm, VTR 354 turbochargers, Building year: 1985

3.1.2 Modifications done to fit the HAM system for the engines

The TC arrangement was not re-matched when the HAM system was installed, instead a TC by-pass arrangement (manual) has been installed to avoid TC surge (at 30-40% load they are very close to the surge line).This was done to save money but also because they didn't know exactly what TC setup to use. 15% of the flow is bypassed from the compressor outlet to the turbine inlet to avoid surge. Power output used is max 80% since they cannot operate on 100% due to compressor overspeed (as a result of water addition with HAM which increases to flow over the cylinders). But the Chief engineer pointed out that they never run the engines at 100% load earlier either since the strength and durability of the relatively light engine is not good enough at that load. The TC speed is kept at max 20000 rpm, corresponding to charge air pressure 1.8 bar and 80% load. Without the HAM the TC speed is about 15000 rpm and the charge air pressure is then 1.5 bar at 80% load.
All Exhaust Gas Boilers (EGB) were replaced in 2003, with the aim to increase capacity to be able to get enough energy to heat up the circulating water for the system.

The receiver was coated with a corrosion protection but later on it was replaced with one made from aluminium and with a special coating.

3.1.3 Operational experience at Mariella

The Time Between Overhaul, TBO of main engines is increased from 12000 to 15000 hours after installing HAM due to decreased engine wear. The combustion chamber is much cleaner than without HAM and the soot deposits are much softer when the HAM system is used and therefore very easy to remove (also from turbine). TC cleaning isn't necessary anymore and the exhaust gas temperature is 20 to 30°C lower due to HAM.

The Specific Lube Oil Consumption (SLOC) has decreased clearly, about 30 %, this is probably due to the fact that cleaner piston tops have reduced the liner polishing. They have also changed the lube oil to oil with a higher total base number (TBN), from TBN 30 to TBN 40 since the HAM operation had a negative effect on TBN of the lube oil

3.1.4 Problems that have occurred

The wet exhaust gases of HAM have increased the soot formation in exhaust pipes and exhaust gas boilers. The exhaust gas plume is more visible with HAM. The cracked HFO used today has poor ignition quality compared to the bunker fuel used before. This gives incomplete combustion and increased smoke at low load and these problems increases by the vapour from the HAM operation. They have improved the situation by heating the inlet sea water to the HAM system additional with an oil burner at low load. The fuel consumption for the burner is 1m^3 per day. In cold condition the water vapour in the exhaust gases condensates and appears as a visible grey "smoke" even if it is not smoke from the combustion. But note that this additional energy is very, very small compared to the main energy which is taken from the exhaust gases for the water heating.

There was some corrosion of charge air receivers after installing HAM. Then they were replaced by acid proof receivers. Now Pielstick (MAN) is testing new coated aluminium receivers on their own expense on Mariella and it looks promising. The insulation of the receiver is very important to avoid condensation of the vapour and to avoid corrosion. They have removed the insulation on the new receiver to test the coating for corrosion protection capability.

The bellows attached to the humidification unit have split and brake apart but now these are changed to shorter and steel reinforced rubber bellows and these one seams to work better.

Some sensors for the system have broken down but none have caused a stop. Once the salt content measurement stuck up and caused heavy salt deposit build up in the humidifier unit which then had to be opened and cleaned.

3.1.5 HAM Maintenance

Anti-scaling or "HAM Treatment" is added to the sea water used in the system and the consumption of the treatment is about 25 litres per three days to a cost of about 1900 Euro per year [4] (total for all four engines). Anyhow some deposits will build up in the sea water heater and in the injection nozzles which then needs cleaning after some time.

Clogging of catch-tank does not occur, but collects some mud and needs manual flushing (goes to the bilge) the same happens to the sea water filters once in a while but normally sand, mud or algae ingress is not a problem. Every third year the humidification unit is dismantled for overhaul, requiring two days of hard manual work but no experts are needed.

3.1.6 NOx emissions

There are several emission tests carried out on the Mariella ship. The test results vary a lot from test to test, especially concerning the NOx and PM emissions. The main reasons for that is probably the quite large impact from the inlet water temperature to the humidifier as well as the load dependent charge air temperature which all effect the evaporation and thereby also these emissions. The effect of the inlet water temperature is shown in table 3.1, the NOx emission reduces from 7.64 to 4.12 g/kWh when the temperature increases from 60 to 90 ˚C. This temperature is of course dependent on the available waste heat which is also dependent on load and other heat consumers on board the ship.

Table 3.1: Inlet water temperature and the amount of NOx emissions according to Marintek [22]

HAM operation 75% load	60˚C	85˚C	90˚C
NOx HAM operation [g/kWh]	7.64	4.67	4.12
NOx reduction [%]	46.9	67.6	71.4

Another uncertainty is whether the start of injection, SOI was changed or not when the HAM system was installed since the chief engineer said that it was changed at the visit but later on when the complementary questions were asked he said that it was unchanged. However very good NOx reduction is achieved at all loads though with highest reduction at low loads.
The NOx emissions at different loads are shown in table 3.2, where 100% load is defined as around 4800kW (but the rated max power is 5750 kW).

*Table 3.2: NOx emissions with normal CAC operation respective in HAM operation according to Marintek and IVL, 100% load ≈ 4800 kW (*according to manufacture)*

Load (80˚C water temp)	25%	50%	75%	100%	100% IVL	ISO E3 cycle
NOx CAC operation [g/kWh]	11.81	13.87	14.40	14.92	14*	14.30
NOx HAM operation [g/kWh]	1.51	3.50	4.67	5.51	2.60	4.60
NOx reduction [%]	87.2	74.8	67.6	63.1	81.4	67.8

The result from the performance test carried out by IVL (Swedish Environmental Research Institute Gothenburg) in table 3.2 above showed a NOx level of 2.6 g/kWh at 100% load which is a reduction of as much as 80% (if starting from 14 g/kWh)! The reason for the big difference is probably due to a difference in the reached absolute humidity, this is also illustrated in the comparison example of HAM and Wetpac in chapter 3.4.

In the Figure 3.1 the results provided by MAN/Pielstick can be found. This is typically NOx emissions from MAN:s three cylinder S.E.M.T. Pielstick PC2.6 test engine and this is the same engine type as the engines on Mariella but with slightly longer stroke and a higher output per cylinder (540kW/cyl instead of 480kW).

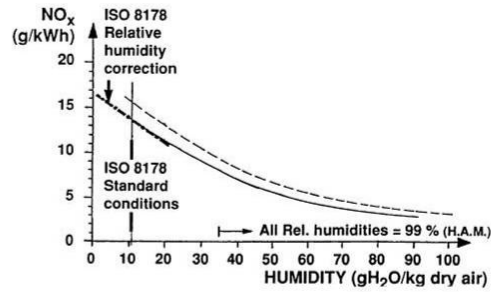


Figure 3.1: NOx emissions versus absolute humidity

3.1.7 PM and other emissions

All emissions except NOx is dramatically increased at 25% load which indicates bad combustion probably because of too low temperature see table 3.3.

Table 3.3: PM emissions with normal CAC operation respectively in HAM operation according to Marintek and IVL (Swedish Environmental Research Institute) [22]

Load (80˚C water temp)	**25%**	**50%**	**75%**	**100%**	**100% IVL**	**ISO E3 cycle**
PM CAC operation [g/kWh]	0.67	0.32	0.26	0.28	-	0.30
PM HAM operation [g/kWh]	1.36	0.53	0.41	0.35	-	0.46
PM increase [%]	*103.0*	*65.6*	*57.7*	*25.0*	-	*53.3*
THC CAC operation [g/kWh]	0.22	0.18	0.17	0.16	-	0.17
THC HAM operation [g/kWh]	0.43	0.23	0.2	0.19	-	0.21
THC increase [%]	*95.5*	*27.8*	*17.6*	*18.8*	-	*23.5*
CO CAC operation [g/kWh]	1.87	0.87	0.65	0.65	-	0.75
CO HAM operation [g/kWh]	2.85	0.85	0.56	0.49	0.43	0.71
CO increase [%]	*52.4*	*-2.3*	*-13.8*	*-24.6*	-	*-5.3*

The PM emission is increased exceptionally much at low loads with a 103% increase! And the situation on high load is not as bad but still the increase is large even there. The reason for the increased emissions is most likely due to reduced temperature and availability of oxygen in the diffusion flame and in the post flame gases which reduce particle oxidation after fuel injection ends. One should also remember that the PM emission is very much fuel dependent (the sulphur content) and can explain a part of the increased PM emission. THC (total hydro carbon) emissions are increased at all loads too but not as much as the PM emission whiles the CO emission is reduced at all loads except for 25% load. Note that the PM, THC and CO is told to be unaffected in HAM operation which seems to be far from the truthful.

3.1.8 The effect on the combustion in HAM operation

The performance Figures above for the HAM system can not be obtained if the waste heat isn't enough at certain point (especially at low loads). In that case the absolute humidity of the charge air will be reduced and the NOx reduction becomes less, but will still be significant. Additionally, the increase of THC and PM emissions will become less.

The oxygen concentration in HAM operation is at its lowest at 25% load (where the NOx reduction is highest), and increases as load increases. This is indicating that the reduced concentration of oxygen is an important parameter influencing the NOx emissions. Except for 25% load were the absolute humidity is very high, the HAM concept seems to have little effect on the timing duration and quality of the combustion. This is supported by observed changes of THC and CO emissions and the fact that according to Pielstick measurements, the ignition delay is not changed. But according to Wärtsilä a *small* change is noticed.

PM emission is the result of the competition between formation and oxidation of particles. Reducing the concentration of oxygen and lowering the temperature levels in the cylinder will reduce the rate of oxidation, leading to increased emission of PM. The effect on the combustion will be explained more in the chapter about the simulations.

3.1.9 Fuel consumption and water heating

The fuel consumption is decreased according to Viking Line (up to 4%) and Marintek* (2%) but unchanged or slightly decreased according to MAN/Pielstick (+-0.5%) [4], [22], [23]. It is difficult to measure the fuel consumption onboard the ship with good accuracy and this can be one explanation to the difference in the reported figures. Another reason in the Mariella case can be the cleaner combustion chambers and the turbines. This might have avoided or reduced the increase in fuel consumption which normally appears towards the end of the service period (when it is time for overhaul).

The reason why the fuel consumption can be reduced is the increased mass flow over the cylinders. Adding water vapour significantly increases the energy content in the exhaust gases, causing an increase of turbocharger speed which gives higher charge air pressure and therefore higher mass flow over the engine cylinders. This can give increased power, or reduced fuel consumption at a given load. But since water vapour have much higher heat capacity than air, the combustion temperature will be lower (=NOx reduction) and give a loss in efficiency which in the end makes the fuel consumption almost unchanged according to MAN/Pielstick. The influence of this negative affect on the fuel consumption depends on how high the combustion temperature is from the beginning (without water). If the temperature is reduced to a level where the combustion starts to be incomplete (increase in THC) and makes the combustion very long the influence will be grater. In that case the fuel consumption will be increased but if the combustion temperature is above this level the increased flow over the engine can give a reduction instead.

The energy for heating the sea water to about 85 ºC (water inj. temp. to the humidifier) is taken from waste heat (the exhaust gases). The energy required to evaporate the water is taken from the charge air which will be cooled by the energy transfer required for the

**Marintek is a Norwegian research institute*

phase change. The heat for heating the injection will not effect the consumption if enough waste heat is available. The heat demand is huge, the total energy requirement to heat the water is 930kw at full load in Baltic Sea condition (around 0.5%.salt). But the amount of waste heat can be a problem depending on what kind of ship it is and where it operates. With salt content of 3.5% (the average for the all seas) the required energy increases to 1280kw due to increased drain back to the sea to keep the salt content in the system below the maximum level (6%) [22].
The water pumps for the HAM system requires of course energy but since the effect is only 4kW it has therefore a marginal affect on the Specific Fuel Oil Consumption (SFOC). The heat required for HAM system is taken from waste heat and does not affect fuel consumption if there are enough of it for other heat consumers onboard. For this concept to be competitive this energy must be available as waste heat (ok at Mariella in most cases).

3.1.10 Costs

The installation cost for the HAM system is almost the same as for a SCR installation, according to Viking Line 2.000.000 Euro for retrofit of all four main engines, same for both systems [4]. For the SCR system they have calculated additional Urea cost of 280.000 Euro/year for operating the route Turku-Stockholm. The additional cost for the HAM system is only water treatment which is nothing compared to the Urea cost (1900 Euro~0.7% of Urea cost). With the HAM system the time between overhaul is increased by 12-15% while SCR doesn't increase TBO. Other maintenance costs are hard to estimate but according to the chief engineer, the HAM system is more reliable and has a longer life time. The fuel consumption is more or less unchanged for both SCR and HAM operation it will not affect the operational costs.
Today it is cheaper to pay the higher port and fairways fees than to install this kind of system according to the chief engineer but the times are changing…

3.1.11 Conclusion from the investigation of the HAM system at Mariella

The NOx emissions are reduced close to 70% in the E3 cycle when the HAM system is used while the PM emissions and THC are increased heavily on low load where the NOx reduction is highest. The fuel consumption is unchanged or reduced. Further observations are that the heat requirement for heating the injection water is huge and has to be taken into considerations when installing a system like this. The HAM system is a quite big and complex system although have very few problems occurred during the 103 000 running hours accumulated since the first system was installed in 1999.

3.2 Introduction to CASS and Wetpac H today

Wetpac H or Wetpac Humidification is Wärtsiläs system for humidification of the inlet air. The system was earlier named Combustion Air Saturation System (CASS) but there are only minor differences between the systems and the main principle is exactly the same. There are two main versions of the system one for power plant applications which uses normal CAC temperature and is used to reach the 710 ppm NOx level. The other version is a marine version which has higher NOx reduction possibilities (up to 40-

50%). This version is optimised case by case depending on desired NOx level and water availability and engine type. In this case the Charge Air Cooler (CAC) temperature can be varied to get the desired performance.
The water is injected into to the hot charge air with high pressure at the compressor outlet before the adjustable CAC. After the CAC is a Water Mist Catcher (WMC) placed to catch the un-evaporated water droplets and the un-evaporated water is then drained out and re-circulated and a Programmable Logic Controller (PLC) unit automatically controls the water injection from the input from the temperature and humidity sensors. The principle of the system can be seen in Figure 3.2.

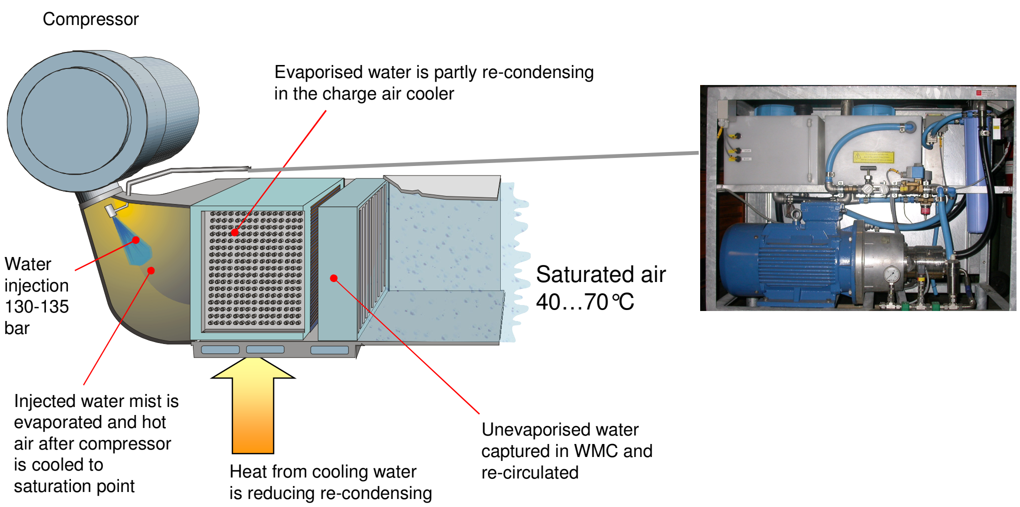


Figure 3.2: The principle of the Wetpac H system

This system is less complicated and therefore less expensive compared to for example the HAM system or Wetpac DWI. It is a flexible system with control of water flow rate and easy switch on/off possibilities. Another advantage with the Wetpac H system is that the fuel consumption will be almost unchanged.
Problem with corrosion have accord but with a new coating of the inlet system the problems are solved. Anyhow could an improvement of the evaporation reduce the corrosion risks and also enables higher NOx reduction. That's why the evaporation issue is investigated and evaluated in later chapters.

3.3 Interesting findings from Wetpac H and CASS tests

In the tests the reached water to fuel ratio is depending on the weather conditions and on the absolute humidity of the intake air before the water injection. More CASS water must be injected to the intake air at dry weather conditions than at humid weather conditions to achieve the required absolute humidity level of the intake air. In this section the water to fuel ratio is based on mass. For example if you have 200g fuel and a water/fuel ratio of 0,5 it means that the total amount of emulsion is 300g and the mass of water in the emulsion is then about 33% if the fuel is HFO.
When comparing the reached NOx reduction from different tests one should also keep in mind that the NOx reduction depends on several engine specific things like for example:

- Combustion peak temperature –engines with higher peak temperature tend to give higher NOx reduction
- engine speed, fuel injection timing and duration –gives different residence time at high temperatures
- cylinder pressure –effecting the evaporation speed and thereby the NOx reduction effect (for emulsion and DWI cases)
- difference in fuel quality between the runs

This means that the NOx reduction can vary a bit even if for example the same CASS setup is used but it is fitted to two different engines with different specification and as always the accuracy in emission measurement can vary depending on what measurement equipment was used at the given test (not the same in all tests).

3.3.1 Effect of injection water temperature and pressure

Figure 3.3 shows the effect of the CASS water heating and water injection pressure on NOx emissions. The heater was able to rise the CASS water temperature from 30°C to about 52°C depending on engine load i.e. the water flow through the heater. Both increased pressure and increased temperature of the water slightly reduces the NOx emissions but on the other hand the increase in pressure and temperature was small (18 bar respective 22 ºC).

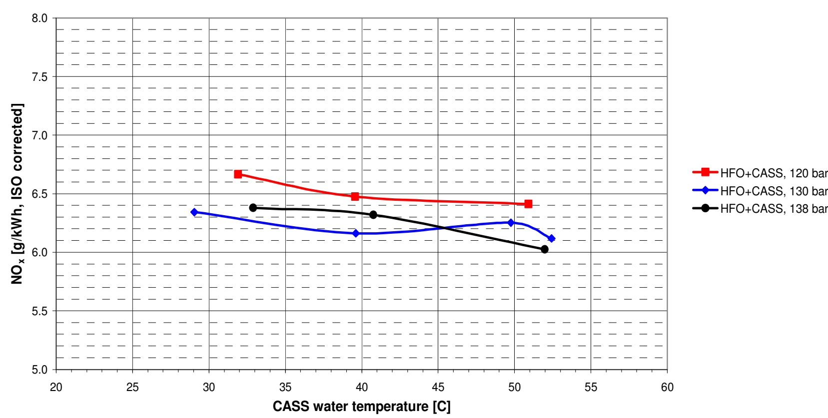


Figure 3.3: the effect of CASS water temperature and pressure on NOx emissions at 75% load, note that the receiver temperature was limited to 65 ºC

Another test with a larger injection water pressure difference was carried out in September 2004. Figure 3.4 shows what happens with the reached absolute humidity when a low respectively a high water injection pressure is used. The absolute humidity is remarkably lower with 30 bar pressure compared with 130 bars from 50-100% load. This shows the importance of small droplets due to the higher pressure with fine spray and good distribution over the flow field.

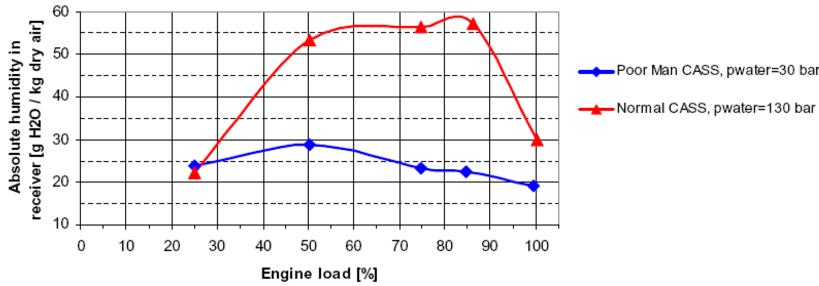


Figure 3.4: Absolute humidity versus NOx emission for different loads at 30 and 130 bar water pressure

By using drain water recirculation the CASS water temperature rises even up to 70°C. In Figure 3.5 the effect of water heating on NOx emissions with Marine Diesel Oil (MDO) at 10-110% engine loads can be seen. The temperature difference between these test series was 20 °C. The recirculation was effective at 25% and 50% engine loads where recirculation reduced the NOx emissions by 0.5 g/kWh. At other higher engine loads any major influence could not be seen. The major reason for this is likely the fact that the CAC was limiting the air temperature to 65 ºC to avoid turbo surge. You are first heating the injection water to get good evaporation but then the humidified air is cooled (and condensed) further down streams in the CAC and therefore no major effect on NOx emissions will appear.

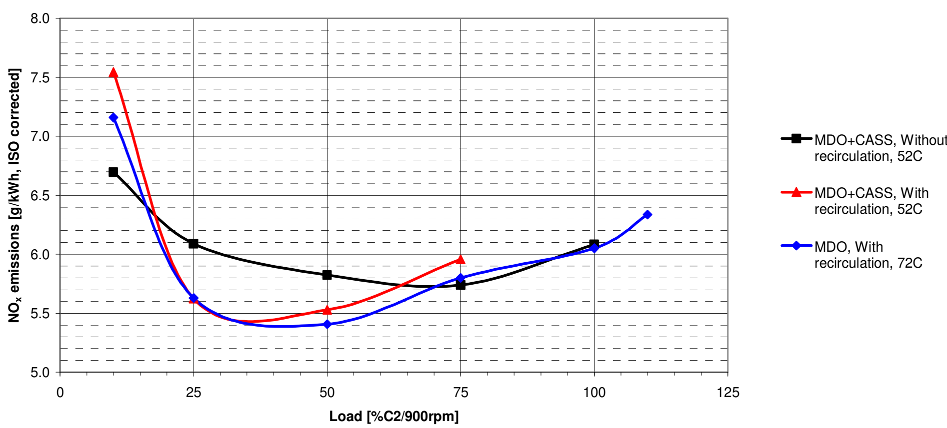


Figure 3.5: an example of the effect of CASS water recirculation on NOx emissions. 135 bar water pressure, note that the receiver temperature was limited to 65 ºC

3.3.2 Turbo issues, air temperature influence on surge and the needed bypass flow

CASS strongly increases the charge air pressure while the air flow stays almost constant this strongly moves the operating line towards surge. By using Air By-Pass (ABP), also called Anti Surge Device (ASD), the operating line can be moved away from the surge line. This is a simple and cheap application on the field and has the benefit of having no performance loss when the CASS system is switched off (standard TC match, Antisurge valve closed). This test showed that the ABP before CASS nozzles solves the surge

problem at 75% to 85% loads and 20-25% bypass flow is needed to avoid surge, see Figure 3.6. As much as 50% bypass flow was tested but it is too much.
Other conclusions are that down boosting does not slow the surge problem and a smaller compressor is not enough and the TC specification has no major influence on the NOx emissions.

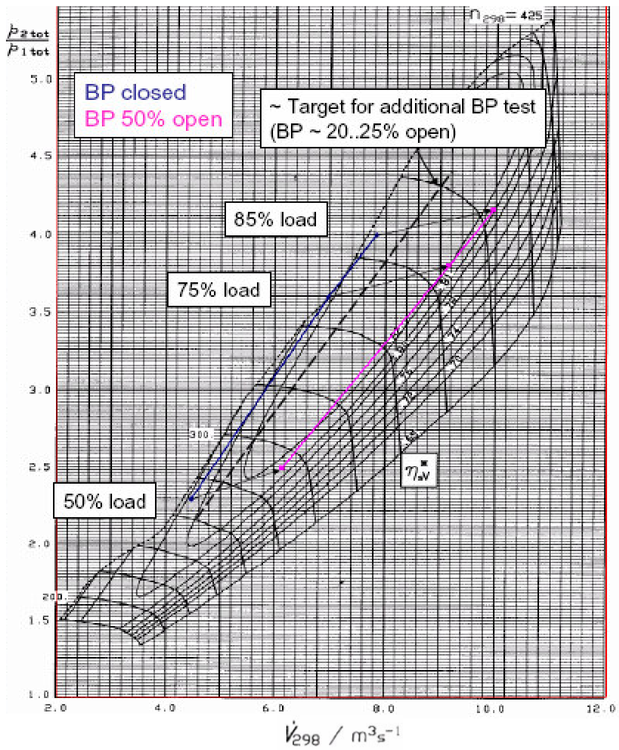


Figure 3.6: Efficiency map for the compressor with the operating lines for 0% bypass (BP) flow, 20-25% BP flow and 50% PB flow

With increased temperature needed for the evaporation and the ability to carry the large amount of water, the needed pressure ratio increases to maintain the output. It is not possible to reduce the pressure ratio without reducing the receiver temperature and accepting an increased NOx since the lower temperature will reduce the amount of vapour in the inlet air. Also note that increased temperature will give increased NOx without increasing the water amount, see Figure 3.7. The lowest NOx level was achieved when a receiver temperature of 90 ºC is used (not shown in the picture) but "only" 60 ºC is possible to use with respect to the highest pressure ratio usable for the TC in question on this engine. On lower load than 50% load the receiver temperature is too low for the evaporation and therefore the CASS system is not effective at these loads.

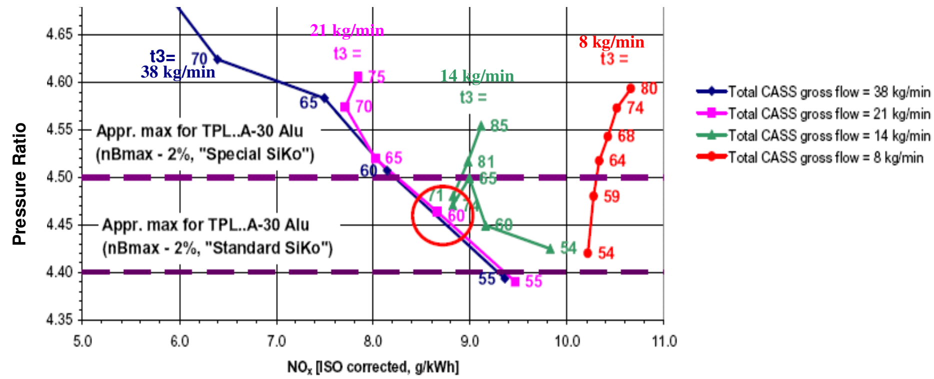


Figure 3.7: The needed pressure ratios at different temperatures (t3) and water flows (kg/min) versus NOx emission at 100% load, the red circle shows the optimum within the desired pressure ratio range

3.3.3 Injected water and the amount of drain water

The amount of drain water depends very much on what CAC temperature is used because it will set the saturation point and thereby the maximum possible absolute humidity but it will also affect the evaporation. The evaporation will be faster and faster with increased temperature. If more water is injected than what is possible to evaporate at a given condition in the inlet system the amount of drain water will increase. This makes it hard to compare the amount of drain water from test to test since it depends on what the evaporation efficiency is at the specific test and how well the injection amount is optimised for the given condition.

One example of the injected amount and the amount of drain water can be seen in Figure 3.8. When aiming to very low NOx levels, below 6 g/kWh in this investigated test, the drain water amount starts to be high. At 100% load a NOx level of 10 g/kWh which corresponds to 670 ppm (ISO corrected) can be achieved without any drain water but for the 6 g/kWh the amount of drain water is 20% of the total amount. At higher water flows than 18 kg/min the evaporation rate is not enough and drain water is inevitable.

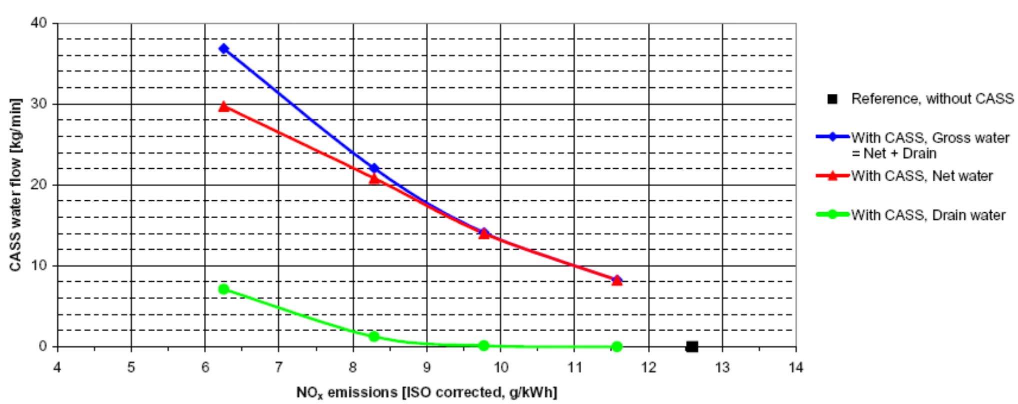


Figure 3.8: Water flow versus NOx emission at 100% load, when aiming for the lowest NOx levels the drain water amounts starts to be high

3.3.4 Emissions and SFOC

Emissions

The NOx reduction result from the test varies a lot depending on load since the charge air temperature is too low on low load to get good NOx reduction at these loads. However, IMO -50% based on IMO test cycle D2 was achieved in several tests but also in test cycle E2. For more information about the test cycles see Appendix A. When the NOx emission reduction was 50%, the water to fuel ratio was around 1.4-1.8 in these tests. The lowest NOx emissions were measured when water is injected before and after CAC. IMO -59% (Test cycle D2) and IMO -64% (Test cycle E2) was the best results achieved. The drain amount was in the range of 35-55% of injected water amount depending on engine load in these cases. Compared to previous CASS tests on same test engine, the drain water amount had grown. Note that the NOx reduction is very much dependent on the CAC temperature used because it will limit the absolute humidity into the engine. With the standard CAC temperature it is not possible to obtain the results above.

The smoke level is slightly increased with CASS and the increase tends to be larger with higher water quantity.

Specific Fuel Oil Consumption, SFOC

The reported change in fuel consumption varies with the load but also from test to test. In one test when the CASS system was used the specific fuel oil consumption increased 0.5-1.5 g/kWh at 50-110% engine loads but at 10-25% engine load (MDO was used) the SFOC was improved and in another test a slightly increased SFOC was reported at all loads.

3.3.5 Conclusions from Wetpac H and CASS tests

The NOx reduction limitation for the CASS and Wetpac system is the CAC temperature. Higher CAC temperature increases the possible absolute humidity and improves the evaporation. The CAC temperature can not be increased too much since the needed pressure ratio will increase and the increased flow will increase the pressure as well. This will cause TC surge at some point and of course the thermal load will at some point be a limitation as well. But according to the test the CAC temperature can be increased to 60-65 ºC in some cases with the use of bypass which will enables higher NOx reduction than what is promised today since the CAC temperature is just slightly increased or unchanged in many cases today. Increased water injection pressure and temperature seems also to be beneficial for improvement. There are definitely room for improvement of today's evaporation efficiency.

3.4 Comparison HAM versus Wetpac H

In this chapter some examples are given to illustrate the difference in reached humidity and also the difference between HAM and Wetpac H. A conclusion about the "mysteriously good" results will also be presented to summarise this investigation of the HAM system.

Example 1: *An example of the HAM system at 100% load taken from table 3.2 in chapter 3.1, the 5.5 g/kWh (at 3.29 bar (a) boost pressure, 76°C receiver temp) NOx level measured by Marintek (case 1) versus the 2.6 g/kWh (at 2.8 bar (a) boost pressure, 76°C receiver temp) NOx measured by IVL (case 2).*

The absolute humidity into cylinders required to reach the measured NOx levels can be estimated from Figure 3.1 in chapter 3.1 to about 50 and 90 g water/kg for case 1 respective case 2. The theoretical maximum absolute humidity at saturation (100% RH) at a pressure of 3.29 respective 2.8 bar with an air temperature of 76ºC is 86.5- respectively 104 g water/dry air (note relatively large influence of the small increase in pressure) this means that achieved absolute humidity corresponds to a relative humidity of 60 respectively 88%. This shows that the achieved absolute humidity varies at a given load and indicates that relative humidity achieved is not 99% but instead a more reasonable humidity level up to about 90% is reached.
According to Wärtsilä´s own "rule of thumb" the amount of water needed to achieve 5.5 g/kWh respectively 2.6 g/kWh starting from 14.9 respectively 14 g/kWh would be 65 respectively 85 g water/kg dry air (ISO conditions). Consequently the graph from Pielstick in Figure 3.1 in chapter 3.1 is a bit optimistic for case 1 and a bit pessimistic for case 2 compared to Wärtsilä´s experience but on the other hand is Wärtsilä's experience quite limited on those very high absolute humidity levels. In this respect the reported very low NOx level seems to be possible and quite realistic.

Example 2: *Comparison of HAM fitted on a modern heavy marine engine like the W46F versus currant Wetpac H on the same engine at high load*

On a W46F engine the receiver pressure is typically close to 5 bar (a) and the receiver temperature is around 55ºC at full load. Under these conditions the theoretical absolute humidity corresponding to around 90% RH (the level reached at Mariella) is only 19 g water/kg dry air but by increasing the receiver temperature to the same level as on Mariella then the theoretical absolute humidity corresponding to 90% RH would be 50 g water/kg dry air. This would give a NOx level of about 5.5 g/kWh according to Wärtsilä "roule of thumb" and only 4.5 g/kWh according to Figure 3.1 in chapter 3.1, if starting from 12 g/kWh (IMO limit today) as reference, which is a reduction of 55-60%.
This NOx reduction is a somewhat better than what Wetpac H performs today but on the other hand the HAM system is using much higher receiver temperatures compared to Wetpac H. The evaporation efficiency of HAM is also most likely better in the big size separate HAM vessel compared to the small injection system used in the Wetpac system.
If instead assuming the use of 2-stage turbocharging and Miller timing the charge air pressure can be as high as 8 bar (a), in that case it would be even more demanding for a system like this. About 35-40% NOx reduction with HAM could be achievable but on

the other hand the NOx starting point with 2-stage turbocharging is 40-45% lower which in total would give a NOx reduction that could be up to more than 60%.

To summarise this, the key elements in the "mysteriously good" results with the HAM system are:

- "old fashion" low boosted engine
- much higher receiver temperature than what is used in Wetpac H
- much more water (pre heated) is used to get the desired amount of vapour which also requires a lot of waste heat
- ➢ The higher temperature and lower pressure together with the use of much more water (and pre-heated) enables the higher absolute humidity compared to Wetpac H and thereby higher NOx reduction

3.5 Emulsion tests

Wärtsilä has experience from several emulsification units. Among the investigated tests the microemulsion is the most frequent tested system. The microemulsion is created in turbotransducers, where fuel and water are mixed in cavitations chambers. The chambers are surrounded by strong permanent magnets. High flow velocities and magnetic fields are causing a vortex phenomenon to produce water in fuel microemulsion. The unit does not mix the emulsion instantaneously as the emulsion is required from the engine and therefore a separate microemulsion buffer tank is needed.

3.5.1 Effect of emulsion on viscosity and fuel heating capacity

In the investigated tests it was confirmed that emulsion viscosity increased when water content in fuel increased compared to integral heavy fuel oil, as expected. This means that the fuel injection temperature has to be higher with emulsion in order to keep the injection viscosity at same level. The effect of the heater on the test engines fuel module was originally 58 kW. When water/fuel ratios 0.2 and 0.3 were tested, it was noted that the capacity of the heater was not enough to keep the viscosity at 20 cSt. The capacity of the heater was increased to 72 kW. With the increased effect it was possible to run with water/fuel ratios up to 0.3.

Stability Problems

The microemulsion is claimed to be very stable but problems were still noticed in the investigated tests. There was for example enough water separation to create rust on the inner surface of the buffer tank and foaming/boiling of the fuel when pressure was too low. Other problem reported during testing was problems with the fuel booster pump which broke down due to cavitations (after around 100h) when water/fuel ratio was increased up to 0.3.

3.5.2 Emissions and fuel oil consumption

In general the fuel oil consumption is moderate affected when emulsion is used but depending on nozzles, water to fuel ratio and load, a slightly increase can be noticed when water is added to fuel with standard injection nozzle. But with the use of a larger hole diameter nozzle a small decrease in fuel oil consumption was detected instead. The load and water to fuel dependence on SFOC can be seen in Figure 3.9.

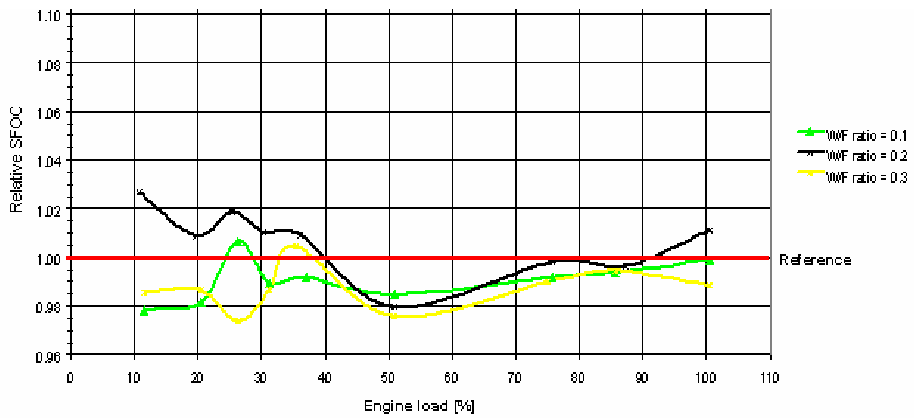


Figure 3.9: Relative SFOC level versus engine load from emulsion test, December 2002

Particulates and smoke
The PM emissions when emulsion is used is very much load dependent, at full load PM and smoke emissions decrease or remain at same level but at part load significant decrease was noticed. The FSN reduction is 0-60% depending on W/F ratio and load, see Figure 3.10.

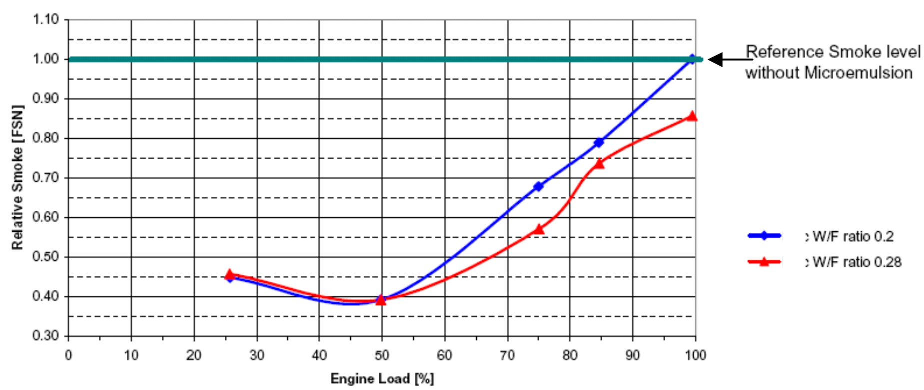


Figure 3.10: Example of filter Smoke Number versus load from an emulsion test

The same smoke reduction tendencies can be seen for both nozzles even though the original smoke level is much higher for larger hole nozzle compared to standard nozzle.

NOx emissions
The NOx emissions are also load dependent when emulsion was used but not as much as the PM emissions. In Figure 3.11 below it can be seen that the NOx emissions are reduced at all load points and the reduction increases with increased water to fuel ratio. The nozzle hole size has also a influence on the NOx emissions, the standard nozzle has slightly lower NOx emission level than the larger hole nozzle.

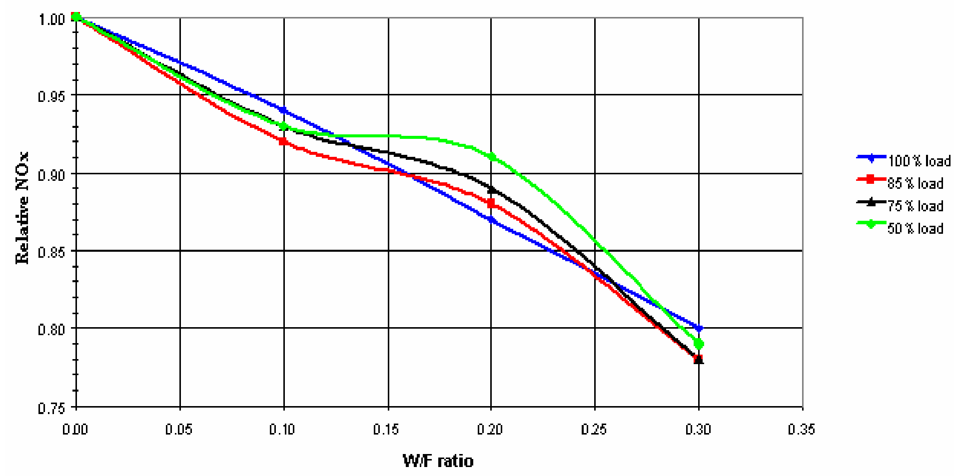


Figure 3.11: An example of the relative NOx level versus water to fuel ratio

The average NOx reduction at 100% load is 0.75% NOx reduction per added percent of water in the Figure 3.11 but the NOx emission reduction varies from test to test in the range from less than 0.6 up to about 1% per added percent of water.
Several tests also showed a positive effect on the CO emissions with the use of emulsion while the THC emissions were slightly increased.

3.5.3 Test of microemulsion in combination with CASS

When microemulsion and CASS technologies are combined, the NOx emissions can be reduced effectively at all engine loads. Figure 3.12 shows a result from a test where CASS and emulsion is combined. The NOx reduction is 50-60% in this case and the reduction is larger than when CASS is used alone at all engine loads. At high engine loads an absolute humidity of 50-55 g H_2O / kg dry air was achieved. The reference absolute humidity without CASS was less than 10 g H_2O / kg dry air.

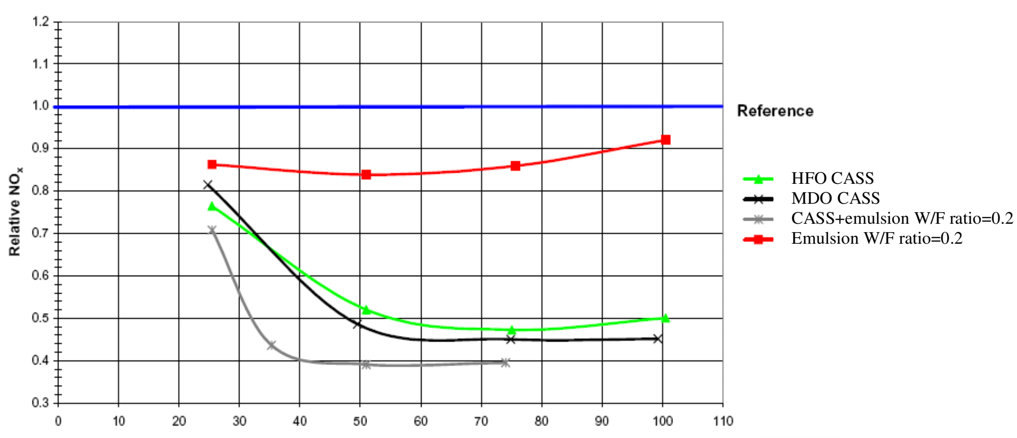


Figure 3.12: Relative NOx emissions versus engine load form a CASS and emulsion test (with standard nozzle)

Other emissions and SFOC

The smoke is at a slightly lower level with the combination of CASS and emulsion compared to the reference smoke level even though the common rail engine has a very low smoke level at all engine loads (especially at low loads). When CASS and emulsion were combined, the smoke is reduced at all loads to the same level as when emulsion is used alone. Normally CASS increases smoke especially at low loads.

The CO emissions are remarkably lower with CASS & emulsion compared to reference while the THC emissions are at slightly higher level.

The combination of CASS & emulsion had no major effect on CO_2 emissions. But based on one test series with CASS & emulsion, it is hard to guarantee the effect of CASS & emulsion on specific fuel oil consumption because the effect varies at different engine loads.

3.5.4 Conclusions from Emulsion tests

With the use of emulsion both the NOx and PM emissions can be reduced and the NOx reduction increases with increased water to fuel ratio. The PM reduction is very much load dependent and the reduction at low and medium load is very high. Depending on what nozzle hole size is used it will affect both emissions as well as the fuel oil consumption. It will also affect the performance when the emulsion system isn't in use. The fuel consumption is though only moderate affected of the emulsion.

With the combination of CASS and microemulsion it is possible to reduce the NOx emissions even further without increasing the PM emissions like CASS tend to do alone and the SFOC was almost unaffected. These results are promising but one should keep in mind that a combination like this will be a more complicated and expensive solution for the customer.

3.6 Tests with steam injection into the intake air

The steam injection tests investigated in this study are; one where steam is injected before the compressor and the other one were steam is injected both before and after the compressor. In the latter one, at engine loads up to 85%, the intake air humidity of 45 g H_2O/kg dry air was the maximum humidity that could be tested within turbocharger and receiver pressure limitations. The maximum achieved humidity at full load was 40 g H_2O/kg dry air.

The required energy to heat water to steam was obtained from exhaust gases and the HT cooling circuit. Figure 3.13 shows a part of the steam injection system at Wasa Pilot Power Plant (WPPP).

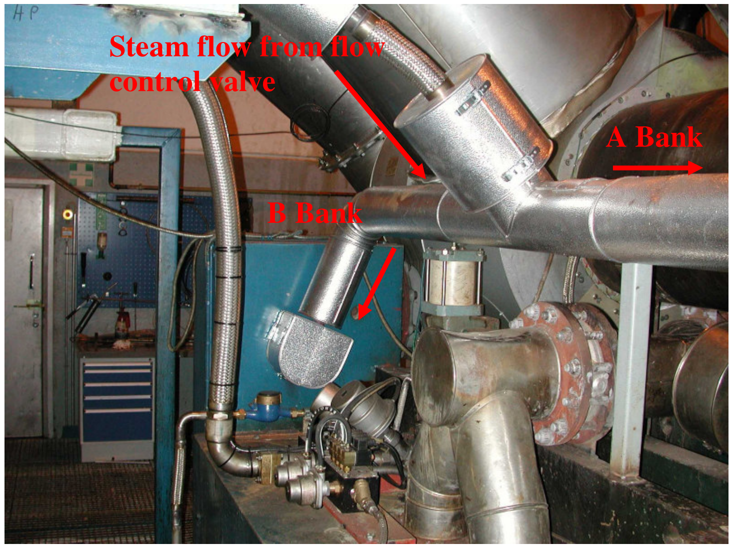


Figure 3.13: a part of the steam injection system at WPPP

The steam injection into the intake air enables the engine to run with fully saturated intake air regardless of ambient conditions. If ambient air is fully saturated no steam can be injected.

3.6.1 The effect on the emissions and fuel consumption with the steam injection

The highest NOx emission reduction was achieved when steam is injected before and after the compressor at the same time. Also steam injection after the compressor was quite effective. About 500 ppm (dry, 15% O_2) NOx reduction which corresponds to around 7 g/kWh was measured when the intake air humidity was 40-45 g H_2O/kg dry air.

Smoke (FSN, Filter Smoke Number) was at the same level at all intake air humidity's but at low engine loads, 25% and 50%, the smoke values slightly increased when the intake air humidity was increased.

The intake air humidification by injecting steam after the compressor has practically no effect on fuel oil consumption and when steam is injected both before and after the compressor the SFOC is increased slightly. Note that the specific fuel oil consumption was measured by using the flow meter i.e. the level of the SFOC is not necessarily correct.

3.6.2 Other interesting findings

When steam was injected only before the compressor, the receiver pressure remained at the same level but when steam is injected after the compressor, the receiver pressures increased when the intake air humidity increased see Figure 3.14. Turbocharger speed increased around 1.0 % at 95% load with the maximum intake air humidity.

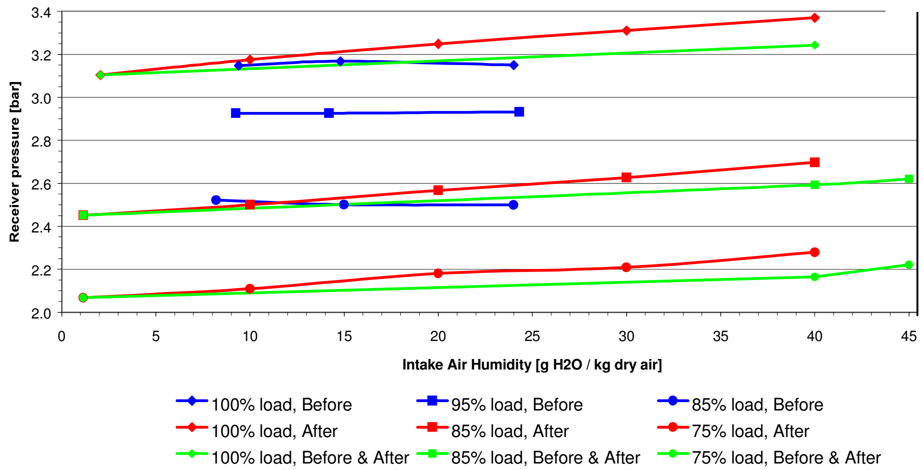


Figure 3.14: Receiver pressure (gauge pressure) for different steam injection locations at high engine loads from Steam injection tests on at WPPP from August 2002 and February 2003

3.6.3 Conclusion from steam injection test

The steam injection gives similar results as the CASS system but the use of steam enables injection also before the compressor and not only after. The NOx reduction is similar as well as the change in fuel consumption. But the increase in boost pressure which is normally the case with water injection does not occur when the water is injected before the compressor. The steam injection both before and after the compressor increased the fuel oil consumption slightly but also over 40% NOx emission reduction is achieved.

Another benefit is that the evaporation of the water is solved before injection to the inlet system which will reduce the corrosion problems if the wall and CAC temperature is kept at a correct level.

4 Comparison & summary of the investigated wet-technologies

First a comparison between all the investigated wet-technology is demonstrated and then each technology is described with pros and cons. In this comparison “only” the wet-technologies are compared with each other to be able to decide which the best “wet-technology” is. They are compared in several aspects and the best technology in one aspect gets the position number one and the second best two and so on. Then all the results from each aspects are summarized and the one with the lowest total position number is the best technology overall.

4.1 Comparison between the wet-technologies

4.1.1 Comparison of the properties of the wet-technologies

In this comparison of the wet-technologies the 11 aspects/category have got the same importance but depending what is most important for the specific customer some of them should maybe have higher importance. This has to be judged from case to case depending on the needed NOx reduction, water availability and the quality of it, the amount of waste heat available and fuel quality etc. However this gives anyway a good overview of which technology is best in each aspect but also the overall competitiveness in relation to each other.

The result from this comparison can be seen in table 4.1, Emulsion has the best overall result and between the others it is quite equal but “humidification with steam” is the wet-technology with the worst over all properties. Note that the desired NOx reduction will have influence on fuel and water consumption as well as PM emissions and costs. In the comparison the *desired NOx reduction is 50%* or more and this means that for example in the emulsion case it is likely that a larger fuel pump and nozzles are needed which will also affect fuel consumption, installation costs and operating costs etc. The operating costs are not mentioned explicit in the table below but it is total sum of the change in fuel consumption, service costs and the cost for the water of required quality. Any eventual reduction in port and fairway fees is not taken into account.

Table 4.1: comparison between all the wet-technologies in the evaluation, the lower the number the better

	Humidification of the inlet air (like Wetpac H)	Humidification of inlet air by steam	HAM	Emulsion	DWI	Stratified Fuel Water Injection	Instantaneous mixing of fuel and water
NOx reduction possibilities	3	3	1	3	2	3	3
PM and smoke reduction	6	5	7	3	4	2	1
Effect on fuel consumption	2	2	1	5	5	5	4
Fuel quality requirements	3	3	3	1	3	2	3
Service/maintenance costs, more exp. if high complexity	1	3	3	2	5	5	5
Price for the system (incl. installation)	1	3	7	2	4	4	4
"Clean" water consumption	7	6	1	2	5	2	2
Water quality requirements	7	2	1	3	3	3	3
Possibility for w-injection adjustment for different conditions, impact when its turned on or of	4	4	4	7	1	3	2
development status/experience time, installations in use	2	6	4	1	2	5	7
Retrofit possibilities -disadvantage if large modifications or space requirements is needed	1	6	7	2	3	3	3
Summary	**37**	**43**	**39**	**31**	**37**	**37**	**37**

4.1.2 Cost comparison

It is very hard to do a cost comparison of several reasons. The main reasons are that it is hard to get Figures about what the investment cost is for a system since it depends on what kind of installation it is (what kind of vessel or power plant, water availability etc) and for how many years the investment is calculated on. Other uncertainties are what the exact NOx reduction will be as well as the maintenance cost for the system will be and if you get any discount on port fees or fairway fees etc. Many of the wet-technologies is newly developed and installed only exists in pilot installations and if the sales volume of the system increases the prices are likely to fall. Anyhow one cost estimation (relatively old) from Transport Canada [5] can be found in table 4.2. The cost and environmental benefits of each technology is estimated based on their hypothetical application on a Canadian case study vessel, the M.V. *Cabot*. This is a RoRo container ship with a 14,600 gross rated tonnage which is powered by two twin Pielstick PC2.5 V12 engines rated at 5.8 MW each. The ship uses IFO 180 fuel oil most (91%) of the time and the engines operate for 6,000 hours per year, at an average load of 85% of the rated maximum output. With an assumed uncontrolled NOx emissions rate of 14.0 g/kWh, the total uncontrolled NOx emissions are estimated at 831 t per year. Note the Continuous Water Injection (CWI) system in the evaluation is a simple humidification system from M.A. Turbo LTD with low NOx reduction potential (only 22%) compared to for example Wetpac H.

Table 4.2: cost estimations for four wet-technologies and a SCR system [5]

	Installed Cost ($)	Operating Cost/(Savings) ($/y)	Annualized* Cost/(Savings) ($/y)	NOx Reduction (%)	Unit Cost ($/tNOx)
Continuous Water Injection	$51,000	($32,000)	($26,000)	22%	($143)
Fuel-Water Emulsions	$325,000	$54,000	$91,000	50%	$217
Direct Water Injection	$413,000	$137,000	$184,000	50%	$443
Humid Air Motor	$1,048,000	$1,500	$120,000	70%	$206
Selective Catalytic Reduction	$1,156,000	$306,000	$436,000	95%	$552

* Annualized costs include operating costs and total installed costs amortized at 10% over 23 years.

As one can see the costs varies very much depending on technology, the costs ranged from a savings of $143 (CWI) per tonne to a cost of $552 (SCR) per tonne of NOx reduced. But if the aim is 50% reduction as in the general comparison in table 4.1 earlier, the water consumption increases a lot, most for CWI but also for emulsion while the fuel consumption increases a more for emulsion than for CWI. This will increase the costs compared to the Figures in the cost comparisons in table 4.2 above and table 4.3 below.

Another cost estimation made by the Swedish Maritime Administration [24] which is mainly based on a cost estimation from Entec [25] and this estimation can be seen in table 4.3.

Table 4.3: cost estimations for DWI, HAM and SCR systems with interesting comments for the different costs linked to each system [24]

NOx reduction cost/t	Capital cost	Operating costs	Life span
Direct Water Injection DWI installed on new ships is estimated to cost € 345-411 per tonne NO_x	Cost of retrofitting is described as relatively high, due to expected need of new cylinder heads which is around ¼ of the cost of a new engine (around € 50/kW)	High water quality is necessary, 90 g/kWh (45% water injection rate). Entec calculated with a distilled water cost estimated € 15/m^3. In many cases, drinking water is used - to a lower cost.	DWI life span is estimated to around 4 years. Rest of the equipment is estimated to have a life time of 25 years.
Humid Air Motor – HAM – installed on new ships is estimated to cost 198 – 268 € /tonne NO_x. As a retrofit 263 – 306 €/tonne NO_x	90-130 €/kW for new built engines and 110-130 €/kW for retrofit.	Maintenance cost is 4 000 €/year for a 5.7 MW engine, around 0.15 € /MWh.	Life span if durable – non-corrosive or galvanised - material is used 25 years.
Selective Catalytic Reduction – SCR For a new ship using MD the cost is estimated to 313-413 € per tonne NO_x and for a new ship using high sulphur residual oil >1,5 % S the estimated cost is 526 -740 €/tonne NO_x	40 – 60 €/kW for new built engines and 60 – 100 for retrofit.	Urea solution €170/tonne. = €2.6/MWh. There are indications of a lowered price-- 120 – 140 € /tonne - due to several companies delivering urea. Transport of the urea may be a considerable part of the urea cost. Maintenance of equipment is required. Entec estimated a need of cleaning equal to € 8 000 per year and ship. The need for maintenance depends on the fuel used. An estimate of operating cost given for one ship (excl. financing) is €10 000 per year.	The catalyst is estimated to require a rebuild every 20 000 hours of operation, when using residual oil.

The NOx reduction costs per tonne depends on the vessel size, the cost per tonne is lower for a big vessel than a smaller vessel. It is more expensive to install a HAM system than a SCR system but the SCR have high operating costs while HAM has exceptionally low. The DWI is expensive to retrofit since it is likely that the cylinder head is needed to be replaced but for a new installation the cost is not that high and the availability of clean water have relatively large impact on the running costs. Note that the SCR cost is very much fuel dependent. With high Sulphur content in the fuel the NOx reduction cost per tonne is very high. Note though that a high NOx reduction can get substantial price reduction depending on location or routs.

4.2 Summary with pros and cons

A benefit for all the wet-technologies in general is that the exhaust gas temperature and engine components temperature decreases more or less while a general disadvantage is that with increased water amount the flow will increase and the boost pressure will increase which can make new turbo matching necessary to avoid surge problems in the TC.

4.2.1 Humidification of inlet air like Wetpac H

Pros:

- This is a relatively cheap wet-technology because it is relatively simple and no engine modifications except for corrosion protection of the inlet system which makes it to attractive NOx reduction method
- With increased receiver temperature it is possible to reach very high absolute humidity and thereby get high NOx reduction
- The vapour tend to give a cleaner combustion chamber and turbine which can reduce engine wear and enable longer service interval
- Relatively proven technology (especially Wärtsilä have quite long experience from this technology)
- Low impact on the fuel consumption

Cons:

- o High water consumption since the vapour have a homogeny distribution in the combustion chamber and not only the flame region
- o Tends to increase PM and smoke with increased water amount as well as HC emissions
- o The highest possible injection amount is dependent on the allowed receiver temperature (if pressure is considered to be given) and are therefore limited in many cases to about 30% NOx reduction with normal receiver temperature
- o Risk for corrosion due to condensation because of insufficient evaporation or too cold walls
- o Risk for clogging and deposit build up in the CAC and droplet catchers/ water mist catchers
- o Evaporated water or water made by reverse osmosis can be needed to avoid clogging and corrosion

4.2.2 Humidification of inlet air with the use of Steam injection

Pros:

- A advantage with steam injection into the inlet air is that the challenge with evaporation of the water is solved externally from the inlet system, this also enables injection of steam before the compressor without risk to damage it
- Enables the use of sea water depending on the steam generator
- Decreased corrosion risk in the inlet system and risk for clogging or deposit build up is reduced
- The fuel consumption is almost unaffected

Cons:

- o High water consumption since the vapour is uniformly distributed in the combustion chamber and not only the flame region as with normal humidification of the inlet air
- o A lot of waste heat is needed to evaporate the water in the steam generators, the higher absolute humidity is therefore limited by this
- o Only a few installations in the world which makes the technology untried and with little experience available the risks for unwanted surprises are higher

4.2.3 Humid air motor (HAM)

Pros:

- HAM is the only wet-technology (together with steam injection in some cases) that can use sea water instead of very good water quality like evaporated water for the humidification
- No impact on fuel consumption or even reduced fuel consumption in some cases
- About 60 to 80% NOx reduction with HAM has been reported from Mariella on high load and even higher reduction on low load. Calculations confirm that this low NOx level is achievable with this "old" engine with relatively low boost pressure. About 60% NOx reduction would be achievable with a HAM system with similar efficiency as on Mariella but equipped to a modern highly boosted engine (like W46F). The highest NOx reduction is achieved at low load
- Increase engine T.B.O. (time between overhaul) 12 to 15 % reported due to reduced engine wear
- Very small additional running cost

Cons:

- o This is quite big and complex and therefore a expensive system to install, the price is quite similar to a SCR system (but HAM is much cheaper to run)
- o The heat demand to heat all the excess water before injection is huge and requires that a lot of waste heat is available
- o Tends to increase PM and smoke with increased water amount as well as HC emissions like in normal humidification of the inlet air
- o A quite large space is needed for the humidifier in the engine room (4*1,2 m depending on engine size)
- o A lot of corrosion protection is needed in the inlet system
- o Engine load acceptance will be affected negatively due to the big air volume after the compressor

4.2.4 Emulsion

Pros:

- Can reduce both NOx and PM even at quite high water amounts
- This is a relatively cheap wet-technology if the engine fuel pump capacity is enough
- Typically no major modifications are needed to the injection system (maybe increased nozzle hole size)
- The running cost is little affected compared to normal operation
- It is a relatively proven technology with experience available
- Possibility to use HFO with high viscosity by using fuel-in-water emulsion
- Moderate water consumption compared to humidification of inlet air

Cons:

- o The possible NOx reduction is limited to the amount of water emulsified into the fuel which is dependent on fuel pump capacity and nozzle hole size, normally 20-30% NOx reduction can be achieved but up to 50% has been reported with larger pump and nozzles
- o the percentage of water is constant and cannot be easily changed and there is always a chance that the mixing fails and that pure water reaches the cylinder
- o the water injection will be as long as the fuel injection since they are mixed, no timing possibilities of the water
- o An undesired temperature decrease early in combustion can occur and cause a ignition delay and engine noise (when higher water amounts are used) at low load since the water comes with the fuel when the fuel injection starts.
- o The viscosity increases with increased water amount and therefore can an increased heating capacity be needed
- o Higher stress on the fuel injection system due to higher temperatures and large flow
- o Depending on the emulsification technology some chemical emulsion stabilizer can be needed with an additional cost
- o The fuel consumption tend to increase with increased amount of water

4.2.5 DWI

Pros:

- DWI allows the fuel-water percentage to be changed for different operations (different loads and conditions as well as NOX reduction levels) and the water spray can point at the flame to get best reduction effect, several injections during the cycle is also possible, can be turned of whenever wanted without effecting the operation
- High NOx reduction potential up to 60%
- Moderate water consumption compared to humidification of inlet air
- Possibility to mainly reduce PM and smoke instead of mainly reduce NOx with the use of late DWI
- A large amount of water can be used
- Relatively proven technology (especially Wärtsilä have quite long experience from this technology)

Cons:

- o The injection system needs to be modified, depending on if the DWI nozzle is integrated with the fuel injection nozzle to one unit or if the DWI nozzle is a separate unit the cylinder head has to be modified more or less. This is expensive and makes retrofit much more expensive
- o Relatively complex and quite expensive water pump and nozzle is needed
- o Since high water pressure is needed the water pump will consume energy from the engine which will have some affect on the fuel consumption
- o The water injection nozzles are exposed (high temperature and pressure) and therefore is it hard to get sufficient life time of the nozzles
- o Risk for wall wetting and contamination of the lube oil if evaporation is insufficient or if something happens to water injection system
- o Tend to increase smoke and fuel consumption when high amounts of water is used

4.2.6 Stratified Fuel Water Injection

Pros:

- Unlike emulsion it is possible to adjust the water amount as well as the timing for different operation conditions and the ignition delay can be avoided since some fuel is injected first
- The cooling effect is better in the flame region compared to normal DWI due to better penetration/mixing with the fuel
- The fuel pump is not a limitation for the amount of water that can be used as it can be when emulsion is used (but the nozzle holes is still limiting)
- Potential for better smoke performance than DWI

Cons:

- o The maximum injection amount is limited by mainly the nozzle holes which have to work good both with and with out addition of water
- o It can be difficult to know exactly how the mixing will be and it is important to have some fuel first because if too much water will end up towards the front of the injection it can cause a significant ignition delay
- o High water quality is needed
- o Corrosion in nozzle and fuel system
- o Quite expensive system similar to normal DWI
- o Relatively high water pressure is needed which increases the energy consumption
- o Unproven technology

4.2.7 Instantaneous mixing of water and fuel

Pros:

- Can reduce both NOx and PM even at quite high water amounts like normal emulsion
- The water injection can easily be timed to get the best performance and ignition delay can be avoided
- the amount of injection water is not limited by the fuel pump capacity or the nozzle hole sizes like in normal emulsion which enables a higher NOx reduction potential
- The same good cooling affect of the flame as emulsion since the water is well mixed with the fuel
- No chemicals for stabilisation of the emulsion is needed since the emulsion takes place in the combustion chamber and there is no risk that only water will reach a cylinder

Cons:

- Quite expensive system similar to normal DWI due to increased complexity and the modifications needed of cylinder head/nozzle
- a "pre water injection" can't take place before the fuel injection during the compression as with DWI
- material wear in the mixing passage from the water spray
- Unproven technology

5 Simulation of 2-stage turbo charging & humidification in GTP

5.1 Introduction to the simulations

The simulation program used in the simulation is GT-Power from Gamma technologies [31] and this program is the most commonly used for engine modelling in the industry. This program is based on one-dimensional gas dynamics that is representing flow and heat transfer of the components in the model. The program is object based and has a graphical interface to the user. GT-Power is a powerful tool for simulation of engine performance and gives good estimation of process flows with temperatures, pressures and pressure waves as well as TC behaviour etc. But since the evaporation of water droplets is very much a tree-dimensional, the one-dimensional program is a limitation in this case and that's why CFD simulation is needed. You can read about that later in chapter 6.2.

The simulated engine is the Wärtsilä 20V32C with 2-stage turbo charging and quite extreme miller timing. In the Miller process the inlet valve is closed before the bottom dead centre (BDC) and after the valve is closed when the piston moves down an expansion of the cylinder charge will occur, leading to a lower temperature of the charge. This will lead to a smaller amount of charge trapped in the cylinder and to compensate for that the charge air pressure needs to be raised to compensate for the reduced charge and that's why 2-stage TC is necessary on this engine. The basic engine data for the engine are:

20 cylinder engine in V-formation
Desired output: 10 000 kw (500kw/cyl)
Cylinder bore: 320 mm
Stroke: 400 mm

The engine should have a 550 ppm NOx-level (dry, 15% O2) without humidification of the inlet air and the NOx-level target with humidification was from the beginning 250 ppm at ISO 3046 conditions (Pressure 100 kPa, absolute humidity 5.97 g water/kg dry air) but this was later on changed to an even more ambitious and demanding target since IMO define a stricter NOx emission limit (IMO -80%) during the work.

From the beginning it was planed to do a NOx model but since no engine test data concerning NOx for this particular engine was available this was out of scope. Instead we have to trust Wärtsilä's NOx-humidity curve which is carefully investigated at a wide NOx reduction range and at many different engines but not with as much as 80% reduction. It is on the other hand doubtable if a NOx model would give a better result when you change the parameters as much as is done in this study and all this has to be evaluated after the engine is tested. To reach the first target an absolute humidity of 34 g water/kg dry air into the combustion chamber according to NOx - humidity curve and as much as 40 g water/kg dry air to meet the new "IMO -80%" limit with the given starting point. The engine performance is simulated with several humidification strategies and with the different absolute humidity's mentioned earlier.

The goal with the simulations is also to find out the most promising setup for the engine test and find out the required specifications in terms of performance issues like TC specification, pressures, required temperatures to be able to make the engine work as supposed when aiming for the ambitious goal (IMO -80%).

5.2 The simulation model

The GT-power model used in the simulations for the reference case is an existing model and with this model as a base the different humidification models were developed. Depending on the case the models had to be quite extensively changed but it is beneficial to have a model with the main geometry, friction and heat transfer coefficients etc tuned to replicate the currant engine as a base for further development of the models to be able to get good accuracy. This was also needful to be able to develop the relatively large number of humidification models in a thesis work.
A GT-post model of the 20-cylinder engine can be seen in Figure 5.1.

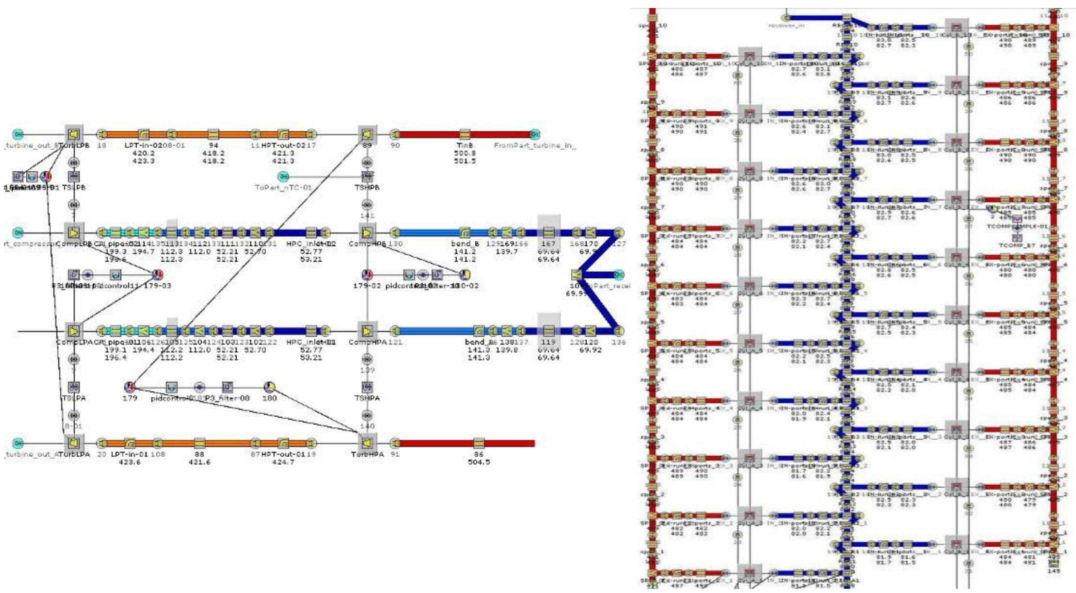


Figure 5.1: The main parts of the GT post model of the 20-cylinder engine

All the simulations are carried out at 100% load since this engine should in the first place be a power plant engine and the TC sizing also needs to be done at full load. The combustion models used in the simulations is a fixed Burn Rate profile which are based on measured pressure curves from the existing engines with one stage TC. This is most likely more accurate than with the use of a wibe function. Another important reason for using this non predictive combustion model is the shorter simulation time since the 20 cylinder engine model would be very slow otherwise.
No NOx model is used as mentioned earlier, instead Wärtsilä's NOx-humidity curve is used instead to determine the needed absolute humidity for the desired NOx level since its doubtable if a NOx model would give a better result when you change the parameters as much as is done in this study. Another concern about the NOx estimation at this very low level is the non thermal NOx like Fuel NOx and prompt NOx since this will be larger and larger part of the total NOx emissions and this part of the NOx will never be reduced by the "wet technologies". This part of the NOx is normally ignored for the

engine type in question but in this case one should have that in mind especially if the fuel used contains a larger amount of nitric oxides.
In the simulations 100% immediate evaporation is assumed which is not the case in the reality, this will give too low temperature at the injection point but further down streams should the temperature be correct so this will only have a marginal effect on the accuracy. It is though important to know when looking at these temperatures. The immediate evaporation is assumed because the evaporation time is very much dependent on the three-dimensional flow field and the droplet distribution as well as the interaction between the droplets which isn't possible with this one-dimensional simulation tool.
All TC maps were not available at the time for the simulations therefore the TC maps used in the model are taken from similar TC. To get the expected efficiency for the real TC, the maps in the model are PID regulated to have certain efficiency at the actual operation point. To get the operation point to the desired place in the map the mass multiplier for the turbine and the compressor was scaled to get the right size of these component and thereby the right output.

The layout of TC and CAC and water injectors
The CACs on the engine and the different injection points used in the humidification cases can be seen in Figure 5.2.

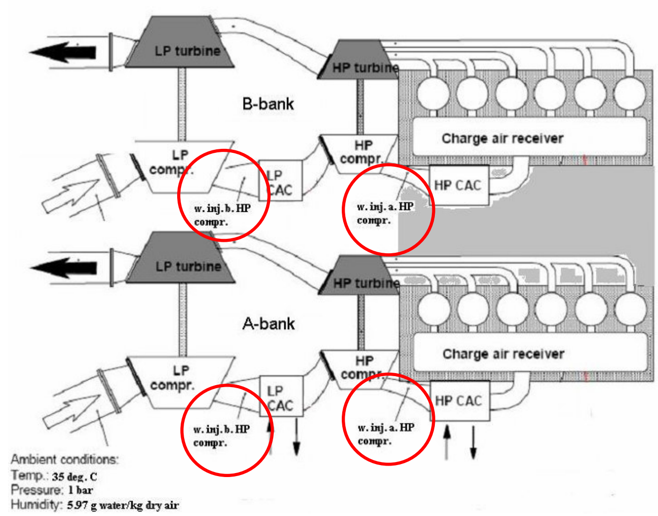


Figure 5.2: The Figure shows a simplified sketch of the engine with the turbo charger system, the charge air coolers (CAC) and the water injection points

Before and after the HP compressors at each cylinder bank there is a CAC, the CAC before HP is called LP CAC and the one after is called HP CAC. Depending on the injected water amount the CAC temperatures had to be adjusted in the humidity cases to higher values to avoid condensation of the injected (and evaporated) water. Some margin to the dew point is desirable and therefore it was decided not to let the relative humidity exceed 90% with any injection amount.

5.3 The simulated cases:

The main cases are presented below with a short description to get an overview. The cases will be explained more in detail together with the result later in this chapter.

1. **Reference:** The engine is the 20V32C with 2-stage turbo charging and miller timing and in this case without water injection. Lower CAC temperatures than in the humidity cases
2. **Humidity:**
    - Absolute humidity level of 34 respective 40g H_2O/kg dry air
    - water injection directly after HP-compressor
    - unchanged heat release
    - In the 40g H_2O/kg dry air case the temperature before HP-compressor has to be increased (85 ºC) to avoid condensation
3. **Humidity with modified heat release**: The same setup as above except for changed heat release
4. **Humidity before and after (B&A):**
    - The same set up as in the humidity case but with water injection before and after the HP-compressor
    - In these simulation the temperature is increased before the Hp-stage also for the lower humidity case (34g H_2O/kg dry air) to the same level as the 40g H_2O/kg dry air case (85 ºC)
5. **Steam:**
    - Same setup as in the humidity case but steam is injected instead of water
    - the temperature does not need to be increased (before the LP CAC) for the 40g H_2O/kg dry air case
6. **Steam before and after (B&A):** Same setup as in the humidity B&A case, except for water is replaced with steam
7. **Steam with modified heat release**
8. **Emulsion**: Same setup as in the reference case but with water emulsion (0.2 water to fuel ratio)
9. **Emulsion & Humidity:** 34 and 40g H_2O/kg dry air absolute humidity in the receiver and addition of water emulsion (0.2 water to fuel ratio)
10. **Emulsion & Humidity with modified heat release:**
    - The heat release is only changed to fit the intake air humidity
    - The change caused by the emulsion is not taken into account
11. **HAM concept on 20V32C:**
    - Replication of the HAM system with the reference engine specification (2-stage turbo charging, miller timing and high compression ratio etc)
    - But CAC is replaced by a HAM vessel
    - Only 40g H_2O/kg dry air abs humidity simulated

Other interesting simulations

1. **Air Waste-Gate (AWG)**: same setup as the humidity but with air waste-gate, only 34 g H_2O/kg dry air humidity is simulated
2. **Bypass:** same setup as the humidity but with bypass from HP-compressor outlet to the HP-turbine inlet, only 34g H_2O/kg dry air

Many different cases are simulated and investigated. But the main focus is on the cases called “Humidity” and “HAM concept“. The reason to focus on these cases is that the humidity cases is a development of the Wetpac H system and the reason for the HAM replication is to se how this promising system would work on a engine like this. Other reasons are that these cases also can reveal many of the important results from similar technologies like steam injection and of course because they are considered to be promising and important. In the result graphs is the Humidity cases named “34g/kg normal CAC”, “34 g/kg 80 ºC CAC” and “40g/kg 85 ºC CAC”. The temperature in the name refers to the LP-CAC water temperature while the HP-CAC temperature is 85 ºC in all cases. This should be compared to the reference case which has a temperature after the LP CAC of a bit more than 50 ºC and the temperature after the HP CAC is around 65 ºC.

Water injection before and after the HP-compressor has the potential to increase power output/efficiency of the compressor and the engine but is very tricky and challenging in reality. Care must be taken to maintain compressor stability, handle erosion risks and blade blasting etc. In the simulations of “Humidity B&A” the water is evaporated before it enters the HP-compressor and this is what we are aiming for, but if instead water droplets were allowed to enter the compressor a better efficiency would be achieved. In this case droplets will enter the compressor bellmouth where they evaporate and performing “compressor spray inter-cooling”. Compressor spray inter-cooling helps to reduce the parasitic work and at the same time increases the air mass flow (higher density) through the engine which will increase the output and the efficiency. This spray inter-cooling is some times used in gas turbines and a rule of thumb for gas turbines is that one percent of injected water relatively to the intake air boosts the turbine power with 5-7 percent [26].
The prime parameters for achieving this are the droplet size, size distribution and the concentration, water quality and temperature. It is of absolute importance to make sure that the droplets are very small to not destroy the compressor blades in the long run and to get the TC manufacture convinced that the compressor will last for at least 50000 hours is also a very challenging task. Also note that if the droplets don’t evaporate the performance benefit will suffer heavily as well as the life time of the compressor. It might be possible to do a droplet catcher which only let the small droplets go through and catches the larger ones to avoid this. Another concern is the required cleanness of the water, probably distillated water is needed to avoid deposits to build up and how to get the distillated water. In the end this is a very demanding technique but can also give nice and attractive advantages.

Steam injection is investigated mainly because with this technology one can have the injection before the HP-compressor without any risks of destroying the compressor blades. This technology solves the problem to get all the injected water to evaporate before it reaches the cylinders since all water have already been evaporated externally. This may also enable the use of salt water (depending on the steam generator) and the temperature in the LP-CAC does not need to be increased because there is no evaporation that reduces the temperature of the charge air. Increased temperature in the CAC before the HP stage would otherwise reduce the efficiency. But on the other hand this is a more expensive solution that needs energy from somewhere else than the charge air to evaporate the water.

The reason for simulating the emulsion and the combination with humidification is that this could be the solution if the required absolute humidity is not reached or if the NOx level for some reason is higher than expected. Another very important advantage is the smoke reduction which all the other wet technologies do not have and that might be necessary to avoid the increased smoke from the wet technology.

Air waste gate and by pass were investigated because if the increased flow will cause surge problem, this can be solved with some of those techniques. It is also interesting to see how this will affect the overall performance of the engine.

Another issue with this investigation was to investigate how the humidification affects the combustion and thereby the heat release as well as to see how the changed heat release will affect the performance. A new burn rate was calculated for the cases named "modified heat release HR" to replicate the combustion with the humidified intake air while to ones without has the same burn rate as the reference.
The result from all the simulations will be presented in the following sections.

DWI was not simulated since the main focus of this study is on the techniques with humidification of the inlet air. Another reason is that Wärtsilä already have developed a DWI system with god performance while a need for further development of a humidification of inlet air system like Wetpac H is needed for the new engine and that's why the focus is on humidification of inlet air in this simulation study.

5.4 Relative humidity, temperatures and pressures

To decide what temperature is needed to have good evaporation and avoid condensation some calculations had to be done. When the water evaporates the temperature will drop due to energy (heat) required for the phase change form liquid to gas. The heat can be calculated with the heat equation (5.1) and if assuming constant pressure (open pipe) the equation for the vaporization heat can be defined as (5.2) and this heat is equal to the reduction in heat of the charge air due to the temperature drop.

$$dq_r = dh - v \cdot dp \qquad (5.1)$$

$$dq = dh = dh_{evap.} = dh_{temp.drop} \ \ [kJ/kg] \qquad (5.2)$$

Because the air from the compressor entering the injection section already contain some vapour this has to be taken into consideration and the enthalpy h for moisture air can be defined as equation (5.3) according to [27] where $x_{vap.}$ is the vapour concentration.

$$h = h_{air} + x_{vap.} \cdot h_{vap.} = c_{p\,air} \cdot T + x_{vap.}(r + c_{p\,vap.} \cdot T) \qquad (5.3)$$

In this equation r is the evaporation heat (phase change) which is the difference in enthalpy for saturated steam and the enthalpy for the liquid according to equation (5.4).

$$r = h'' - h' \qquad (5.4)$$

The evaporating heat for water is huge, about 2500 kJ/kg depending on temperature and pressure. The Specific heat capacity for air, $c_{p_{air}}$ is about 1.0 kJ/(kg*K) and the specific heat for vapour, $c_{p_{vap.}}$ is around 1.86 kJ/(kg*K).

By combining the equations, the temperature drop of the humid air due to the evaporation of the water can finally be assumed with equation (5.5).

$$\Delta T = \frac{x_{vap.add} \cdot r}{Cp_{d.air} + x_{vap.before} \cdot Cp_{vap.}} \tag{5.5}$$

Depending on the injection water temperature the energy for heating the water will reduce the temperature of the air flow as well but this energy is very small compared to the energy for evaporation, especially in this case when the injection water have high temperature from when it is injected. Anyhow, the equation for heating the water is defined in equation (5.6) and the specific heat for water is 4.19 kJ/(kg*K).

$$h_{water} = c_{p_{water}} \cdot (T_{water\_after} - T_{inj.water}) \tag{5.6}$$

If the injection water is cold this can have some impact on the final temperature since the specific heat for water is high. The final temperature of the moisture air after the injection is also dependent on the heat transfer to the walls but this is not taken into account in the calculations above.

In the reference case (without water) the temperature is only 140-145 ºC after the HP-stage and the reason for the relatively low temperature is the low pressure ratio over this stage since the pressure split between LP and HP is about 65/35. If the desired humidity is 34g H_2O/kg dry air about 28g H_2O/kg dry air needs to be injected since the ambient air contains 5.97g H_2O/kg dry air at ISO 3046 reference conditions. With this injected amount and the flow of around 18 kg/s the temperature would drop about 65 ºC according to the equations above but if the aim is 40g H_2O/kg dry air the drop would be around 85 ºC at 8 bar (a) pressure.

The desired maximum relative humidity is 90% and with this limit the lowest temperature allowed is 78 in the 34 g/kg case and 82 in the 40 g/kg case at 7.5 bar (a) pressure which is the pressure in the reference case (the dew point is 76 resp. 79°C). Already the temperature in the 34 g/kg case will be too low due to the evaporation but when the water is introduced, the charge air pressure will increase close to 8 bar (a) and another 2 ºC will be needed to compensate for the pressure increase, in all 80 and 84 ºC is needed.

The results from the simulation concerning the temperatures for the inlet can be seen in Figure 5.3, the temperature drop due to evaporation after the compressor for 34- and 40 g/kg is 64 resp. 82 ºC , the temperature drop due to evaporation (difference between purple and green line in the figure) and the LP-CAC temperature had to be increased to 85 ºC to avid condensation in the 40 g/kg case. It is close to the desired temperature or even under for the 34 g/kg case when the LP-CAC has normal temperature.

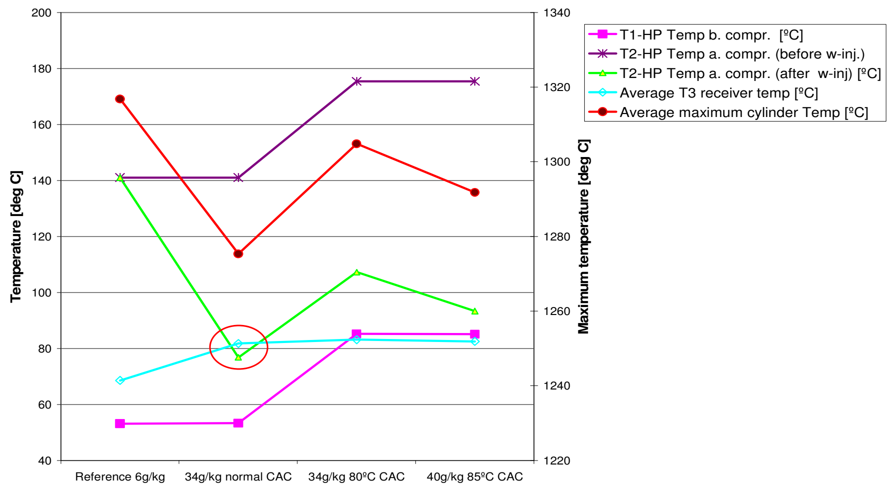


Figure 5.3: Temperatures in the inlet system and the maximum temperatures for the four main cases

In the "34g/kg normal CAC" case, the HP-CAC is working as a heater (red circle in the figure 5.3) while in the 40g/kg case, the HP-CAC is instead used to bring down the temperature before the air enters the receiver. If the CAC is disconnected it is possible to improve the evaporation (or maybe increase the amount of water) but this can cause other problems. Further increased temperature will increase thermal stress but it will more importantly increase the maximum cylinder temperature if the absolute humidity is kept constant. Another problem in this scenario can be boiling of the cooling water in the CAC if the flow is too low. Increased water amount will also be problematic due to increased receiver pressure since it is already high and surge problem in the compressor is likely to occur in that case, this will be explained later.

The influence of receiver pressure and temperature is great, a small change in receiver pressure or temperature has a large influence on the relative humidity and therefore is the temperature distribution in the receiver also be important. The temperature distribution in the receiver is not homogenous, there are a few degrees difference in temperature between the cylinders and this affects the relative humidity quite a lot. Between the hottest cylinder, number 10 and the coldest cylinder, number 1 there are up to 10% difference in relative humidity, see Figure 5.4.

In the 34 g/kg cases there is no problem to stay under the desired maximum Relative Humidity (RH) no matter which injection technology is used but for the 40g/kg is it only the case with steam injection before and after the HP-compressor that is under the limit (marked with a green circle in the Figure) due to the highest receiver temperature. Note that all cases are either under or very close to the limit if one calculates the RH from the average receiver temperature and therefore it would be beneficial to insulate the cold part of the receiver to reduce the difference. Another solution is of course to rise the HP-CAC temperature a bit more but this would increase the fuel consumption and increase the thermal load even more. Anyhow it is important to have the possibility to adjust the CAC temperature for testing.

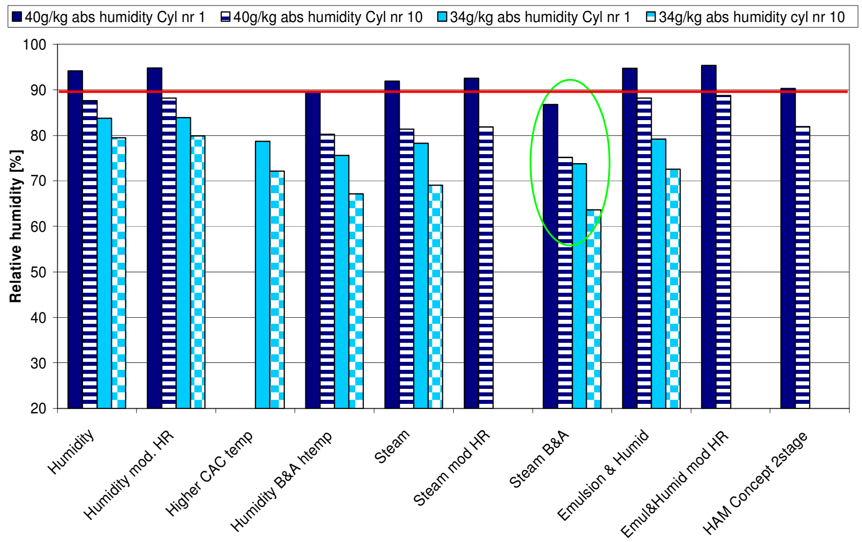


Figure 5.4: The relative humidity at different cylinder inlet for all cases with the desired humidity of maximum 90% marked with a red line

Normally the receiver temperature should be as low as possible to get as much charge to the cylinder as possible in order to get increased output and reduced compression temperature. But high temperature is needed for the humidification to avoid condensation of the evaporated water. The increased temperature can also be beneficial when miller timing is used to avoid ignition problem and long ignition delay at lower load (since the expansion with miller timing can reduce the temperature too much).

5.5 The affect on combustion due to humidification

To estimate the change of the combustion when introducing humidification of the inlet air the apparent Heat Release and Burn rate had to be investigated from measured pressure data. From the pressure data the apparent Heat release was calculated in an Excel program and the apparent Burn Rate was calculated in GT-Power with the use of the EngBurnRate object.

5.5.1 Heat release when humidification of the inlet air is used

Unfortunately there were no available data for W20V32C with and without humidification but to get an understanding and an estimation of how the heat release changes. Data from a similar but smaller engine type W6L20C was investigated.
The heat release calculated from pressure trace collected from CASS test on the engine W6L20C are shown in Figure 5.5 and the burn rate calculated from the same pressure trace can be seen in Figure 5.6.

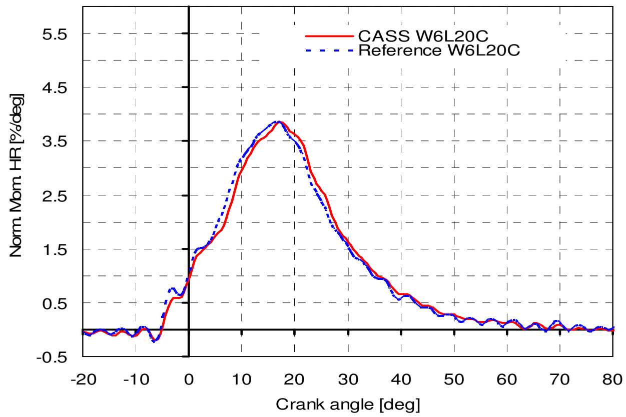


Figure 5.5: Normalised momentaneous heat release with respect to crank angle degrees

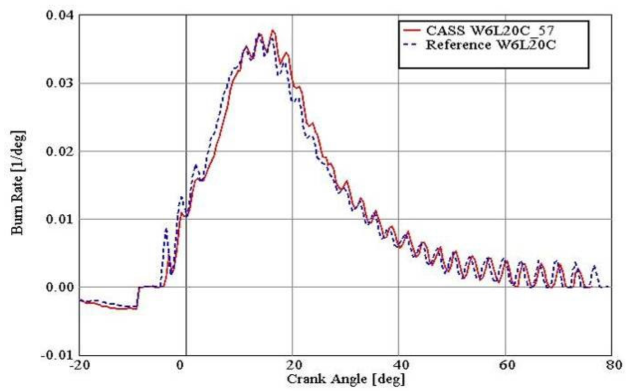


Figure 5.6: Burn rate with respect to crank angle degrees

When comparing the shape of the graphs in these Figures one can make the conclusion that the shape of the apparent Heat release and the burn rate are about the same except for a bit more fluctuations for the burn rate which is very much depending on calculation technical issues (order of the cubic-fitting smoothening filter), also note the different units.

The absolute humidity is about 4g H_2O/kg dry air in the reference case and about 57g H_2O/kg dry air in the CASS case. The most important conclusion from the calculations is that the combustion and thereby the heat release is little effected by the humidification since the shape of the curves are almost the same for the reference and the CASS case but there is though a small delay of about one deg when the humidified air is introduced. This is easier to see in Figure 5.7 of the cumulative heat release below. The small change in heat release is also supported by observed changes of THC and CO emissions and the fact that the ignition delay is not changed, according to test results from Wärtsilä and Pielstick.

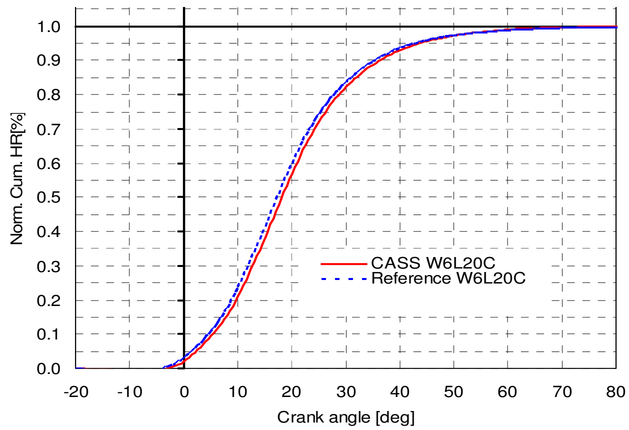


Figure 5.7: Cumulative heat release with respect to crank angle degrees

From these analyses the burn rate was delayed with about 0.5 deg in simulations for the humidification cases named modified heat release. The delay was scaled to the actual (34 and 40 g/kg) lower humidity level used in the simulation compared to the measured case.

5.5.2 Heat release when emulsion is used

Figure 5.8 shows the calculated apparent heat release for the much larger W6L46C2 common rail engine. The heat release from a common rail engine has a different shape of the heat release compared to a pump injected engine and was therefore not usable for the simulated engine which does not have common rail. The HR-curve for the common rail engine flattens out at the top which is not the case for the engine with classical injection system (twin pump) but it is anyhow interesting to see how the combustion changes when emulsion is used.

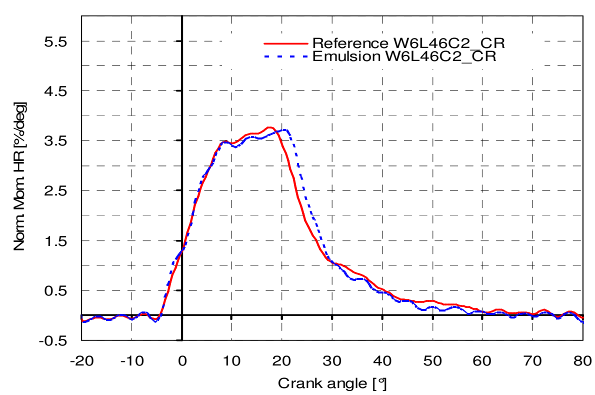


Figure 5.8: Normalised momentaneous heat release with respect to crank angle degrees

When emulsion is used the maximum HR is decreased and with a bit delayed HR top, i.e. more of the HR takes place at later crank angles. This is more obvious when looking at the cumulative heat release, see Figure 5.9. The emulsion changes the HR differently than the humidification of the inlet air but since data from the engine in question with emulsion was not available at the time for this investigation it was decided to only change the HR for the humidity cases.

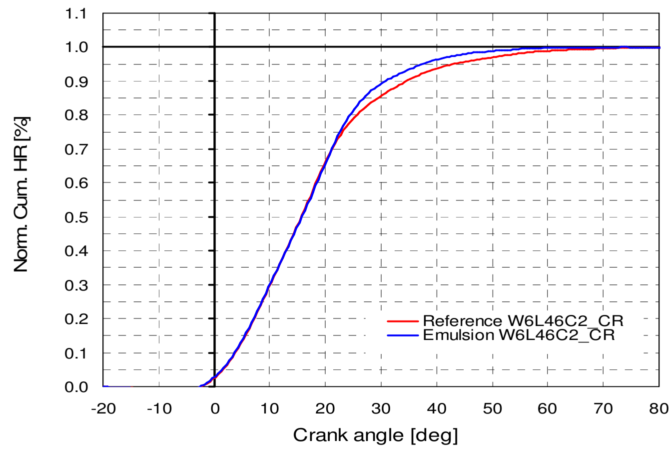


Figure 5.9: Cumulative heat release with respect to crank angle degrees

5.5.3 Maximum cylinder temperature and heat capacity

When the charge air is humidified the maximum cylinder temperature will drop. The drop in temperature is mainly caused increased mass (added water and higher boost pressure) and increased heat capacity. The maximum cylinder temperature is reduced with about 50 ºC from reference case to the case with 34g H_2O/kg dry air humidity, see Figure 5.10, even if the temperature at the start of combustion is higher with humidification (the receiver temperature is 13-14 deg higher than reference), see Figure 5.11. Note that the Figures below shows the average temperature in the cylinder charge for a given crank angle and in a diesel engine the local flame temperature around the diesel spray varies a lot in the cylinder from point to point. This means that the temperature can be much higher at the hottest spots in the reality. Some of this phenomenon can be seen in the burned zone temperature which is about 300 degrees higher than the average cylinder temperature, see figure 5:12. The change in heat release is not taken into account in the Figures below since the change is very small as mentioned earlier.

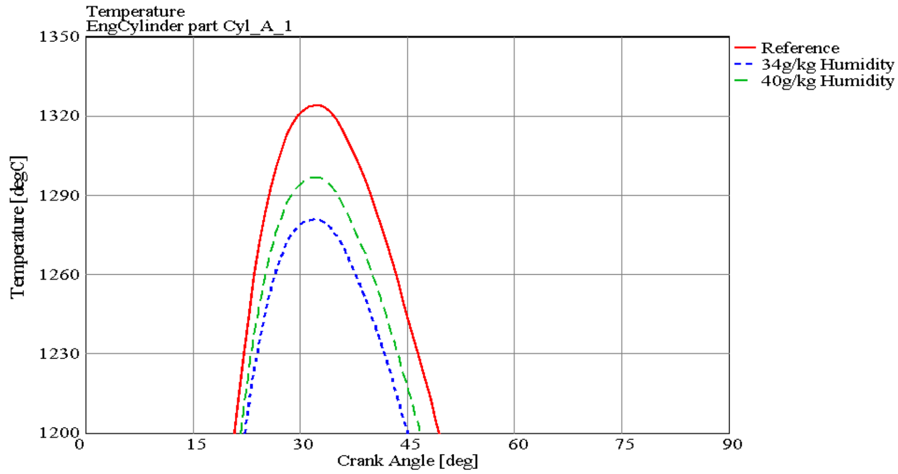


Figure 5.10: Average maximum cylinder temperatures, note that the receiver temp is 13-14 ºC higher in the humidity cases compared to the reference case

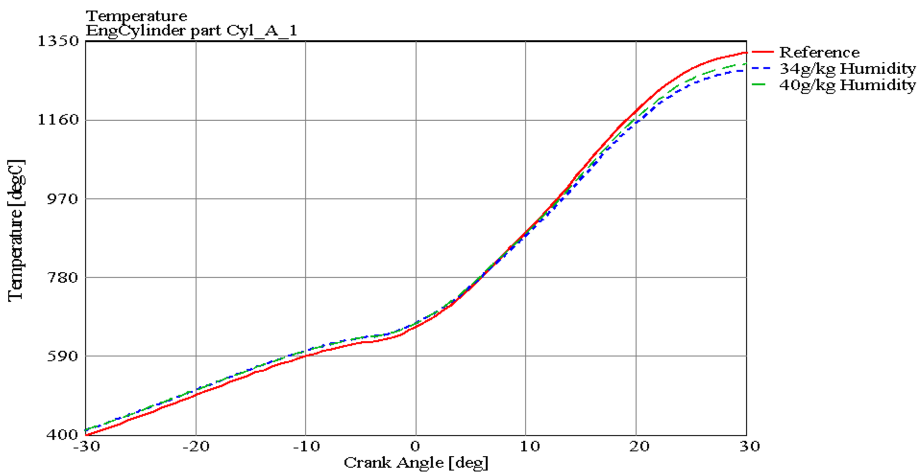


Figure 5.11: Cylinder temperature during the compression, the temperature is higher at the start of combustion in the humidity cases but even though lower than the reference at the maximum

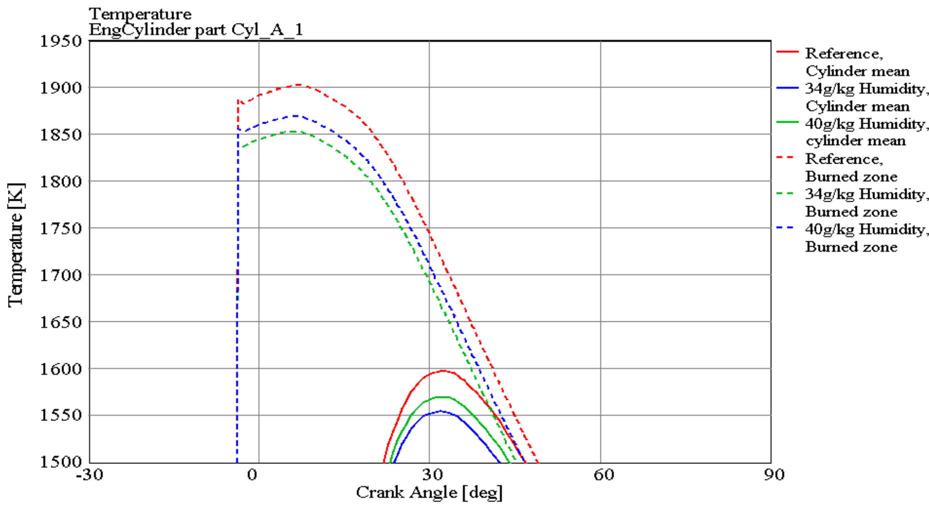


Figure 5.12: Burned zone and average cylinder temperature with respect to crank angle degrees

Normally the boost pressure increases with increased humidity (vapour) due to the increased flow over the cylinders and the energy to the turbine. The increased mass needs more energy to be heated which reduces the maximum temperature with a given amount of fuel. This means the receiver pressure increases with increased water to fuel ratio but if the temperature in LP-CAC (before HP-compressor) is increased the pressure will instead fall (worse efficiency) and the maximum cylinder temperature will go up again, see Figure 5.13 and Figure 5.14.

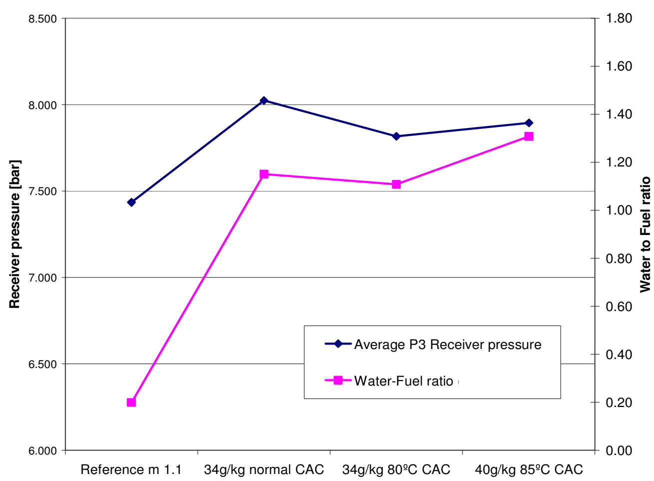


Figure 5.13: receiver pressure and water-fuel ratio on the y-axis and from the names on the x-axis the temperature and humidity can be found

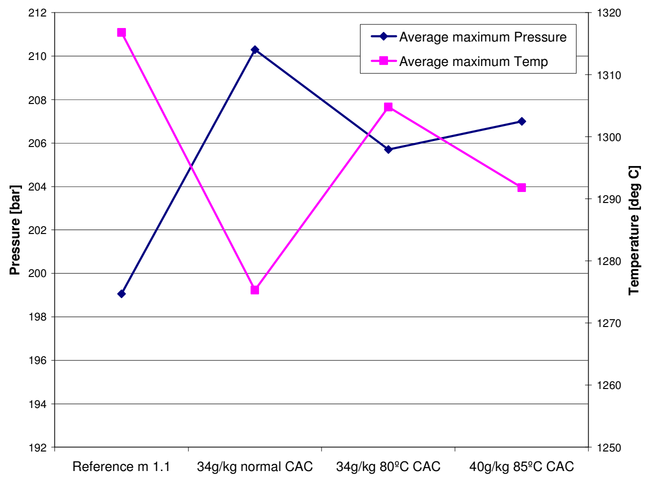


Figure 5.14: Maximum pressure and maximum temperature for the four main cases

In the Figure 5.13 and 5.14 one can see that when the cylinder pressure goes up due to the increased humidity, from reference case to the "34g/kg normal CAC" case and the temperature goes down. Then (the following case to the left, "34g/kg 80 ºC CAC") when the humidity is constant but the temperature in the LP-CAC is increased the pressure goes down and the cylinder temperature goes up. This is the reason why the 34g/kg case has lower average maximum cylinder temperature than the 40g/kg case (even if the receiver temperature is almost the same for these cases) since the temperature before HP-compressor had to be increased to avoid condensation in 40g/kg case but not in the 34g/kg case. The increased temperature before the compressor reduces the boost pressure with 0.2 bars which is the main reason why the 40g/kg case not has a lower maximum temperature than the 34g/kg case with normal LP-CAC temperature.

The specific heat increases with increased absolute humidity which of course will contribute to reduce the temperature in the humidity cases and the increase of the specific heat is about 40-50 J/kg-K, see Figure 5.15.

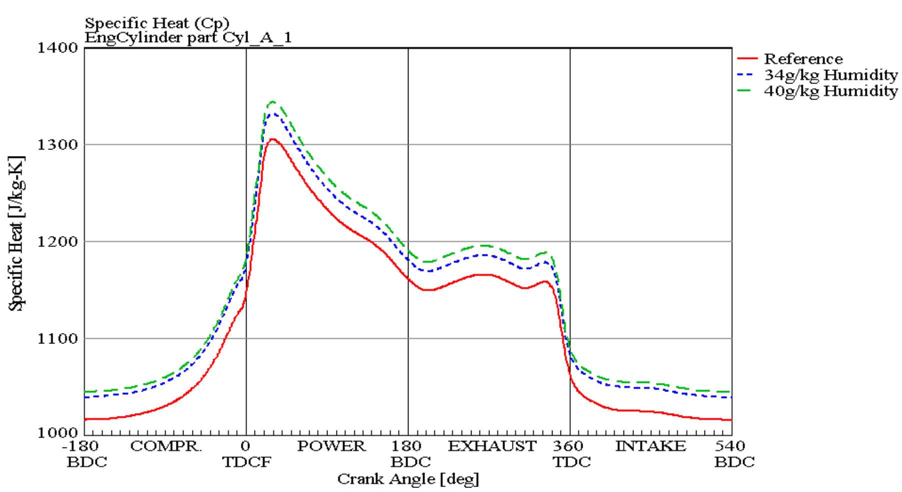


Figure 5.15: The increase in heat capacity or specific heat during the cycle.

The maximum mean temperature for all the simulated cases can be found in Figure 5.16. In general the cases with modified heat release have the lowest cylinder temperature due to a bit later (delayed) combustion which reduces the temperature compared to identical case with unchanged HR. The cases with injection before and after HP-compressor have higher maximum cylinder temperature than the cases with injection only after HP-compressor due to the increased temperature before the HP-compressor (to avoid condensation). This reduces the receiver pressure which will lead to increased maximum temperature (as in the 40g/kg case above). The conclusion from this comparison is that the cases with the highest humidity have not always the lowest temperature as one might think.

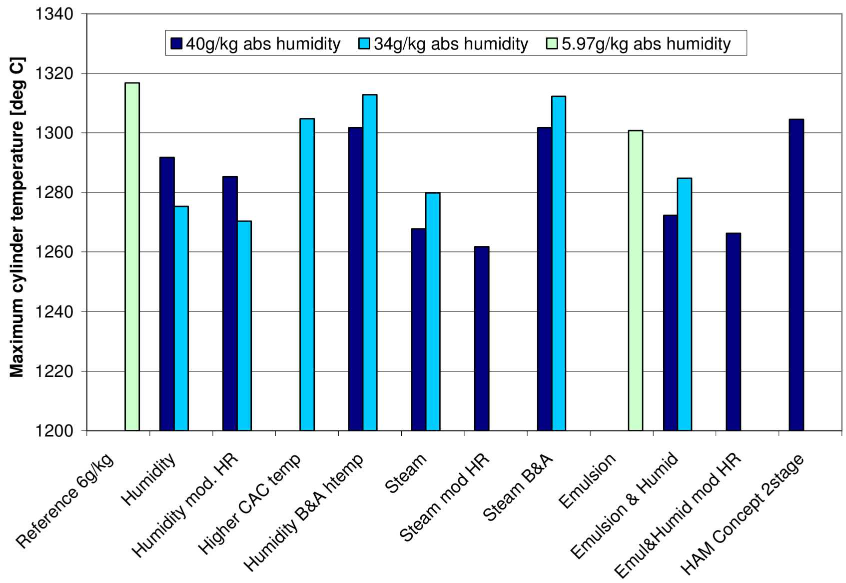


Figure 5.16: The maximum cylinder temperature for all the cases, note that the "Emulsion & Humid" cases have higher absolute humidity due to the water from the emulsion, the right value is 40.6 resp. 46.6g/kg instead of 34 and 40g/kg as marked in the Figure

Steam injection with modified heat release has the lowest temperature this is mainly due to the fact that the temperature in the LP-CAC does not need to be increased even when aiming for the 40g H_2O/kg dry air humidity (this is done in all other 40g/kg cases) since the water is already evaporated and therefore will not cause any temperature reduction of the charge air.

5.5.4 NOx issues

The most important factor for NOx formation in a diesel engine is the high combustion temperature and in that respect so far the "steam modified HR" and "emulsion and humidity" looks most promising but the oxygen and nitrogen concentration is also important for the NOx formation. With increased humidity both oxygen and nitrogen concentration will be reduced since the vapour will dilute the charge air and thereby reduce the lambda value (lower lambda => less oxygen), see Figure 5.17. From this

Figure one can see that the "steam modified HR" case has higher lambda than for example "steam B&A" case which had high cylinder temperature but the "emulsion & humidity" is again the most promising case regarding the NOx mainly due to the highest absolute humidity (46.6g H_2O/kg dry air). When the receiver pressure is reduced the air flow and the effective lambda will be reduced with the same magnitude at a given humidity (but increased humidity gives of course lower lambda).

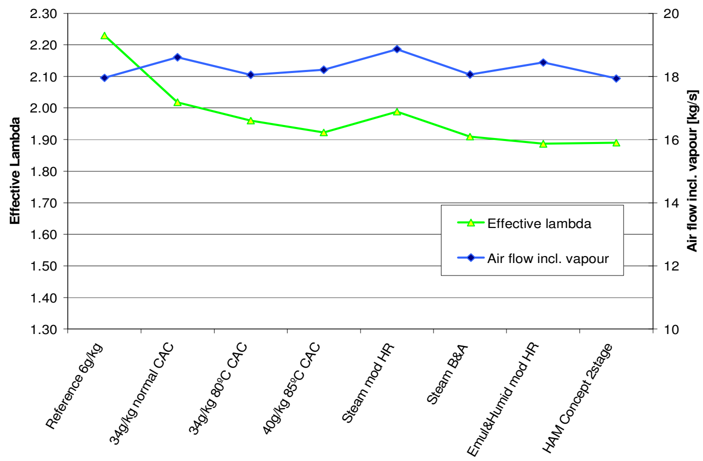


Figure 5.17: Effective lambda and air flow for comparison, the effective lambda reduces with increased humidity or reduced receiver pressure

The vapour is replacing air and dilutes the cylinder charge which reduces the O2 and N2 concentration. This is beneficial for reducing the NOx emissions. The cases with the lowest maximum temperature and the most diluted cylinder charge have therefore the best possibilities for reducing the NOx formation.

5.6 Cylinder pressure

All cases with humidification gets increased receiver pressure and a higher receiver pressure gives high cylinder pressure as expected. The maximum cylinder pressure can be seen in Figure 5.18 and the receiver pressures can be found in Figure 5.19. The maximum pressure follows the receiver pressure pretty much, i.e. the cases with high receiver pressure have therefore also high maximum pressure. The cases with changed HR have lower cylinder pressure at a given humidity due to the slightly delayed combustion but the difference is small.

When water is introduced the margin to the maximum cylinder pressure recommendation for the actual engine is small or even excided in some cases. But engine tests have shown that some of the increased receiver pressure will not increase the maximum pressure as much as an equal increase in pressure without vapor would do. A few bars static difference in the maximum pressure between different cylinders was noticed.

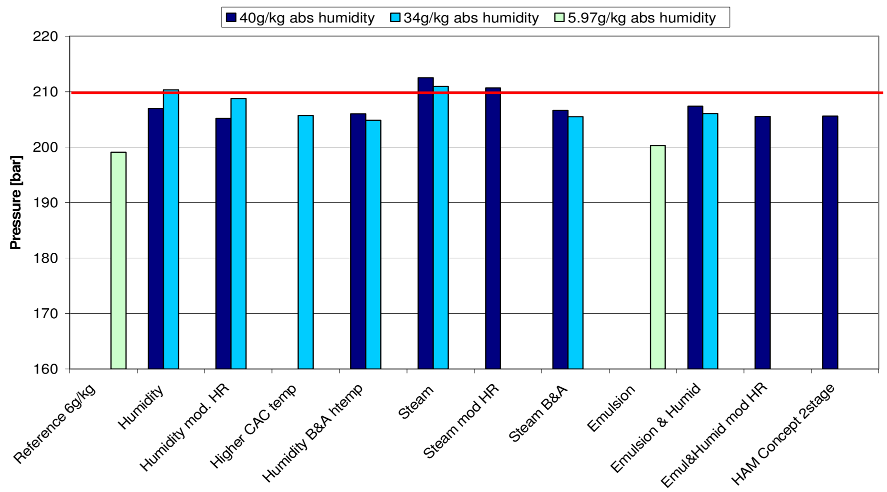


Figure 5.18: The maximum cylinder pressure for all cases

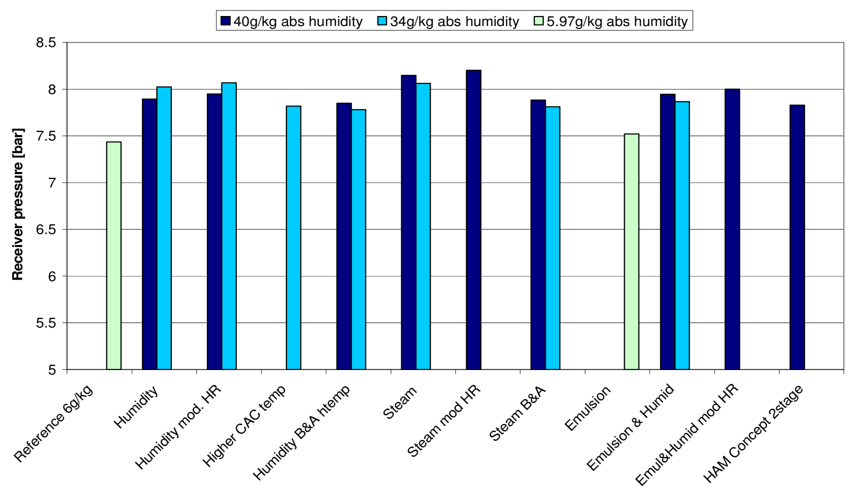


Figure 5.19: the receiver pressure for all cases

5.7 Turbo issues

5.7.1 Influence from the injected water on the TC

When water is introduced to the charge air the operation point for the compressors will be changed while the operating point for the turbine is almost unaffected. The problem in this case is that TC-setup has to work with good efficiency both with and without the humidification and not run into surge in either case. This means that the operating line/point has to have same margin to the surge line when the humidification of the inlet air is used and when it is not in use the operating line/point have to still be at a filed in the map with good efficiency.

The pressure split between the LP and HP stage is around 65/35 which means that the pressure ratio is about 4 for the LP-stage and about 2 for the HP-stage. Due to the high pressure ratio over the LP-stage the operating point at high load is high up in the operating field and the margin to the speed limit is small while the TC speed for the HP stage is not a problem because of the lower pressure ratio.

The turbo speed of the LP-stage increases more with increased amount of vapour (due to the increased pressure) than for the HP-stage. The turbo speed of the HP-stage increases instead with the increased temperature before the HP-compressor (to avoid condensation) due to decreased density, see Figure 5.20. The difference in speed is though quite small but on the other hand the TC speed is quite close to the speed limit of 27000 rpm.

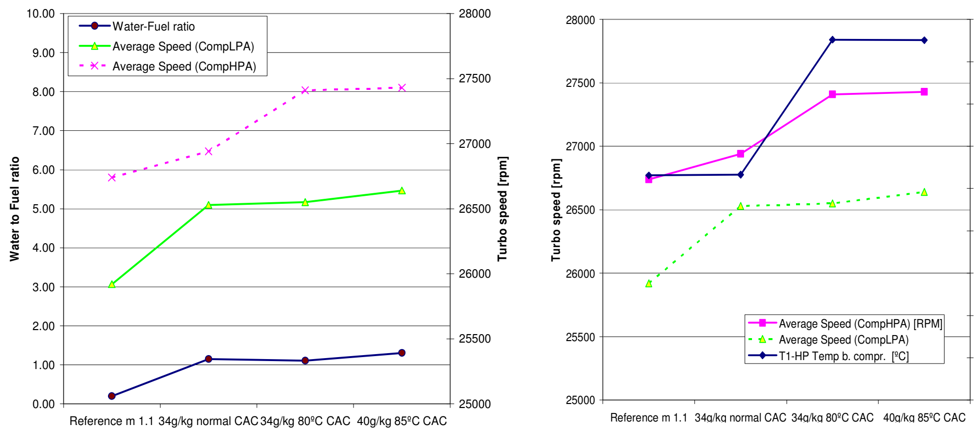


Figure 5.20: Turbo speed and water to fuel ratio (to the left) respective turbo speed and temperature before HP-compressor (to the right) for four different cases

The LP-turbo speed for all cases is shown in Figure 5.21 and one can see that the cases with the highest humidity has the highest speed and in some cases the margin to the speed limit might be too small. “Emulsion & humidity mod HR” has the highest speed but all cases with modified HR tend to have higher speed than cases without.

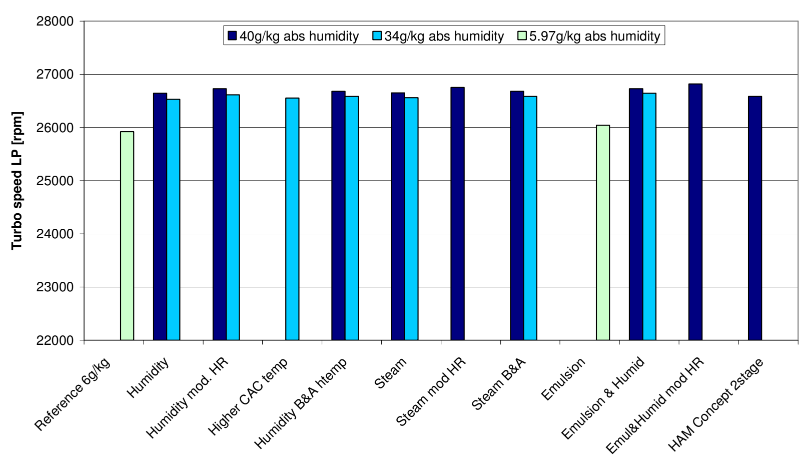


Figure 5.21: The LP-Turbo speed for all cases

The LP-compressor map is shown in Figure 5.22 with the operating points for the reference, 34 and 40g/kg cases. In the 34 g/kg case the point is at a good position but for the 40g/kg case the operating point is very close to the surge line. To be able to run the engine test at this humidity level without risk for surge the diffuser ring needs to be changed. With another "smaller" diffuser ring it is possible to gain 5% in surge margin and in that case the margin would be almost the same for the 40 g/kg case as what the 34 g/kg case has now in Figure 5.22.

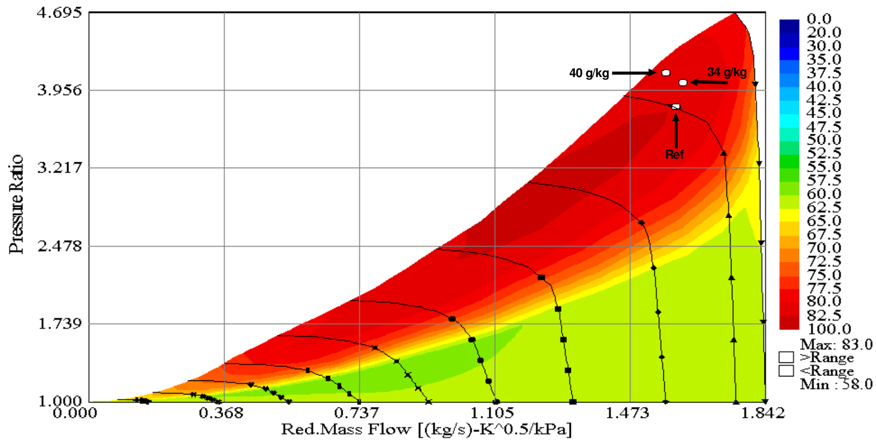


Figure 5.22: The operating point in the efficiency map for three different humidity's for the LP-compressor

For the HP-compressor it is possible to find a setup that works with and without humidification. When mass multiplier 1.1 is used instead of the original 1.2 the operation point is ok for both cases which means that the compressor should be 0.1 units smaller than in the original Reference case. The pressure ratio over the HP-compressor is slightly increased in the 34g/kg case but not in the 40 g/kg case, see Figure 5.23. The operation points have moved to the left which means that the mass flow is reduced and the reason for that is that the density goes down when the temperature is increased before the HP-compressor.

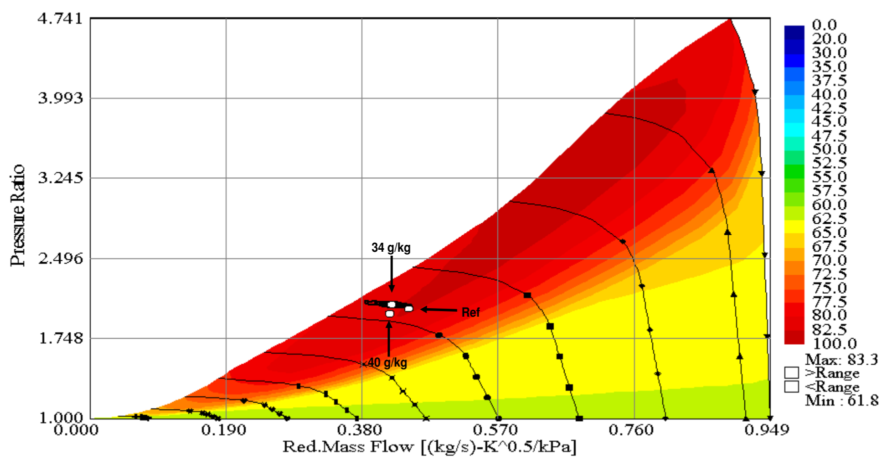


Figure 5.23: The operating point for three different humidity's in the efficiency map for the HP-compressor (mass multiplier 1.1 is used instead of 1.2 for the presented cases)

5.7.2 ASD/Bypass and Air Waste Gate

Instead of resize the TC modules to avoid surge and get the right performance two other solutions were investigated for comparison, Anti Surge Device (ASD)/Bypass and Air Waste Gate (AWG). The setup is like in the “34g/kg humidity normal CAC” except for the Bypass or AWG. Bypass is used to avoid surge by redirecting some of the flow from the compressor outlet to the turbine inlet and in the AWG case charge air from the receiver is “wasted” out to the surroundings via a valve.

The Bypass device was simulated with three different orifice diameters, 37, 30 and 20mm (named B.P C1, C2, and C3) with one orifice for each cylinder row while the AWG was simulated with the following waste gate diameters per cylinder bank: 25, 35 and 45mm (named W.G. C1, C2 and C3). A bypass keeps the pressure ratio or slightly increases it but the flow over the compressors increases and the LP-TC speed increases greatly, see Figure 5.24. The Air waste gate on the other hand reduces the pressure but doesn’t affect the flow over the LP-compressor like the Bypass does.

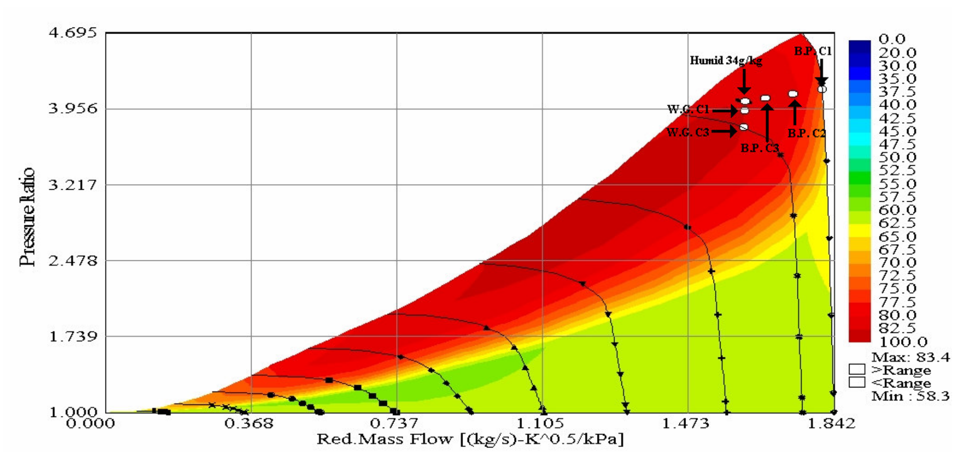


Figure 5.24: Efficiency map for the LP-compressor when bypass and AWG is used

The HP-stage is again less affected, the pressure ratio stays almost the same while the flow increases just slightly with increased bypass or AWG flow, see Figure 5.25

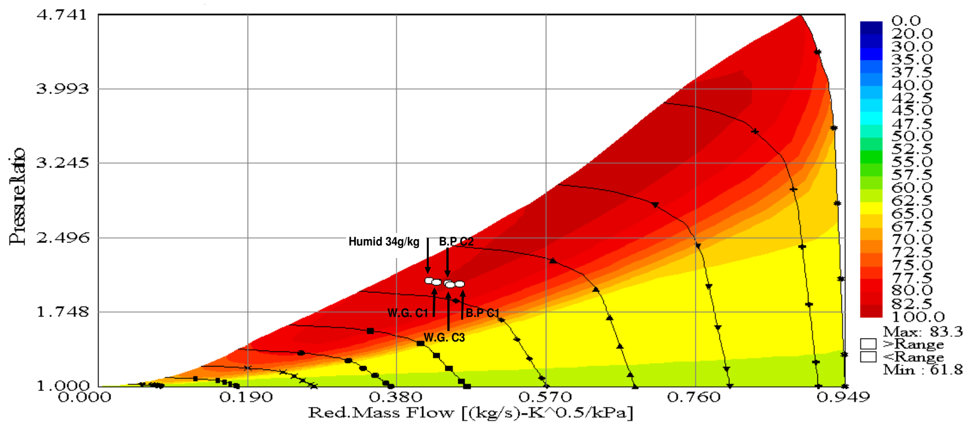


Figure 5.25: Efficiency map for the HP-compressor when bypass and AWG is used

The main conclusions from these simulations are that if you open the bypass little too much the LP turbo speed will rapidly be too high but note that there is a compromise where it may work (30mm orifice). The disadvantage with both of these solutions is the fuel consumption penalty, this will be explained more later. Due to the very limiting TC speed in this case the AWG is a better solution for the engine test and will therefore be used but only as a safety feature. The surge problem should be solved with a properly chosen size of the HP-compressor stage and a "smaller" diffuser ring for the LP stage as explained earlier.

Fuel consumption when Bypass or Air waste-gate is used

Both Bypass and Air WG increases the fuel consumption. The consumption increases with increased "waste flow" even though the increase is moderate when a reasonable restriction diameter is used, see Figure 5.26. The fuel consumption increases at maximum 4g/kWh in the worst case and in that case the waste flow is about 10%. Note that the case setup is like in the "34g/kg humidity normal CAC" except for the Bypass or AWG. Tests have though showed that a moderate bypass can reduce the consumption but this depends on where the actual operation point is. Bypass is used to move the operating point from a point, normally close to the surge line, to a point a bit more to the right in the map the TC normally have a higher efficiency there and in that case the consumption will go down but if the TC have the same efficiency in the new point the consumption will instead go up.

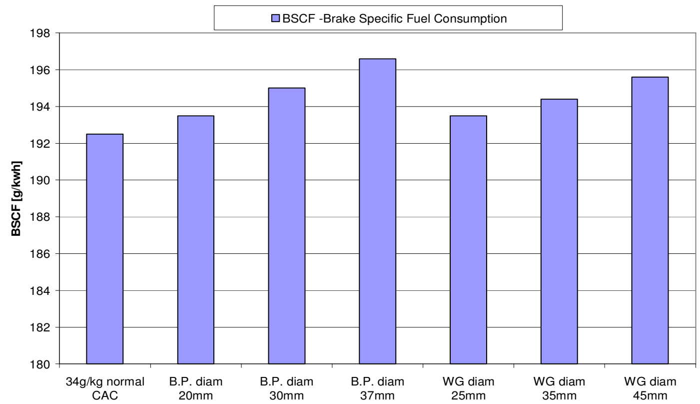


Figure 5.26: Brake Specific Fuel Consumption (BSFC) for the bypass and AWG cases

5.8 Fuel consumption for all cases with humidification

The purpose of the investigation of the fuel consumption was to see how the different humidification strategies effect the consumption and what will happen when the water amount is increased since different tests have shown both lower and higher consumption.

In Figure 5.27 is the fuel consumption for different humidity levels and temperatures shown and as one can see increased humidity gives lower fuel consumption at a given temperature (with “Wetpac H” setup) but increased temperature before compressor gives again higher fuel consumption.

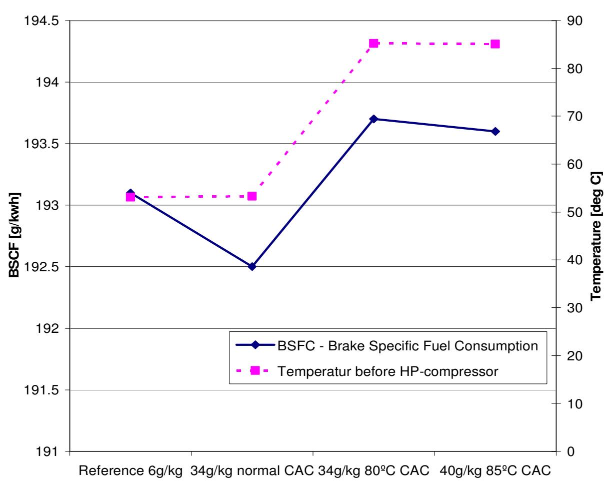


Figure 5.27: BSFC for different humidity levels and temperatures

Figure 5.28 shows the fuel consumption for all cases and one can see that the cases where emulsion is used the consumption increases. The steam injection has the lowest fuel consumption together with the humidity cases but the cases with modified HR has higher consumption than similar cases without changed HR due to the delayed combustion, as expected. Note that the scale in the Figure is chosen to visualize the difference and that the difference in fuel consumption is though small. The heat release change due to the emulsion is not taken into account but the change is likely to be quite small.

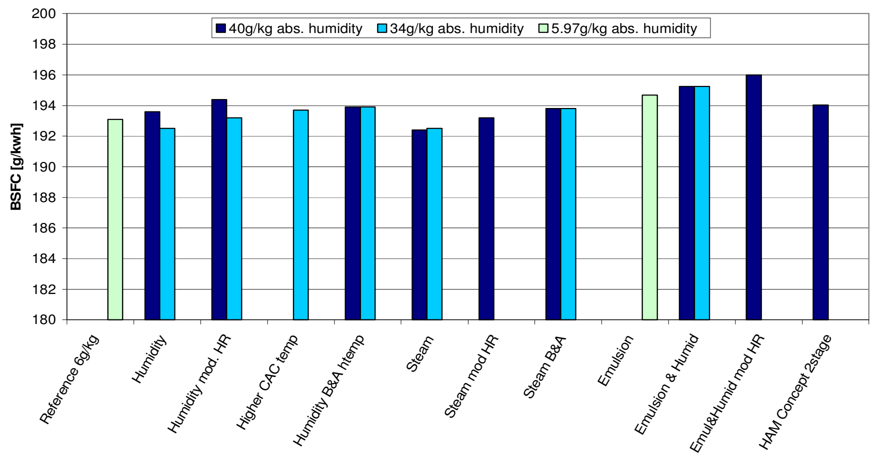


Figure 5.28: BSFC for all cases

The unexpected in the results above concerning the BSFC is that the HAM system has slightly higher consumption than the "humidity" cases since the reported consumption Figures from HAM have been lower than for a humidity system like Wetpac H. A likely reason for the increased consumption is the higher pressure drop over the HAM humidifier and the increased receiver temperature. But the difference in fuel consumption is though very small and is in the margin of error. The correlation can be seen in the humidity cases between high pressure and low consumption

5.9 Wetpac versus HAM concept on a 2-stage TC engine

A huge amount of water (10 times more than what evaporates) is injected into the hot charge air. It reduces the charge air temperature a lot and the CAC should not be necessary any more. But with the much lower humidity used in this case (40g H_2O/kg dry air) compared to what's used at Mariella it would be beneficial to have a CAC after the humidifier. Another solution could be to reduce the water injection temperature to bring down the temperature of the charge air but that could reduce the evaporation as well. The main temperature drop is caused by the evaporation but the huge amount of water that doesn't evaporate helps to cool the charge air before it is drained out and recirculated. In this case at 100% load the temperature reduction is about 5 ºC, see Figure 5.29, but at low load the 85 ºC injection water helps to heat the charge air which will increase the evaporation and therefore the NOx reduction will be higher at that load.

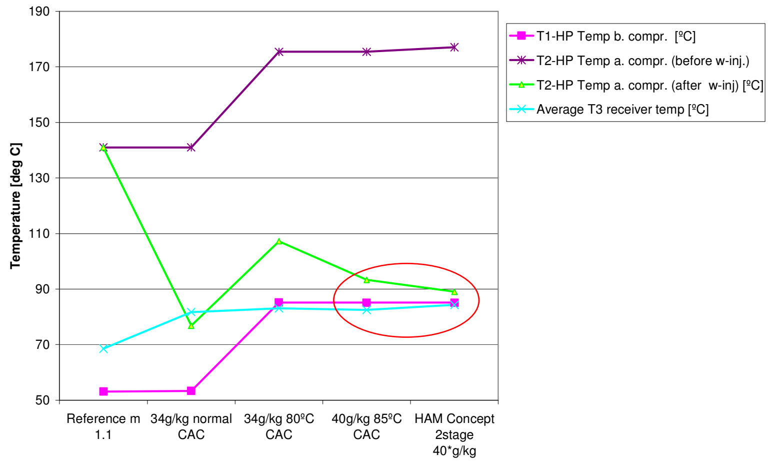


Figure 5.29: Temperature before and after the HP-compressor and also the temperature after the water injection, in the red circle you can see the temperature reduction due to the excess water heating and the increase of the receiver temperature

The receiver temperature is increased two to three degrees in the HAM case compared to the other cases since no HP-CAC is used and the temperature reduction from the excess water is not enough to keep the temperature at the same level as in CAC operation. The reason for that is that the injected water amount is less than in what is used on Mariella since lower humidity is needed (and possible). The maximum temperature is higher and the maximum pressure is lower in the HAM case than in the 40g/kg humidity case, see Figure 5.30 below. The temperature is higher due to increased receiver temperature and the maximum pressure is reduced as a consequence of the reduced boost pressure.

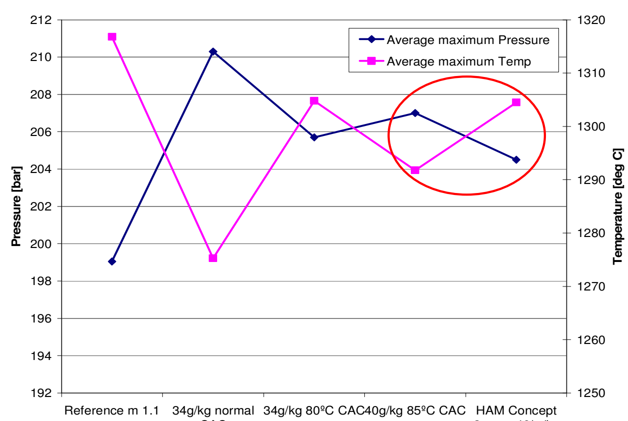


Figure 5.30: Maximum pressure and temperature, the temperature is higher and the pressure is lower for the HAM case than the 40 g/kg humidity case

One should also have in mind that the HAM system require a complete redesign of the intake system and the accuracy of the HAM simulation is probably affected by this (due to lack of data for the HAM vessel and assumptions about the evaporation) and it is also tricky to get the drain water amount correct in the simulations as well as make sure that no water is continuing in liquid form after the drain point into the cylinders.

5.10 Other interesting findings

Temperature before HP turbine

The temperature before turbine is slightly decreased in all cases (due to the increased heat capacity) except for the cases with injection before and after compressor but the temperature is well under the limit of about 550 ºC in all cases, see Figure 5.31. The differences in temperature between all the cases are though relatively small.

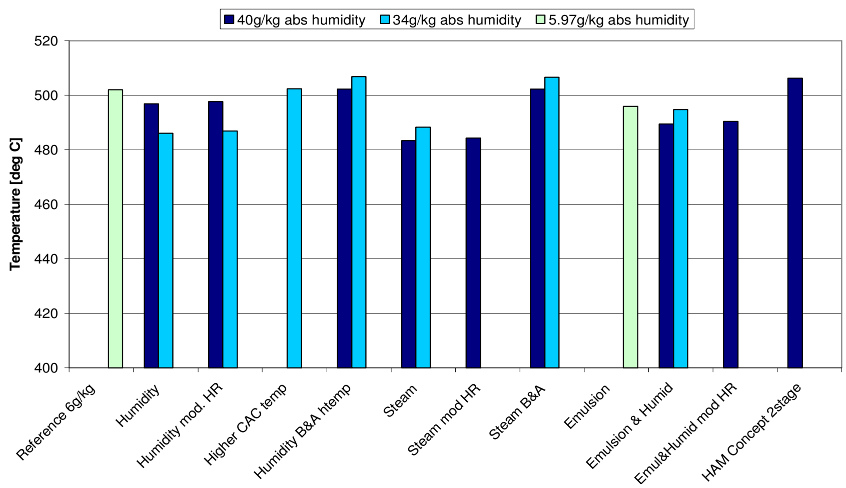


Figure 5.31: Temperature before HP-turbine

Receiver wall temperature measurement

A wall temperature measurement was carried out on the 20V32 in Vaskiluoto when it runs at 110% load. The wall temperature varies a lot at different locations in the receiver, from 63-83 ºC. At the coldest spots the temperatures of the flow and in the humidity cases the temperature of the flow will be increased. In that case the difference between the temperature in the engine room and the flow will increase which will give a higher cooling effect and can therefore cause condensation. This means that it is strongly recommended to have insulation of the receiver at least at the coldest spots to avoid condensation at the receiver walls, the higher temperature the better in this case.

5.11 Conclusions from the simulations

The more advanced cases/technologies investigated in this study is also the most promising for reducing the NOx emissions and also concerning evaporation issues, condensation risks etc. "Emulsion & Humidity" and the steam injection are the most promising cases in these aspects with the lowest maximum cylinder temperature and oxygen concentration. The cylinder temperature is reduced with about 50 ºC in the best cases. This is caused by the increased mass from the water injection (higher pressure) and the increased heat capacity. The receiver pressure increases with increased water to fuel ratio but this can partly be avoided by increasing the temperature before HP-compressor which is necessary in the 40g/kg case to avoid condensation. Increased humidity gives lower fuel consumption while increased temperature (before compressor) gives higher fuel consumption. The increased temperature and receiver pressure makes it necessary to change the diffuser ring on the LP-compressor and for the HP-compressor one can choose a compressor size which gives good performance both with and without humidification. This is a better solution than to use bypass or waste gate and the bypass should not be used at all since the LP-speed increases in this case and the margin to the speed limit is already small. AWG will be used on the test engine as a safety feature if surge will occur when elaborating with the humidity levels and the engine settings.

The emulsion & humidity was expected to be the best one due to the higher absolute humidity into the cylinders (46.6g H_2O/kg dry air) but the steam injection cases has surprisingly low cylinder temperature. For the steam injection case the problem with the evaporation is solved externally and will not cause any problems in the receiver and the risk for condensation is less than in the other cases (lower RH). The main drawback with both of these technologies is the increased complexity and thereby the high costs for the systems but another drawback in the emulsion & humidity case is the increased fuel consumption while the drawback in the steam injection case can be the high maximum pressure which can be too high. Finally the drawback for the "steam B&A" is the risk for damage of the HP-compressor if droplets will be formed before entering the compressor blades.

Another conclusion is that the Apparent Heat release is little effected when water is introduced to the inlet air, only a small delay have been noticed but note that the emulsion effects the heat release differently than the air humidity.

With the desired maximum relative humidity of 90% the lowest temperature allowed after CAC is 78 in the 34g/kg case and 82 in the 40g/kg case at 7.5 bar pressure which is the pressure in the reference case. A good HP CAC temperature is about 85 ºC. Even though the receiver temperature is increased 12-15 degrees compared to the reference the exhaust temperature is decreased or unaffected in most cases.

The receiver temperature is increased in the HAM case compared to the other cases since no HP-CAC is used and the temperature reduction from the excess water is not enough to keep the temperature at the same level as in CAC operation at full load with the 40g/kg humidity. The reason for that is that the injected water amount is less than in what is used on Mariella since lower humidity is needed (and possible). The maximum temperature is higher and the maximum pressure is lower in the HAM case than in the 40g/kg humidity case since the temperature is higher due to increased receiver temperature and the maximum pressure is reduced as a consequence of the reduced boost pressure. In the end the HAM system does not seem to be any better than the "normal" inlet air humidity in a NOx reduction perspective. The HAM case has also slightly higher fuel consumption but the main benefit with the HAM system is that sea water can be used.

6 Water evaporation and nozzle configuration

6.1 Water evaporation, what effects the evaporation

An estimation of the evaporation of a single droplet was carried out to see the effect on the evaporation time when changing relative humidity/absolute humidity, temperature, droplet size and pressure. The purpose of this estimation was not to get absolute time for the evaporation but instead the get a hint about the relative difference in time when each parameter was changed by a certain percentage from the reference value. The absolute evaporation time is very hard to estimate without CFD program where simulation of the interactions between the droplets, the surroundings and the three dimensional flow can be made. The CFD simulation result will be presented later in this chapter.

To estimate the relative difference in evaporation time manually for a single droplet some simplification was done to be able to calculate it manually. In the calculation the surrounding temperature was assumed to be 100 ºC which is somewhere between the start temperature when no water is introduced and the final condition when all the water is evaporated. A constant surrounding pressure is assumed in the calculation and at the droplet surface the wet bulb temperature is assumed which is likely to be the average condition but it depends on the injection water temperature and how well the droplet is ventilated etc. This was also done to limit the number of parameters and simplify the calculation. The equation used to estimate the evaporation time is equation (6.1) [29] where the partial pressure is calculated from the saturation pressure (6.2) [30] via the relative humidity (6.3) or the humidity ratio (6.4). The accuracy of the equation for the saturation pressure (6.2) is not so good for temperatures higher than 100 ºC therefore one should be a bit careful with the conclusions in these cases.

$$t = \frac{R\rho_p d_p^2}{8D_v M\left(\frac{p_d}{T_d} - \frac{p_\infty}{T_\infty}\right)} \tag{6.1}$$

Where:
- t - time [s]
- R - gas constant 8.315 [J/(mol K)]
- ρ_p - density of the liquid [g/m^3]
- d_p - droplet diameter [m]
- D_v - diffusion coefficient [m^2/s]
- M - molecular weight [g/mol]
- p - partial pressure of vapour [Pa]
- T - temperature [K]
- ∞ - conditions removed from the particle
- d - conditions at the droplet surface

$$p_{sat.} \approx a_0 + T(a_1 + T(a_2 + T(a_3 + T(a_4 + T(a_5 + T \cdot a_6))))) \tag{6.2}$$

Where:

a_0	6.107799961
a_1	$4.436518521 \cdot 10^{-1}$
a_2	$1.428945805 \cdot 10^{-2}$
a_3	$2.650648471 \cdot 10^{-4}$
a_4	$3.031240396 \cdot 10^{-6}$
a_5	$2.034080948 \cdot 10^{-8}$
a_6	$6.136820929 \cdot 10^{-11}$

$$p_{part.vapour} = RH \cdot P_{saturation} \tag{6.3}$$

$$p_{part.vapour} = \frac{x \cdot P_{tot}}{0.62198 + x} \tag{6.4}$$

The diffusion coefficient is also uncertain in the calculation and was estimated with equation (6.5) according to [28] but this gave a too small value according to [27] at normal atmospheric pressure and temperatures (should be 2.5*10^-6 at these conditions). This value should be compared with experimental data to get good accuracy.

$$D_v \approx 1.0956 \cdot 10^{-6} \cdot p^{-1} \cdot T^{1.76} \tag{6.5}$$

The droplet diameter can also be estimated with equation (6.6) [29] where ø is the Fuchs correction factor which has to do with the mass transfer by the diffusion but this coefficient can be ignored for particles larger than 2 µm.

$$\frac{d(d_p)}{dt} = \frac{4 D_v M}{\rho_d d_p R}\left(\frac{p_\infty}{T_\infty} - \frac{p_d}{T_d}\right)\phi \tag{6.6}$$

Figure 6.1 below shows the result from the calculation of the evaporation time and the values of the parameters the reference case is the following:

- 5 bar abs pressure
- 100 ºC average temperature removed from the droplet
- 50g H_2O/kg dry air absolute humidity
- 20 µm droplet size

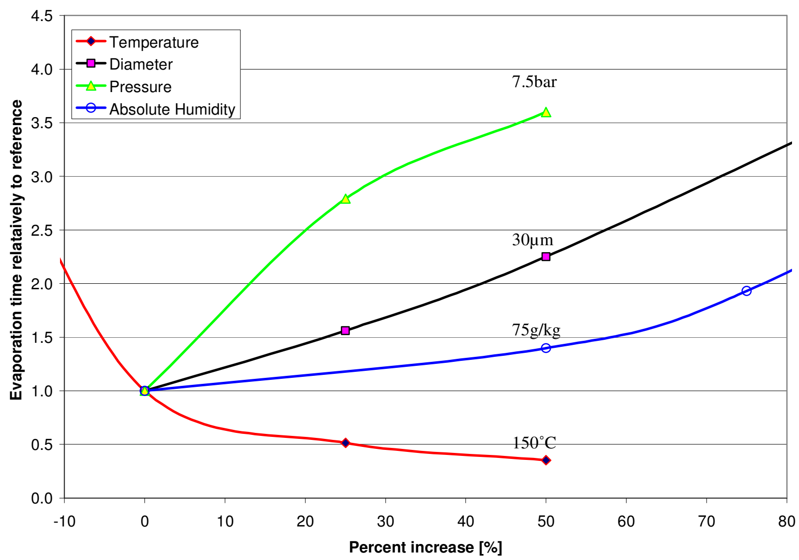


Figure 6.1: evaporation time for a droplet assuming average conditions when pressure, droplet size, absolute humidity and temperature is increased separately compared to the reference

In Figure 6.1 above, one can see that a relatively small reduction of the temperature have a very large impact on the evaporation time and the reason for that in this case is that we are close to the saturation point and the result would be a bit different with another start point but the reference were selected due to that it reflects the conditions after the TC on a modern heavy marine engine with humidification of the inlet air.
If the surrounding temperature reduces with given absolute humidity the difference between the actual temperature and the droplet temperature (wet bulb temperature) becomes less and less which also makes the difference in partial vapour pressure and saturation pressure becomes less and less since these pressures are functions of these temperatures. If instead the absolute humidity or the total pressure increases at a given temperature, the saturation level (=relative humidity) will increase since the partial vapour pressure increases but not the saturation pressure, the evaporation time will increase as shown in the Figure above. The large impact of increased pressure on the evaporation time is one of the reasons why 2-stage TC and humidification is so challenging as mentioned earlier.
The evaporation time is also seriously dependent on the droplet size, when the droplet size is doubled the evaporation time is four doubled and the droplet size itself is mainly dependent on injection pressure and nozzle design.

The air pressure and the absolute humidity is needed and can not be changed so the main variables left that can be changed are the droplet size, temperatures and another important thing, the flow field and the droplet distribution which effects the interactions between the droplets as well as the charge air and this will be presented in the next chapter.

6.2 CFD simulations

The CFD simulation was carried out by Alfred Herman Selvaraj with support by Lars-Ola Liavåg in the CFD group at Wärtsilä. They had already done a model for the flow distribution for the new 2-stage turbocharged engine. With the result from the GT-power simulations as input the CFD simulations was carried out with the water injection. The purpose for this CFD simulation is to do a quick check if it's seams possible to evaporate the amount of water needed to reach the desired absolute humidity level with the high pressure and relatively low temperature we have after the HP-compressor without big changes to the inlet system. The purpose is also to get a hint about the time it takes to evaporate the water and to get a Figure about the condensation level. The evaporation time is impotent since the distance from the injection point to the first cylinder is short, the water vapour reaches the first cylinder about 0.6 seconds after injection.

It is hard to find all the explanations to the entire phenomenon since it so many parameters effecting the evaporation and the result is not always univocal. The main explanations about how the evaporation will be under these circumstances are though described.

The time available for the CFD simulation was very limited and therefore simplifications had to be done mainly of the CAC and without the existing model for the "normal" 2-satge turbo charged engine this would not be possible at all with this time frame. The main simplifications/limitations of the model are the following:

- The CAC is very simplified in the CFD model
- The water circuit temperatures in the CAC is not correct in CFD model, the standard temperature is used (328-336K) instead of the higher temperature needed to avoid condensation (~353-358K) which was used in GT-power simulations
- The lower CAC temperature aggravate more condensation in the cooler section and lower the temperature downstream the CAC than it should with the right temperature but it is even though interesting to see what happens after the CAC
- This fault in temperature together with the very simplified model of the CAC makes the result only valid from the injection point to the inlet of the CAC
- The initial droplet size distribution from the nozzles used in the simulation is thought to correlate well enough compared with Danfoss specification but there is an uncertainty about at what distance from nozzle tip at which supplier has measured its specified distribution.

6.2.1 Simulation setup

In the first simulations the effect of nozzles configuration, charge air temperature and amount of injected water was evaluated. The setup for all the cases can be seen in table 6.1 and 6.2 below. In these cases the water injection pressure and temperature was kept constant at 100 bar and 40 ºC.

Table 6.1: the different nozzle configurations used in the first CFD simulations

Case	***Danfoss specification***	***Nozzle capacity (l/min @ 100 bar) & [size (mm)*]***	***Nozzles per bank***	***% Deviation from required flow rate (for abs. hum. 34 g/kg air)***	***% Deviation from required flow rate (for abs. hum. 40 g/kg air)***
C1	180Z1915	0.92 [0.35]	16 resp 20	-2.6	0.28
C2	180Z1910 + 180Z1920	0.42 [0.25] + 1.55 [0.50]	8 + 8 resp 10 + 10	4.3	6.8
C2 Large Nozzle	180Z1915 + 180Z1920	0.92 [0.35] + 1.55 [0.50]	8 + 8	-	7.69

Table 6.2: The setup for the first simulated cases with nozzle configuration, desired humidity and charge air temperature

Case	***Nozzle setup***	***Desired absolut humidity (g/kg)***	***Temperature after compressor (ºC)***	***Injected water amount (l/min)***
C1-H34-141	C1	34	141	29.44
C2-H34-141	C2	34	141	31.53
C1-H34-175.4	C1	34	175.4	29.44
C2-H34-175.4	C2	34	175.4	31.53
C1-H40-175.4	C1	40	175.4	36.80
C2-H40-175.4	C2	40	175.4	39.20
C2-H40-175.4	C2 largeNozzle	40	175.4	39.52

In the second simulation run the effect of water injection pressure, temperature and the amount of injected water (to reach the desired absolute humidity) is evaluated one by one while the charge air temperature and nozzle configuration were kept constant. The charge air temperature used in these simulations was 175.4 ºC and the nozzle used was only the Danfoss 180Z1910 which is the smallest nozzle they provide. The nozzles needed were placed in a number of rows with 12 nozzles in each row. Table 6.3 shows the case setup for the second simulation run.

Table 6.3: The setup for the second simulation run with water injection temperature, pressure and number of nozzles

Case	***Nozzle capacity* (l/min)***	***Injection humidity if 100% vap. (g/kg air)***	***% Deviation from required flow rate (for abs. hum. 40 g/kg air)***
T25-P100-N48**	0.42	40.32	0.8
T25-P150-N36*	0.5542	39.91	-0.22
T70-P150-N36	0.5542	39.91	-0.22
T25-P150-N48	0.5542	53.21	33.02
T85-P200-N36	0.6739	48.53	21.32
T85-P200-N48	0.6739	64.7	61.75

*** Nomenclature example T25-P100-N48:*
Injn. Droplet temperature, ºC
'P100'- Injn. Pressure, bar
'N48'- No. of 180z1910 nozzles per bank

* For injection pressures > 130 bars is the flow characteristic*'T25'-* extrapolated using:

$$Capacity\ (l/\text{min}) = -1.282 \cdot 10^{-6} \cdot P^2 + 0.002843 \cdot P + 0.01567$$

The water is injected radially directly after the HP-compressor outlet and from the injection points down streams to the CAC the simulation is frozen at three points (S1-S3), see Figure 6.2. The flow field with the flow velocities at the CAC inlet (section S3) can be seen in Figure 6.3. After section S3, from the CAC entry and down streams one should be careful with the conclusion from the simulation results since the model is not really accurate from that point. In the Figure you can also see the droplet sizes and the red spots are large droplet formations caused by the condensation.

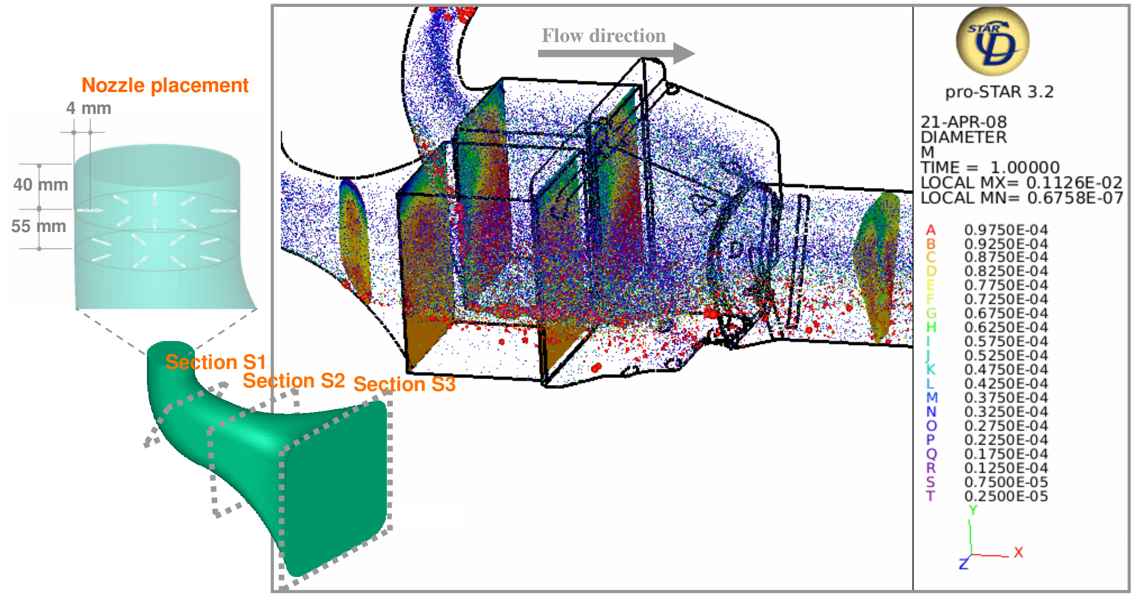


Figure 6.2: The analyzed sections with droplet distribution and droplet size in the inlet system from HP-compressor outlet to the receiver entry

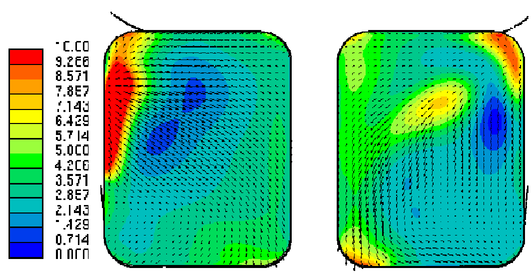


Figure 6.3: Normal velocity contours superposed on in-plane velocities near section S3

6.2.2 Effect of charge air temperature, c1(c2)-h34-141 Vs c1(c2)-h34-175.4

The charge air temperature turned out to be the most important parameter with the biggest influence on the evaporation. Comparing the cases with higher charge temperature, cases c1-h34-175.4 and c2-h34-175.4 show respectively ~22% and ~38% increased in vapour mass fraction entering cooler compared to c1-h34-141, which only yields 46.5% vapour mass fraction at section S3, 0.5s after injection. With the lower temperature, 141 ºC, one can see to the right in Figure 6.4 below that droplets stick together (the red bubbles). The obvious conclusion is that 141 ºC is too low even when aiming for 34g H_2O/kg dry air absolute humidity and with the higher temperature (175 ºC) some core richening can still be noticed to the right in the Figure. The increased temperature decreases the relative humidity and increases the difference between the actual air temperature and the temperature at the droplet surface which is sets the speed of the evaporation.

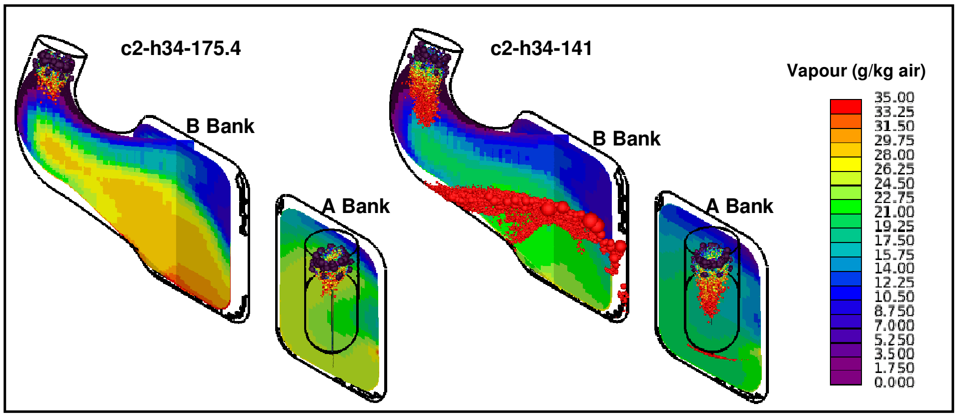


Figure 6.4: Vapour concentration and distribution for two cases with different charge air temperature, note that the max. vapour magnitude on the scale is clipped

6.2.3 Nozzle setup and droplet size sensitivity

The nozzle setup has a big influence on the vaporization and uniformity of the droplets. For the uniformity of the droplets a clear trend can be found but for the evaporation it is much harder to get a clear answer since there are so many factors that affect it. In this section these factors will be analysed and some reasons why they are effecting the evaporation will be explained.

The uniformity of the vapour and the deviation between the A- and B banks decreases with smaller droplets and when the Nozzles are of the same sizes. Figure 6.5 shows the uniformity index (UI) and Figure 6.6 the deviation between the banks of the vapour at section S3. Nozzle setup C1 gives consequently the best uniformity and the smallest deviation since the small droplets disperse better than large droplets.

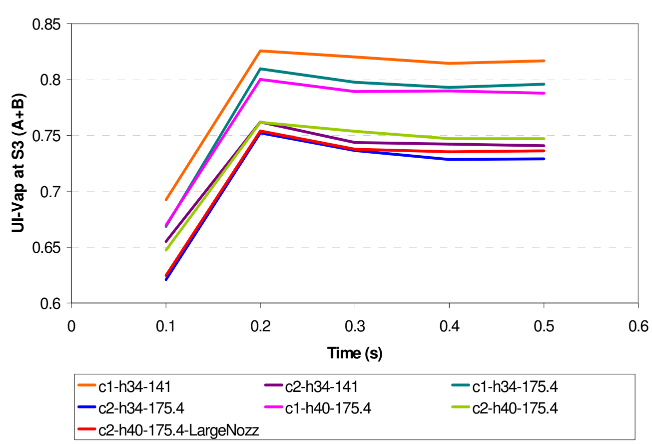


Figure 6.5: the vapour uniformity at section S3

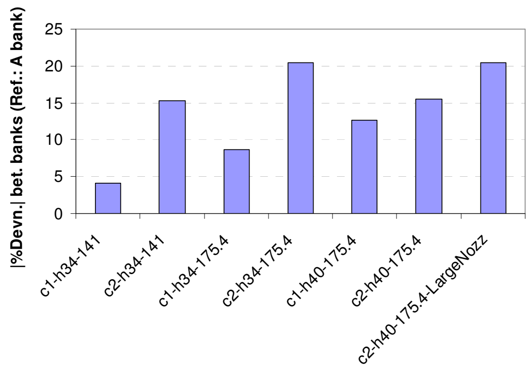


Figure 6.6: the deviation between the banks

The vapour yield for all nozzle setups with 175 ºC temperature (after HP-compressor) in the first simulation run can be seen in Figure 6.7. Until now the C1 nozzle configuration looks like the best case for the evaporation but as you can see in the Figure there is no univocal correlation between the uniformity (the c1 cases) and the vapour yield. The fluctuation in liquid mass depends on that water stick on to the surface and then flushes away but the vapour mass fraction entering cooler is stabilized in all cases by 0.5s after injection commencement.

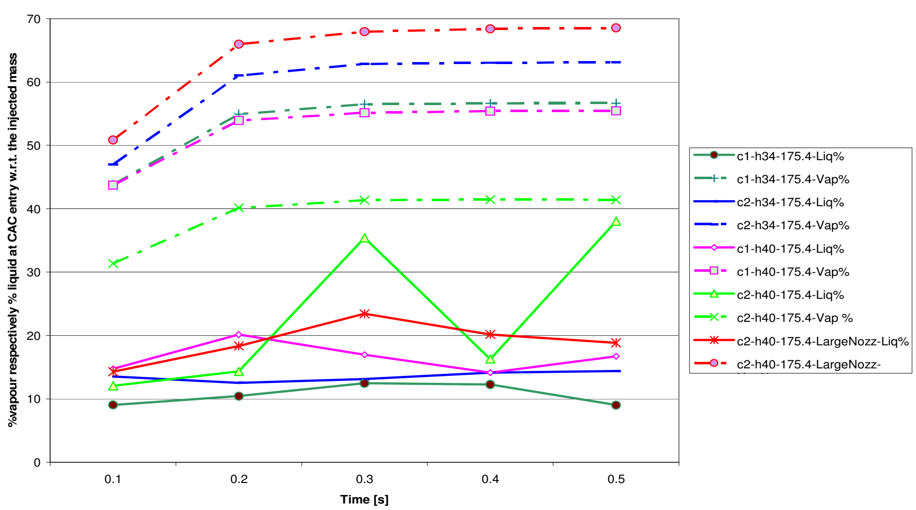


Figure 6.7: percent vapour and liquid entering the CAC with respect to the injected mass for the first 0.5s

When a smaller amount of water is injected, aiming for 34g H_2O/kg dry air humidity, the combination of the smallest nozzles (i.e. smaller droplets) and the largest nozzles in the c2 cases have good affect on the evaporation but when aiming for higher humidity it has the opposite effect. The average droplet temperatures are higher in c2-h40-175.4 at section S2 and S3 than for nozzle setup c2-h34-175.4 this is indicating that the droplets are less ventilated or too rich to be relatively better evaporated. To avoid this problem the c2-case with larger Nozzles was carried out and the vapor mass distribution for that case can be seen in Figure 6.8.

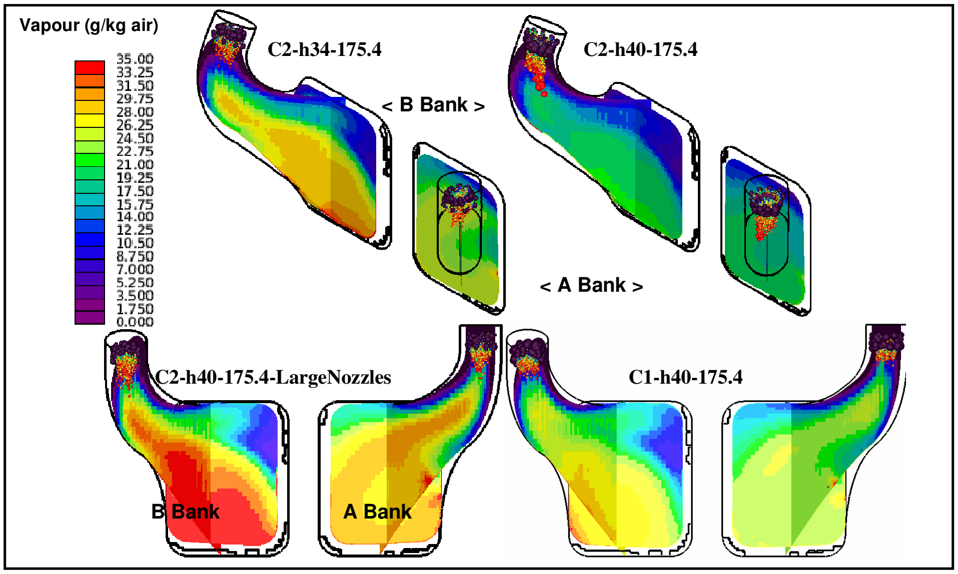


Figure 6.8: The vapour mass distribution for the c2 cases with 175.4 ºC temperature and the best C1 case, note that the maximum vapour magnitude on scale is clipped

The result was an improvement but still there are areas with very low vapour mass and also saturated areas with too much water. Inter-bank vapour deviation for c2-h40-175.4-LargeNozz has increased possibly due to differences in droplet residence time between banks because of the larger droplet masses. The bend of the pipe is not god for the vapour and droplet dispersion since the mass-forces makes a high water and vapour concentration at the bottom of the pipe which also can be seen in Figure 6.8 and in Figure 6.4. Together all this is indicating that the droplets are too large for further improvements of the vapour yield. The only solution left concerning the nozzle setup was to try to use only small nozzles (smaller than the c1) to get a better uniformity and thereby also reach the areas with low humidity, for example the hottest spots are in the upper part or the "roof" of the inlet to the CAC.

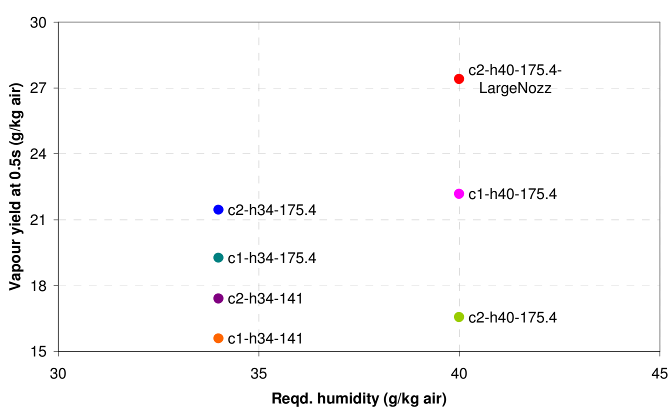


Figure 6.9: the absolute amount of vapour for all the cases in the first simulation run

In Figure 6.9 the reached absolute humidity for the first simulation run can be seen and the reached humidity is far from the desired humidity. Since the first simulation run didn't give enough understanding a second run of simulation was carried out. In this run a sensitivity test of smaller droplets, the effect of water injection pressure, injection water temperature as well as the effect of radically increased injection mass was simulated.

The sensitivity analysis was done to find out what the required injection droplet size is to achieve 40g H_2O/kg dry air absolute humidity. The water is injected with a uniform distribution and near to charge air flow velocities at the inlet plane which can be seen to the left in Figure 6.10. The sizes selected are 10 and 20 µm which is the smallest droplet sizes that is likely to be provided by a reasonably priced water injection system. This droplet sensitivity analysis showed that small injection droplet sizes lead to tremendous improvement in vapour uniformity between the banks compared to all previous test cases and the UI increased up to 95%. The vapour yield also increased especially for the 10 µm case droplet size with a maximum of about 83%, which also is the best efficiency among the simulated cases but the absolute amount of vapour is still under target. When comparing the 10 µm case with the 20 µm case in Figure 6.10 one can observe that the with the very small droplets the core can again starts to be enriched but remember that in this case droplets are injected over the whole cross section and this is why radial injection with injectors locating along duct walls is suggested.

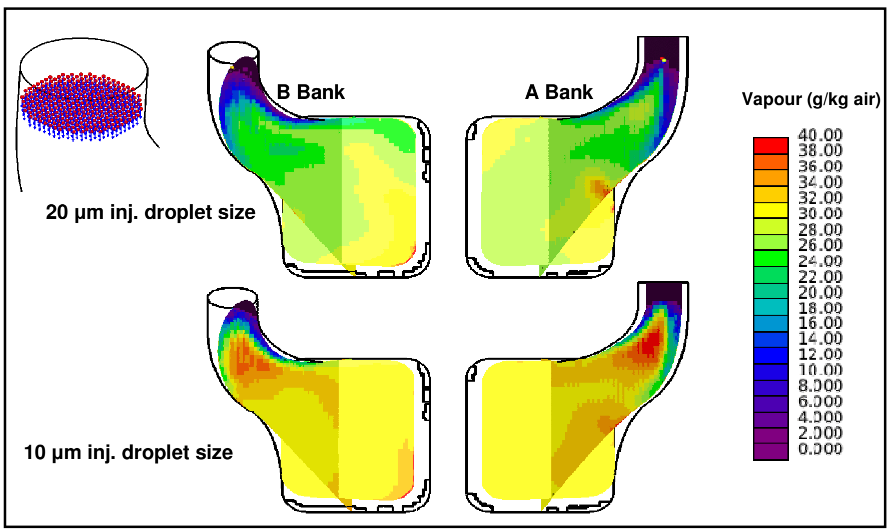


Figure 6.10: the effect of droplet size on the vapour mass distribution from the droplet sensitivity test, note that the maximum vapour magnitude on scale is clipped

Because of the good result from the sensitivity test with smaller droplet sizes the subsequent test was therefore simulated with all the nozzles of small and equal size (Danfoss 180Z1910 nozzles in c1 configuration). With smaller nozzles the number of nozzles of course had to be increased to get the same amount of water injected and more rows of nozzles are needed. This can be a design problem since the space available is very limited at the desired injection place after the HP-compressor. The benefits from the smaller droplets visualized in the sensitivity test above could also be seen when the increased number of smaller nozzles was used in the second test run, see Figure 6.12. Smaller droplet sizes of the order of those from the 180z1910 nozzle proved to be a high-scope parameter to increase the amount of vapour and the most important reason for that was found in the abrupt improvement in the deviation between the banks vapour uniformity. The total evaporation efficiency improved a lot, about 10 units at a given injection amount but the evaporation in the A-bank exceptionally improved compared to the first test run. When comparing Figure 6.11 with Figure 6.12 the avoided core richening, the more homogeny vapour concentration and the increased vapour yield (in most cases) can be noticed in the second simulation run with only small nozzles. In these cases the core of the field is quite dry in the beginning but at CAC entry the vapour distribution is very homogenous. The data for the cases can be seen in table 6.4.

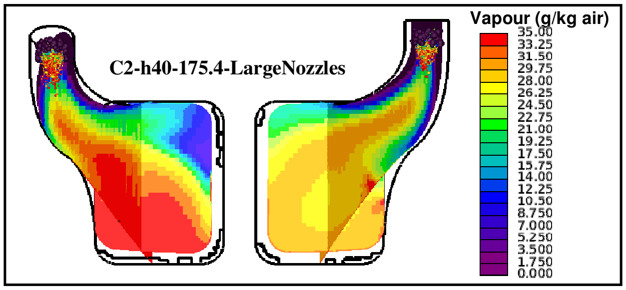


Figure 6.11: The vapour mass distribution for the c2-case with larger Nozzles from the first test run, note that the maximum vapour magnitude on scale is clipped

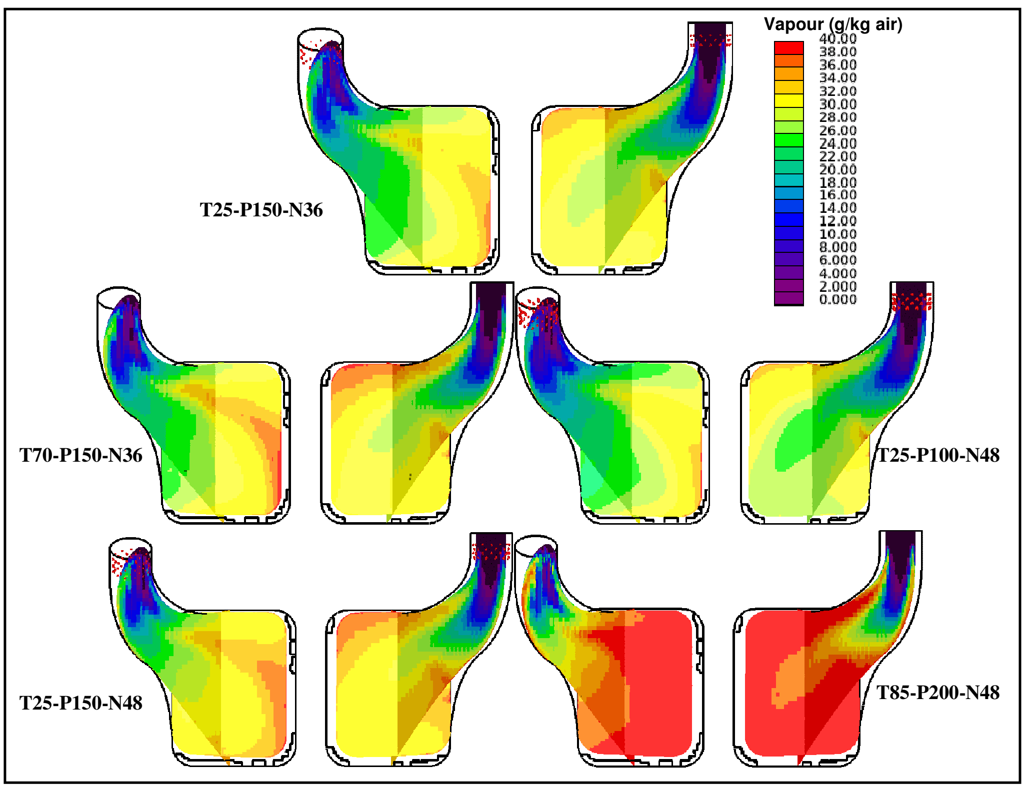


Figure 6.12: the effect of droplet size on the vapour mass distribution from the second simulation run (all small nozzles), note that the maximum vapour magnitude on scale is clipped

Table 6.4: The achieved efficiencies, deviation between the banks and the excess of injection water for the cases in Figure 6.12

Case	***Evaporation efficiency%***	***Vapour yield (% target) at S3 (A+B)***	***UI |% Deviation| between the banks (Ref. A Bank)***	***% more water inj. than required if 100% eff.***
20 μm droplets, uniform inj.	70.5	70.5	5.2	0
10 μm droplets, uniform inj.	82.9	82.9	2.7	0.0
T25-P150-N36	76.8	76.6	1.9	-0.22
T70-P150-N36	79.1	78.9	1.9	-0.22
T25-P100-N48	71.7	72.3	1.7	0.8
T25-P150-N48	65.7	87.5	0.8	33.0
T85-P200-N48	65.3	105.6	0.3	61.75
c1-h40-175.4	42.4	42.5	13.0	0.28
c2-h40-175.4	52.7	55.0	20.5	6.8
c2-h40-175.4-largeNozzle	63.6	68.8	20.5	7.69

6.2.4 The effect of water injection temperature

This effect is shown in Figure 6.12 (*T25-P150-N36 vs T70-P150-N36*), when the temperature of the injected water was increased from 25 to 70ºC the vapour yield increased by about 2.5%. The reason for the improved evaporation is due to the core of inlet duct being relatively less saturated and more droplets with higher droplet temperatures were observed. With a longer residence time (longer or wider pipe) the improvement increased temperature should most likely be even better since the conditions for continued evaporation is there.

6.2.5 The effect of water injection pressure

The effect of increased injection pressure from 100 bar to 150 bar and the injected water quantity kept almost unchanged by reducing the number of nozzles from 48 to 36 can be seen in Figure 6.12 (*T25-P100-N48 vs T25-P150-N36*). Vapour UI retained high magnitudes and the deviation between the banks stayed low and this was accomplished by the ability of the small droplets to reach the tighter geometric radius of the inner upper hot roof of the CAC entry. By increasing the injection pressure alone by 50 bar under these circumstances increased the vapour yield by 4.5%.

6.2.6 The effect of water injection mass

In Figure 6.12 one can compare case *T25-P150-N36 vs T25-P150-N48* and see the effect of increased injection mass. About 11% increase in vapour yield is predicted with a mass excess of 33% of what's needed for the target humidity if 100% would have taken place. But the increased vapour yield hence comes at the cost of about 11% decrease in absolute evaporation efficiency. The decreased efficiency is caused by more local richening which gives coalescence of droplets and a larger presence of droplet entities with larger sizes. The conclusion from this is that the evaporation efficiency decreases with increased mass and it is likely to be even worse with increased mass.

6.2.7 Residence time

In an attempt to find the possibility of improving the evaporation by extending the inlet duct up streams just by adding a 1m straight pipe to increase the residence time for the droplets and thereby improve the evaporation. The assumption was that this would not affect the flow field so that the affect of increased residence time could be evaluated. This assumption turned out to be wrong. Small changes in the velocity and pressure fields along the straight extension led to major differences in the flow field. The droplets carried away towards the walls by the radial momentum of the swirling inlet flow and which lead to film formation at the walls of the bend to the inlet to the CAC and due to this numerical instability were also encountered due to excessive wall wetting by water droplets in a few areas. This result may however serve as a good caution for the designers to not simply extend inlet pipe length dimension to achieve increased efficiency, since one also need to take the flow field characteristics into account. The only option left is to actually perform a re-optimisation of the air box geometry for use with Wetpac H.

6.2.8 Conclusions from the CFD results

According to the simulation it is possible to reach an absolute humidity of 40g H_2O/kg dry air but with a quite low evaporation efficiency of about 65%. In this case recirculation of the drain water is of course needed to decrease the water consumption. The increased charge air temperature before the HP-compressor to increase the outlet temperature from 140 to at least 175 ºC is absolutely necessary to achieve the desired humidity. To the left in Figure 6.13 below one can see the achieved humidity with respect to target, the excess of water needed and the injection water temperature and to the left in the Figure is the actual vapour efficiency displayed.

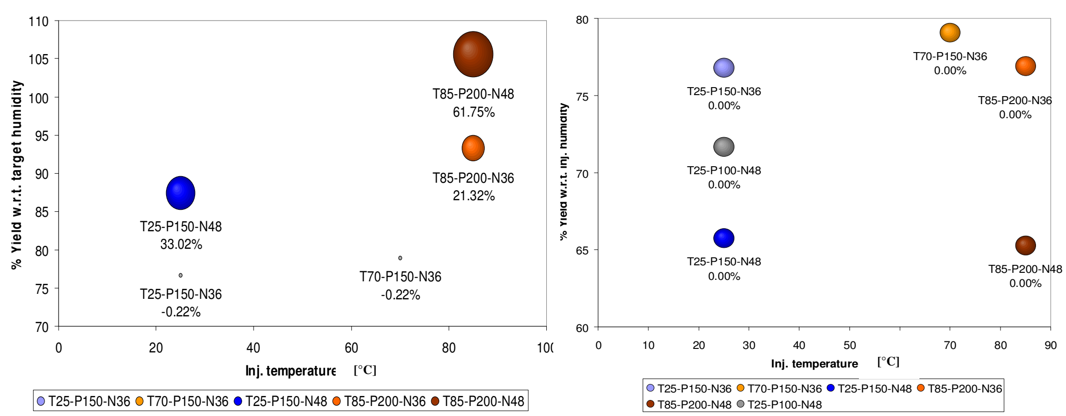


Figure 6.13: achieved humidity with respect to target, the excess of water needed and the injection water temperature. The size of the bubbles is symbolizing the total amount of water and to the right is the actual vapour efficiency displayed

Further improvement of vaporization is not easy without increasing the charge air temperature which will increase the fuel consumption and the thermal load. A redesign of the inlet for a more homogeneous flow and a longer residence time can improve the efficiency further. A redesign of the inlet system before the CAC entry for a more homogeneous flow and a longer residence time can be challenging since the space available is limited. This will also need extensive simulations of the flow but it might anyway be necessary to increase the efficiency. Increasing water pressure further to decrease the droplet sizes and improve the distribution is also challenging due to the high pressure (the needed strength and expensive equipments etc). The injection water temperature and pressure should though be as high as possible. A design that could increase the residence time is presented in the next chapter.

6.3 Alternative injection system and nozzles

6.3.1 Different nozzle setups and a CAC inlet for increased the residence time

Pointing directions of the nozzles

Earlier the pointing directions were against the flow but according to the CFD result it looks like the radial injection (90 deg angel) provides better droplet distribution. This should be investigated more but the CFD resources were too limited in this case to do that. It would be interesting to test for example 45 degrees against the flow or something in between.

Nozzle placement and a CAC entry for increased residence time

The design of the HP-CAC cooler is very compact and the reason for that is to make the engine as short as possible. This makes it hard to fit in all the nozzles needed for the water injection system and to get enough time for the evaporation. When the smallest nozzles were used (used in the second simulation run) three rows are needed to fit all the nozzles (36-48 nozzles). It can be a design problem to fit all the nozzles in the limited space and also to make sure that the strength is sufficient for 8 bars pressure. The strength of the pipe has to be ensured by the wall thickness, distance *A* in the Figure below. The dimension of the Danfoss nozzles can be seen in Figure 6.15. The inner diameter of the pipe is 230 mm and the distance between the rows with nozzles are:

- *B = 50mm (distance from the top of the pipe to first row)*
- *C = 20 mm (distance between the nozzle row)*
- *D = 20 mm(distance between the nozzle row)*
- *The distance from the last nozzle row to the bend start is about 14 mm*

To improve the evaporation and get more time for evaporation the flow distance can be increased. But in this case the design must be compact and this makes it hard to increase the flow distance. Another CAC entry than what was used in the simulation can be seen to the left in Figure 6.14. The alternative CAC entry design is made in an attempt to increase the residence time by directing the humidified air flow in an additional loop before entering the CAC. This additional loop gives about twice the residence time. But in addition to the residence time a homogeneous flow field is important and in this aspect it is hard to say how this will be with the new design without doing simulations of the flow. This and other solutions should be investigated further before finalising the Wetpac H concept for this specific engine.

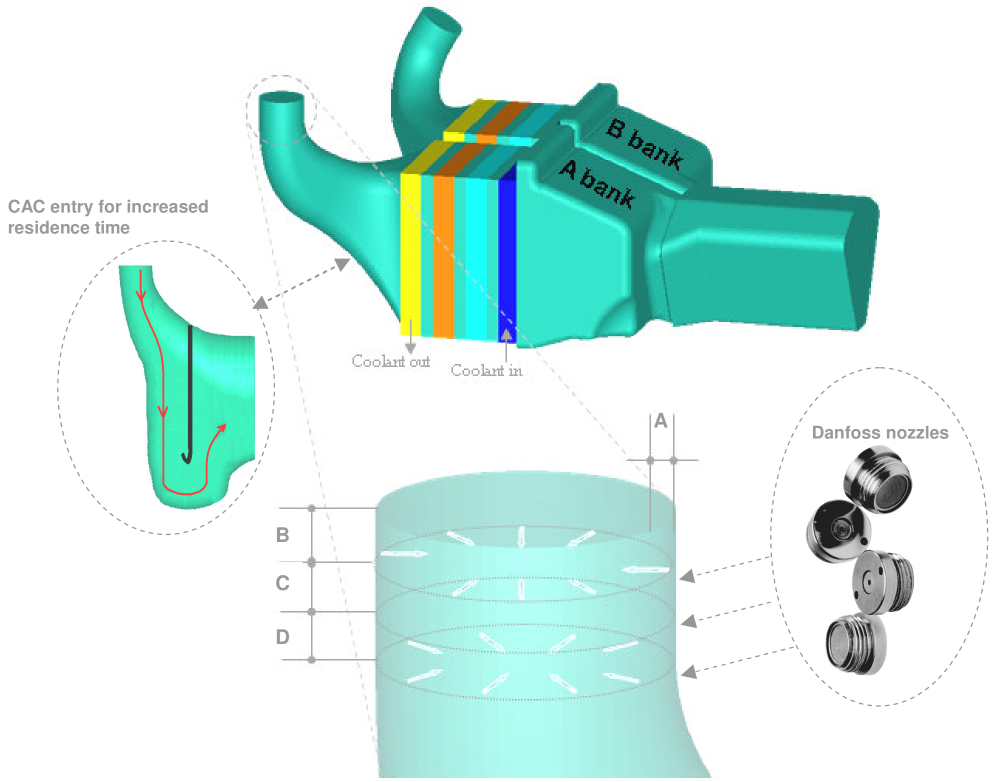


Figure 6.14: the standard CAC inlet and the CAC entry for increased residence time and the Nozzle placement used in the second simulation run

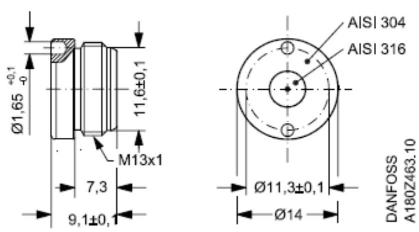


Figure 6.15: dimension of the Danfoss nozzles used in the simulation

6.3.2 Effervescent atomization nozzle

Effervescent atomization is an interesting technology which has the benefit of producing small droplets and at the same time only needs low pressure. A disadvantage might be the need of compressed air but on the other hand this is normally available in close connection to the engine (for start up etc). Effervescent atomization is a method to get atomization of two fluids by getting a small amount of gas into the liquid (bubbles) before it is ejected from the atomizer, in our case water and air. The atomizer basically consists of two orifices separated by an expansion chamber. The atomization starts at the inlet orifice with the pressure drop followed by the bubble growth into the liquid stream which starts to brake up inside the expansion chamber. Then the pressure drop and the velocity increase to a choked flow at the discharge orifice provides further atomization of the small droplets distributed at a wide cone angle (more or less depending on hole diameter). To get the desired cone angle and the desired droplet sizes requires optimization of the two orifices and the residence time (volume) in the expansion chamber. This type of spray provides truly remarkable small droplets, well below 50 µm for pressure difference less than 300 kPa. Compared to spray formation by mechanical means the droplets are more homogenous, the spray has wider cone angle and much shorter penetration length [32]. Furthermore, the amount of atomizing gas required is considerably less than what is employed in all other twin-fluid atomization techniques [33] and the pressure needed of both water and air is small. This kind of nozzles is for example used in jet engines.

One of few tests of effervescent atomization at higher pressures than atmospheric is made by Purdue University [34]. It was found that continuous increase in ambient air pressure above the normal atmospheric value causes the mean drop size to first increase up to a maximum value and then decline again. An explanation for this characteristic was derived in terms of the various contributing factors to the overall atomization process. It was also observed that mean drop sizes and drop size distributions are not that sensitive for nozzle discharge orifice diameter at different air pressures.

The main drawback with effervescent atomization nozzles is that compressed air is needed for the water injection which means that additional equipment is needed and this will probably lead to increased costs.

Another nozzle type used in gas turbines is the high fogging swirl nozzle from AxEnergy Ltd. The droplet distribution from a nozzle called AxE10 can be seen in Figure 6.16 [35].

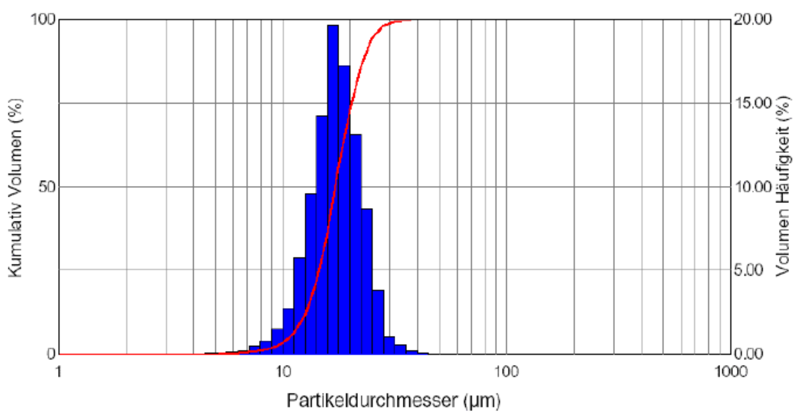


Figure 6.16: the droplet sizes are measured at a distance of 50mm with a wind speed of 4 m/s and a water pressure of 140 bar, the average droplet size is less than 20 µm

6.3.3 Other injection placements, Water injection close to the inlet valve

One solution for nozzle placement is to place the water injection nozzle close to the inlet valve like the fuel injectors placed in a fuel-injection system on a spark ignited engine and the injection takes place only while the inlet valve is open. A setup like this has been tested in several investigations with good NOx reduction results but very little are mentioned about the durability [36], [37]. This could enable increased volumetric efficiency by cooling the air into the cylinder and also in the cylinder during compression. In this case corrosion in the inlet system could be avoided but instead could it give corrosion in the cylinder head and in the combustion chamber. The biggest problem is though the risk for contamination of the lube oil if water droplets stick to the cylinder walls. The residence time is already too short for the evaporation under the investigated conditions but if one can tolerate that very small water droplets enters the cylinder like a fog it might be possible. In that case it could be a cost effective and relatively simple solution. It would be interesting to test it on an engine but it might cause wear on the cylinder as well as the piston and there is a large risk that the water droplets will end up in the lube oil. This is though dependent on how small droplets that could be provided from the nozzles.

7 Changes to the engine & test suggestions for the test in 2009

The necessary changes to the engine for the Wetpac H test in 2009 (on 20V32):

- Modification of the LT cooling circuit on both LP and HP CAC to be able to have at least 85 ºC temperatures in all CAC
- Change diffuser ring on LP-compressor if we will be able to test 40g H_2O/kg dry air absolute humidity (or use Air WG for that case)
- The HP-compressor should have the setup corresponding to mass multiplier 1.1 form the compressor in the model, i.e. 10% larger compressor (works fine with and without water injection)
- New pump unit which is able to provide 200 bar water pressure and the needed water amount is about 60 liter/min (depending on evaporation efficiency)
- Use as high injection water temperature as possible
- A new CAC inlet for both A and B bank with the water injection nozzles placed as suggested above
- Insulation of the inlet system from the point of water injection to the cylinders (or the hole inlet system) since the minimum temperature at any point down stream the water injection should not be less than 82 ºC when aiming for 40g H_2O/kg dry air humidity
- An air waste gate is maybe not necessary but can be nice to have if higher absolute humidity is required or if surge problem for some other reason occurs

Other suggestions for the test

- Increase temperature if the evaporation is not good enough
- Change the design of the inlet box to the CAC to improve the homogeneity of the flow field and increase the residence time if the evaporation efficiency is not sufficient
- Ability to also use the combination of emulsion and humidity if we do not reach the desired NOx-level with only humidification of the inlet air
- If possible, test effervescent atomization nozzles for comparison or other nozzles
- Test different injection angles

8 Recommendations for further tests and investigations

8.1.1 Test DWI with an additional injection during the cycle, DWI+LDWI

Today one injection is used during the cycle optimised for NOx reduction and to get the smallest increase of fuel consumption. But with an additional injection towards the end of the fuel injection (LDWI, see also chapter 2.5.2) the last injected fuel could be pushed from the area close to the nozzles with low oxygen concentration further out to areas with better conditions for burning the last fuel and reduce the slow "after burning". This can make the combustion duration shorter and reduce PM. In this case it is important that the water injection is directed to hit the fuel spray. This is quite easy to test and has a good potential to improve today's DWI system. It would also be interesting to inject a relatively small amount quite early in the compression stroke before the normal injection with the goal to try to achieve a bit reduced compression work ("wet compression"). This can be used to reduce the maximum pressure or enable earlier SOI but also to increase the NOx reduction further. To be able to realise this possibilities to adjust timing, duration and amount is necessary as well as possibilities for several injections during one cycle.

8.1.2 Build a test rig for evaluation of temperature, pressure and nozzle configuration and tests

With a relatively simple test-rig in scaled to for example ¼ of the real size it would be possible to perform very interesting and useful testing of the Wetpac H nozzle setup without dependence on test time on a real engine. This test-rig could provide data to validate the CFD simulations of the evaporation behaviour. It does not need to be complicated and can consist of:

- A pipe and depending on what to replicate some bends might be needed
- an electrical heater
- pressure sensor
- humidity sensor
- thermometer
- Compressed air from a normal air compressor
- Restriction valves (for both air and water)
- Water injection nozzles and water pump
- Malvern droplet size analyser (option)

The pressure from a normal air compressor could be adjusted by a restriction valve to the desired level and the flow can be adjusted with a restriction in the end of the test pipe. The temperature should be adjusted with the use of an electrical heater and a water pump with pressure adjustment possibilities is also needed to adjust the water pressure to desired level. The nozzles should be fitted to the pipe in a separate piece for easy removal and flexibility in the nozzle setup (different locations in relation to the pipe, different angles etc). A Malvern droplet analyser might not be that important since the reached absolute humidity at a given condition will reveal the nozzle performance anyway. But for a deeper investigation the suggested droplet measurement equipment could be rented from for example a University. An interesting droplet size measurement device and an effervescent atomizer nozzle are shown in Figure 8.1.

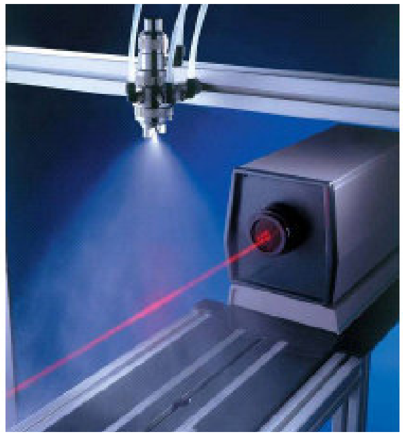


Figure 8.1: Malvern Analyzer (Malvern 2600) which is a laser measurement device for droplet size measurement, the picture shows a spray from effervescent atomization nozzle and the measurement device [32]

8.1.3 Develop a more sophisticated control system

Develop a more sophisticated control system with drain water control and salt content measurement to reduce the needed amount of water. This should be done to avoid that more water than what is evaporating by measuring the amount of drain water. Today the desired humidity or saturation point is the limiting factor for the control but it does not necessary mean that the saturation point or the desired humidity is reached even if enormous amount of water is injected. If the amount of drain water starts to be too high the injected amount should be decreased or the CAC temperature increased (if possible depending on the operating conditions) to decrease the drain water. The main factor limiting the NOx reduction today on Wetpac H is the CAC temperature used in the field. A lot higher temperature and NOx reduction have been achieved during testing. With a more advanced control system it should be possible to use higher CAC temperature and just reduce or increase temperature and injection amount depending on the actual operating condition or if an engine parameter is changed for some reasons. Today the calibration of the system has a lot of safety margin for worst condition but with the possibility to adjust the system for the actual condition this can be reduced and the NOx reduction increased in most cases. A more advanced control system could also enable the possibility to reduce NOx as much as possible depending on the ability of water or depending on other performance parameters (like SFOC, temperature and ambient humidity etc) depending on what the customer wants. Another benefit could be the possibility to provide different "NOx reduction modes" depending on where or in what application the engine will be used just by changing some parameter in the control system (Wetpac -20% or -40% etc).

8.1.4 Test instantaneous mixing of fuel and water and in combination with other wet-technologies

Instantaneous mixing of water and fuel is an attractive technology and should be tested to see how the performance will be in terms of emissions and fuel consumption. It would also be interesting to test instantaneous mixing of water and fuel (alternatively normal emulsion) in combination with humidification of the inlet air. The emulsion or instantaneous mixing of water and fuel is in that case used to cool the flame and give more kinetic energy of the spray which gives better reshuffle of the air and fuel. This could improve the combustion and reduce both NOx and PM. The humidification of the inlet air is used to dilute- and give more heat capacity to the air to reduce NOx even further. The increased flow over the cylinders and the increased TC effect is also beneficial (depending on maximum pressure and TC setup) for the fuel consumption and can hopefully compensate a bit for the increased fuel consumption normally achieved with emulsion. If instead instantaneous mixing of water and fuel is used in the combination the effect on the SFOC is unknown and is therefore interesting to discover.

9 Conclusions

The investigation of the CASS and Wetpac H system in use today showed that the NOx reduction limitation is the CAC temperature and the evaporation efficiency. Higher CAC temperature increases the possible absolute humidity and improves the evaporation but the CAC temperature can not be increased too much since the needed pressure ratio will increase and the increased flow will increase the pressure as well. This will cause TC surge at some point and of course the thermal load will at some point be a limitation as well. But according to the test the receiver temperature can be increased to 60-65 ºC (by increasing the CAC temperature) in some cases with the use of bypass which will enable higher NOx reduction than what is promised today since the CAC temperature is just slightly increased or unchanged in many cases today. Increased water injection pressure and temperature seams also to be beneficial. But there are definitely room for improvement of today's evaporation efficiency.

The investigation of the HAM system on Mariella showed that NOx emissions are reduced close to 70% in the E3 cycle while the PM emissions and THC is increased heavily on low load where the NOx reduction is highest. Additionally very few problems have been reported during the operation of the HAM units at Mariella. The key elements in the "mysteriously good" results with the HAM system compared to Wetpac H are mainly:

- "old fashion" low boosted engine
- much higher receiver temperature than what is used in Wetpac H
- much more water (pre heated) is used to get the desired amount of vapour which also requires a lot of waste heat
- ➢ The higher temperature and lower pressure together with the use of much more water (and pre-heated) enables the higher absolute humidity compared to Wetpac H and thereby higher NOx reduction

From the GT-Power simulation study it was concluded that the more advanced cases/technologies investigated are also the most promising for reducing the NOx emissions and also concerning evaporation issues, condensation risks etc. The NOx reduction effect with the introduction of water (or vapour) is achieved by reducing the local maximum combustion temperatures in the combustion chamber as mentioned earlier but also by reducing the concentration of oxygen by the addition of inert media with high specific heat (vapour). NOx formation is mainly dependent on the combustion temperature but also the availability of oxygen and by reducing the temperature and/or oxygen concentration the NOx emissions will therefore be reduced. The reason for the reduced temperature is the increased heat capacity and the increased mass from the added water.

"Emulsion & Humidity" and the steam injection are the most promising cases in these aspects with the lowest maximum cylinder temperature as well as the lowest oxygen concentration. The emulsion & humidity was expected to have the best temperature reduction due to the higher absolute humidity into the cylinders (46.6g H_2O/kg dry air due to the water addition in the fuel) compared to the other. The simulation of the HAM

system on the 2-stage TC engine does not seem to be any better than the air humidity in a NOx reduction perspective. The HAM case has also slightly higher fuel consumption but the main benefit with the HAM system is of course that sea water can be used.
Another conclusion is that the Apparent Heat release is little affected when water is introduced to the inlet air, only a small delay has been noticed but note that the emulsion affects the heat release differently than the air humidity.

The receiver pressure increases with increased water to fuel ratio but this can partly be avoided by increasing the temperature before HP-compressor which is necessary in the 40g H_2O/kg dry air case to avoid condensation. Increased humidity gives lower fuel consumption while increased temperature (before compressor) gives higher fuel consumption. The increased temperature and receiver pressure makes it necessary to change the diffuser ring on the LP-compressor and for the HP-compressor one can choose a compressor size which gives good performance both with and without humidification. This is a better solution than to use bypass or waste gate and the bypass should not be used at all since the LP-speed increases in this case and the margin to the speed limit is already small.

In a comparison between all the wet-technologies evaluated in the study, the Emulsion has the best overall result and between the other it is quite equal but “humidification with steam” is the wet-technology with the worst over all properties.

According to the CFD simulation it is possible to reach an absolute humidity of 40g H_2O/kg dry air but with a quite low efficiency of about 65%. In this case recirculation of the drain water is of course needed to decrease the water consumption. The increased charge air temperature before the HP-compressor to increase the outlet temperature from 140 to at least 175 ºC is absolutely necessary to achieve the desired humidity. This requires the possibility to adjust the temperature in the LP-CAC to a higher level, up to 85 deg C. The HP-CAC also needs to be adjusted to avoid that the saturation point is reached in the receiver and a suitable temperature is in this case also around 85 deg C. But it might be needed to adjust it at different loads due to the thermal load or other performance issues.
Further improvement of vaporization is not easy without increasing the charge air temperature which will increase the fuel consumption and the thermal load. A redesign of the inlet system for a more homogeneous flow and a longer residence time can improve the situation. But this can be challenging because the space available is limited and this will also need extensive simulations of the flow and the nozzle setup. Increasing water pressure further to decrease the droplet sizes and improve the distribution will be challenging due to the high pressure (the needed strength and expensive equipments etc). According to the simulation is 200 bars water pressure needed and the water injection temperature should be as high as possible.

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Appendix A, ISO 8178 test modes and weighting factors

D1: Power plants

Speed	Rated speed		
Load [%]	100	75	50
w. factor	0.3	0.5	0.2

D2: Marine installations gensets with intermittent load

Speed	Rated speed				
Load [%]	100	75	50	25	10
W. factor	0.05	0.25	0.3	0.3	0.1

E2: Marine installations, constant speed and controlled pitch

Speed	Rated speed			
Load [%]	100	75	50	25
W. factor	0.2	0.5	0.15	0.15

E2: Marine installations, fixed pitch

Speed [%]	100	91	80	63
Load [%]	100	75	50	25
W. factor	0.2	0.5	0.15	0.15