Efficiency Optimization of Biomass CHP Gas Engines

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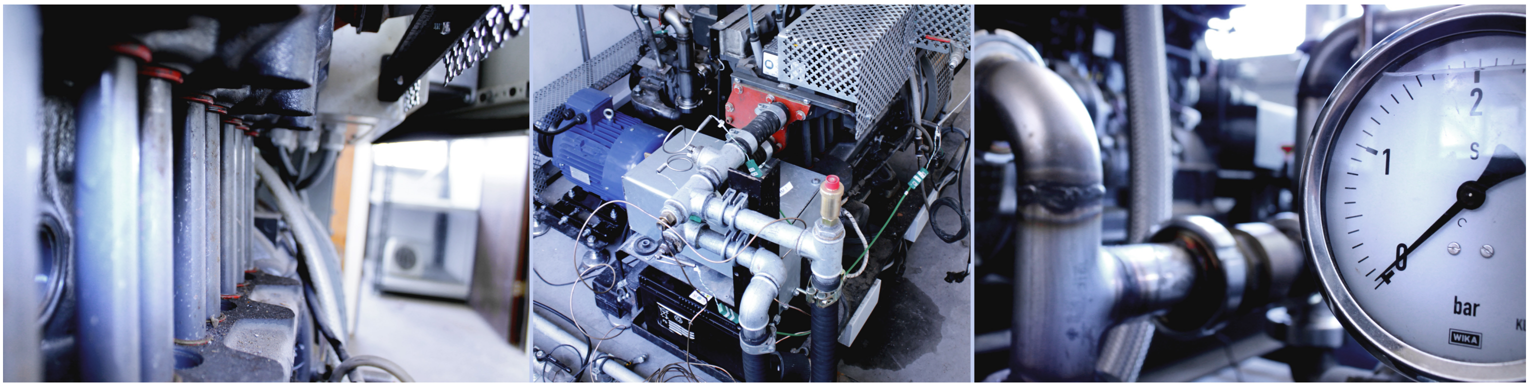

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Summary

Energy from biomass can to a large extent directly substitute fossil fuels and is therefore a very cost-effective way to reduce carbon emissions. As central biofuels, biogas and product gas from thermal gasification can be produced from a very wide range of biological resources by proven technologies. These gaseous fuels can be directly used in current natural gas engines, with combined heat and power generation. Natural gas engines are traditionally operated at lean conditions to obtain high efficiencies and low levels of NO_x emissions. Product gases have a distinct disadvantage to natural gas, as their heating value is 8 times lower, which results in reduced power. However, as these fuels contain a large inert fraction and low levels of hydro carbons, the heating value of the stoichiometric gas-air mix is only reduced by 24%. The large inert fraction also makes the fuels more resistant to knocking and reduces peak temperatures, which makes them applicable for operation near stoichiometric conditions, pressure charging and higher compression ratios.

Near stoichiometric, the engine has a higher power density, but emissions will increase. Hence a three-way catalytic converter (TWC) is applied and emissions of CO, NO_x and unburned hydrocarbons (UHC) are measured before and after. A Lister Petter natural gas engine setup with four-cylinders, 1.86 L, and spark ignition was operated with synthetic biogas, TwoStage product gas and LT-CFB product gas. Biofuels were tested for performance and emission levels, by varying the excess air-fuel ratio (λ), ignition timing and boost pressure.

Biogas showed the highest power output, with 20 - 30% higher than for the product gases. Across tests, the power increased 20 – 25% by lowering λ. The electric efficiency of the engine varied between 24 – 33% for the product gases, showing high values at low λ. Thus there is a clear incitement to operate product gas engines close to stoichiometric condition: increased power and efficiency. Pressure charging showed up to 80 – 110% increased power at 900 mbar boost pressure, but NO_x emissions increased significantly. Studying the ignition timing, it was seen that biogas had an optimum between 20 – 22 crank angle degrees before top dead center (CADBTDC), while the TwoStage gas was 5 – 12 CADBTDC. This is an interesting result, as product gas timing can be retarded for optimization of both power and emissions.

NO_x emissions are mainly controlled by the combustion temperature. The effect of λ was limited on emissions at slightly lean conditions, but affected emissions at stoichiometry and very lean combustion. The LT-CFB, with a high inert content of 70%, displayed excellent NO_x control with values below 100 mg/nm^3 across λ-values. The ignition timing showed to be effective in reducing NO_x, especially for the TwoStage gas that reached sufficiently low levels at λ=1.03 with optimum ignition timing. The biogas could only reach NO_x through either lean or sub-stoichiometric operation with the TWC.

CO is mainly controlled by the λ. For stoichiometric and very lean conditions the emissions are high – especially for product gases that have high levels of fuel-CO. However, only at very low and rich λ-values was the CO level seen to be above the favourable regulation before the TWC. The catalyst showed excellent ability to convert CO in the exhaust, reaching conversion of 98 - 100% across tests and gases. Applied for sub-stoichiometric biogas operation, the TWC reduced emissions below regulation.

The simple and mass produced TWC proved highly effective for reducing CO emissions and can applied to existing plants to reduce levels significantly. The TWC was ineffective in reducing UHC and NO_x, but did however reduce NO_x and CO below regulation for sub-stoichiometric biogas operation.

Dansk resume

Energi fra biomasse kan i stor stil substituere fossile brændsler direkte i den nuværende fossilbaserede infrastruktur og er derfor associeret med relative lave omkostninger. Biogas og produktgas fra forgasning er centrale biobrændsler, der kan produceres ud fra en bred vifte af bioressourcer via kendte teknologier. Disse gasbrændsler kan direkte erstatte naturgas i nuværende gasmotorer for kraftvarmeproduktion. Drift med naturgas er ofte med mager forbrænding, for at hæve virkningsgraden og mindske NO_x-udledningen. Sammenlignet med naturgas, har produktgas en meget lav brændværdi på kun 1/8, hvilket vil mindske elproduktionen. Men fordi at gasserne indeholder store mængder inert gas og kun lidt kulbrinter, er denne værdi kun 24% for en støkiometrisk brændstof-luftblanding. Andelen af inert giver brændslerne højere tolerance for bankning og reducerer forbrændingstemperaturen, hvilket gør dem egnede til drift ved støkiometriske forhold, trykladning og højere kompressionsforhold.

Tæt ved støkiometrisk har motoren en højere effekt, men emissionerne er tilsvarende højere. Derfor er en 3-vejs katalysator (TWC) monteret og CO, NO_x og uforbrændte kulbrinter (UHC) emissioner er målt før og efter. En Lister-Petter naturgasmotor med 4 cylindre, 1,86L og gnisttænding blev sat i drift med biogas og produktgas fra TwoStage og LT-CFB forgasseren. Biobrændslerne blev testet for effekt, virkningsgrad og emissioner ved at variere luftoverskudskoefficienten (λ), tændingstidspunktet og trykladningen.

Den højeste effekt blev opnået med biogas, der viste 20 – 30% højere værdier end produktgasserne. På tværs af brændslerne steg effekten med op til 25% i takt med at λ blev mindsket. Elvirkningsgraden for LT-CFB gas blev målt til 24 – 33%, højest ved lave λ-værdier. Der er derfor et klart incitament for drift nær støkiometriske betingelser. Trykladning resulterede i op til 80 – 110% effektforøgelse ved 900 mbar, men NO_x emissionerne steg kraftigt som følge af tryksætningen. Ved at justere tændingstidspunktet blev det påvist, at biogas optimalt set skal antændes mellem 20 – 22 grader krumtapaksel før toppen af cylinderen, imens TwoStage produktgas skal tændes mellem 5 – 12 grader før. Dette er interessant for produktgassen, da en forsinkelse af tændingen kan optimere både effekten og emissionerne.

NO_x emissioner er primært kontrolleret at forbrændingstemperaturen. Effekten af λ-værdi var relativt lille ved mager drift, men betydelig ved støkiometrisk og meget mager forbrænding. LT-CFB gassen består af 70% inert, og producerede meget lave NO_x emissioner under 100 mg/nm^3 på tværs af λ-værdier. Tændingstidspunktet viste sig effektiv til at mindske emissioner under grænseværdierne, særligt for TwoStage-gassen, hvor optimal tænding muliggjorde drift ved λ=1,03. Biogas nåede kun under NO_x emissionsgrænsen ved meget mager og understøkiometrisk forbrænding med TWC'en.

CO udledningen er primært kontrolleret af λ. Ved støkiometrisk og meget mager forbrænding er udledningen høj – særligt for produktgas, der har en stor andel brændsels-CO. Ved disse betingelser oversteg udledningen grænseværdierne. Katalysatoren viste meget høj reduktion af CO niveauet på 98 - 100% på tværs af gasser og λ-værdier. Ved understøkiometrisk drift med biogas, kunne TWC'en reducere emissionsniveauet under grænseværdien.

TWC er en simpel og masseproduceret katalysator, der er påvist at kunne fjerne store mængder CO og kan relativt nemt installeres på nuværende motoranlæg. TWC'en var ineffektiv til at fjerne NO_x og UHC, men kunne reducere niveauet af både CO og NO_x ved understøkiometrisk for biogas til under grænseværdierne.

Introduction

There is a need to replace fossil fuel-based energy with a carbon neutral energy system. As a nearly carbon-neutral fuel, bioenergy can to a large extent directly substitute fossil fuels in the current heat and power infrastructure, making it a very cost-effective way to reduce CO_2 emissions. Because biomass is a limited resource and has a lower calorific value than fossil fuels, the thermal efficiency is of high importance, with regards to both heat and electricity.

Combined heat and power (CHP) generation is currently done at: large steam boiler plants with high electrical efficiencies and lower total (CHP) efficiencies; and decentralized natural gas engine CHPs with lower electric efficiencies and high total efficiencies. Emissions between the scales also differ, as large plants usually have lower emissions due to more complexity and equipment, and smaller plants have to rely on simple equipment and process optimization. Because of these reasons, it is interesting to investigate utilization of gaseous biofuels and process optimization of natural gas engines with simple cleaning equipment.

Organisation

This chapter includes a brief overview of the project partners, the technological background for the project and the purpose of the project. The organization is presented in Table 1.

	Enterprise/Institution	Resource Persons	e-mail
Responsible Entity	DTU Chemical Engineering	Jesper Ahrenfeldt	jeah@kt.dtu.dk
Partner 1	Dansk Gasteknisk Center	Torben Kvist Jensen	tkj@dgc.dk

Table 1 – Partners and sub-contractors in the project

Background

Current gas engine technology is developed for the existing natural gas infrastructure. Natural gas engines are spark ignition engines and are usually operated with: high compression ratio and turbocharging to increase efficiency and power density; and lean burn (excess air combustion) operation to minimize knocking, reduce NO_x emissions and increase efficiency. The main drawback of lean burn operation is that there will be increased emissions of unburned fuel in the engine exhaust due to lower temperatures, and that the power density is lower than for operation closer to stoichiometric conditions.

An alternative approach can be taken in order to reduce emissions to a minimum and obtain high power outputs. If the engine is operated near stoichiometric conditions and a three-way catalytic converter (TWC) is applied on the exhaust, emissions can be reduced effectively while high power is obtained. TWC is a commercial technology that is widely applied in the automotive industry. However, the technology has not yet been applied to gaseous biofuels in engines, except for single research studies with biogas. Near stoichiometric operation will increase NO_x emissions due to higher temperatures, but lower the unburned fuel emissions, due to effective combustion. At stoichiometric conditions, the unburned fuel emissions will however increase due to oxygen starvation, but NO_x will decrease. Using low excess air in the exhaust, allows the TWC to reduce both unburned fuel and NO_x emissions simultaneously and hence reduce emissions below traditional lean burn operation.

This approach does however have limitations with regards to turbocharging and compression ratios because of engine knocking, as the combustion temperature will increase closer to stoichiometric operation. Also, only spark-ignition engines are possible for this operation. Decreases in turbocharging and compression ratio will lead to lower power densities and lower efficiencies. However, research has shown that gaseous biofuels such as biogas and product gas have increased resistance to knocking, because large shares of inert components and lower contents of higher hydrocarbons. Product gas has also shown that operation near stoichiometric conditions does not necessarily reduce engine efficiency, as natural gas does - on the contrary the operation seems to be increasingly efficient with regards to power and efficiency[1]. Thus there is a clear incitement to operate near stoichiometric for product gases.

Gaseous biofuels as biogas from anaerobic digestion and product gas from thermal gasification are significantly different from natural gas. Biogas consists of CH_4 and CO_2 and has a relatively high calorific value, while having a significant share of inert gas. Product gas is a mixture of several gases including N_2, H_2, CO_2, CO and CH_4 with a relatively low calorific value and a high share of inert gas. These biofuels have some advantages compared to natural gas in engine operation. The relatively high amount of inert (CO_2, N_2) and lower hydrocarbon content in the gasses, reduce NO_x emissions significantly, due to a higher heat capacity and dilution effects, that lowers the maximum combustion temperature and reduces knocking in the engine. The fuels also have lower amounts of hydrocarbons, compared to natural gas that also contain higher compounds such as C_3H_8. These higher compounds are usually very flammable and can cause knocking due to auto ignition.

Because of these fuel characteristics, it is interesting to test gas engines with lower excess air ratios, approaching stoichiometric operation to increase the electrical efficiency and increase power. However, this operation will cause an increased amount of emissions in the exhaust. Unburned CO and CH_4 will increase due to a slightly incomplete combustion and NO_x emissions increase due to high temperatures. Research has also shown that adjusting the ignition timing can reduce emissions and increase power and efficiency for product gas[2].

Biogas and product gas production technologies

This section will present a brief description of the technologies that are used for producing the gasses used in this project: anaerobic digestion and thermal gasification.

Biogas is produced from biologically converted biomass through anaerobic digestion processes. The technology is commercial and has been widely applied for heat and power production using gas engines. The technology is characterised by having relatively high feedstock flexibility and gas quality. The process utilizes microorganisms to degrade sugars, fats and other biodegradable compounds into approximately 70% CH_4 and 30% CO_2. Anaerobic digestion can process a variety of biomasses and wastes and is mostly used for agricultural wastes (especially manure), industrial wastes from food industry (slaughter waste, sugary wastes) and municipal waste water treatment. However, lignin and non-biological compounds are not convertible. The process is also of interest as the residual sludge from the process usually has a high fertilizing value.

Product gas from thermal biomass gasification is produced by converting solid carbonaceous feedstock into gaseous fuels at high temperatures. The technology is early commercial, with several full-scale and pilot plants in operation. Most gasification technologies partly oxidizes the fuel to reach temperatures between

700-1300°C and hereby decompose the fuel into a mixture of gaseous components called product gas. The process can because of this characteristic convert most feedstock ranging from green and woody biomasses to industrial wastes as municipal waste and sewage sludge. The process can be highly efficient, converting the chemical energy in biomasses to gaseous fuel at values above 90%. The produced gas is usually low-calorific with a gas composition of approximately 40-70% inert N_2 CO_2, along with CO, H_2 and CH_4. The main drawback of most gasification systems is the low gas quality, where higher organic tars can pose a challenge. Tars condensate at high temperatures and can clog and corrode systems including engines. The gas thus has to be completely clean of these compounds.

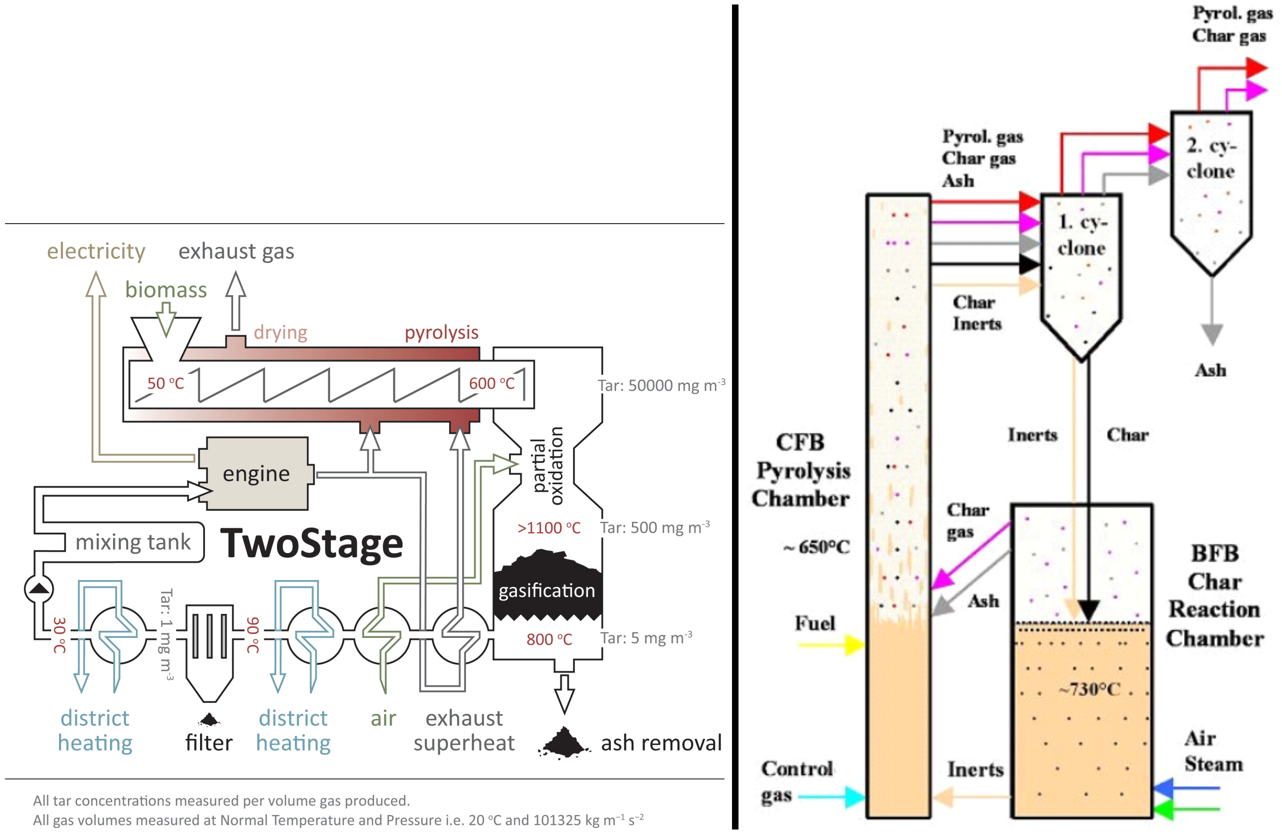

Figure 1 – Flow chart of the TwoStage Viking gasifier (left) and Low-temperature circulating fluid bed (LT-CFB) gasifier (right).

The two utilized product gases in this project is from the TwoStage (also called Viking) and Low-temperature circulating fluid bed (LT-CFB or Pyroneer) gasifiers located at the Technical University of Denmark. The two technologies are shown on flow diagrams in Figure 1. The TwoStage gasifier processes wood chips at above 1100°C and are characterised by high efficiency of 93% and excellent gas quality without tars. Using separate pyrolysis and gasification stages with a partial oxidation in between, the gasifier is able to reduce tars internally without any gas cleaning but a simple bag filter. The TwoStage gasifier has been built commercially up to 2 MW_{th} and is used for heat and power production with a gas engine. The LT-CFB gasifier is designed for processing high-alkali fuels with low ash-melting temperatures (e.g. straw, sludge) and is therefore operated below 750°C. The system uses air-blown dual fluid beds to gasify the fuel in two stages. The low temperature causes the efficiency to be high around 90%, but also results in poor gas quality with a high tar load. As the produced ash is not melted and mineral are retained as solids, it has a high fertilizing value, with high amounts of potassium, phosphorus, chlorine, sulphur etc. Unlike the TwoStage gas, LT-CFB has not been tested for engine operation before.

Project objectives

The project aims at optimizing gas engine operation with regards to power, electric efficiency and emissions. Specifically, this is done by varying the air-fuel ratios, ignition timing and pressure charging of a spark ignition engine. The applied fuels are simulated biogas, TwoStage product gas and simulated LT-CFB product gas. Simulated gases are mixed from gas cylinders and the TwoStage gas is live gas directly from the Viking gasifier. The overall aim is set at small-scale engines with district heating production.
Using these fuels in a modified engine setup, a TWC is installed on the exhaust to reduce emissions of unburned fuel and NO_x. Testing will focus on keeping a high performance, while operating within the emission limits given by the Danish government.

Research activities

This section presents the necessary background knowledge of engine operation, experimental setup, and the research results.

Background

This project will focus on gas engine optimization, with regards to emissions and performance when using selected biofuels. Fuels, engine and the three-way catalytic converter will be briefly described. The engine will be optimized by varying key parameters, which difference on the operation will be presented in this section:

- **Ignition timing (θ_{ign}):** During the combustion cycle, the spark ignition occurs before the piston reaches top dead center in order to allow the combustion time to happen. The ignition timing is measured in crank angle degrees before top dead center (CADBTDC). There is an optimal ignition timing that increases both power and efficiency of the engine. This point varies with engine speed, load and fuel as the combustion will have different rates and conditions based on these parameters. Ignition timing influences mainly maximum pressure and temperature in the cylinder during combustion.
- **Specific air-fuel ratio (λ):** The λ-value is a measure of the air-to-fuel ratio used (AF) compared to the stoichiometric ratio (AF_{st}). Thus if $\lambda=1$ the combustion is stoichiometric, if $\lambda>1$ then there is excess air and the combustion is lean and for $\lambda<1$ the combustion is rich and there is no excess air in the exhaust. This ratio is naturally dependent of the used fuel and affects the combustion process with regards to pressure, temperature and emissions. The temperature is the highest at slightly rich conditions around $\lambda=0.9$ and significantly lower at very lean conditions. The excess air-fuel ratio is defined as:

$$\lambda = \frac{AF}{AF_{st}}$$

- **Pressure charging (P_c):** Pressurization of the inlet air/gas mixture to the cylinder will increase the density of the air. This allows additional fuel to be added to the cylinder, which will increase the power output of the engine. Thus the power of an engine with a given displacement volume can be increased. The efficiency can increase slightly as well, as the relative increase in power can be larger than that of the mechanical friction. Pressurizing the air causes increases in maximum pressure and temperature in the cylinder.

Fuel characteristics

The properties of biogas and product gas are very different for that of natural gas. Biogas contains a large share of inert CO_2 and its heating value is still high, while product gas has a much lower heating value and a large share of CO_2 and N_2. The inert fraction will lower the maximum temperature in the cylinder, as it dilutes the flammable mixture, which leads to reduced knocking and NO_x emissions. Natural gas is a blend of primarily CH_4 and other lighter hydrocarbons such as C_3H_8. These lighter hydrocarbons are easily flammable and have increased tendency to cause knocking. Product gas and biogas contain no such light hydrocarbons and are thus more knock resistant than natural gas. Hence increased compression ratio and pressure charging might be applied or a richer combustion can be carried out without engine knocking.

The heating value of product gas is typically a factor of 6-8 times lower than natural gas, which would indicate a large loss of power in the engine. However, as Table 2 shows, the product gas requires less combustion air for the stoichiometric mix of fuel and air, and the resulting heating value of the air/gas mixture is only 24% lower than that of natural gas. Thus only a small decline in power is expected in comparison.

Fuel	CH_4 [vol%]	CO [vol%]	CO_2 [vol%]	H_2 [vol%]	N_2 [vol%]	LHV [MJ/nm^3]	$LHV_{st,mix}$ [MJ/nm^3]
Natural gas	-	-	-	-	-	39	3.4
Product gas (TwoStage)	1	20	15	30	34	6	2.6

Table 2 - Gas composition and lower heating values (LHV) for natural gas and product gas from the TwoStage Viking gasifier. $LHV_{st,mix}$ is the LHV of the air-fuel mixture at stoichiometric conditions [3].

Emission control

The most important pollutants for stationary gas engines are NO_x, CO and hydrocarbons. The only significant hydrocarbon emissions are expected from the biogas and it is primarily thought to be because of slip fuel and will not be discussed further.

CO emissions

CO emissions typically originate from slip-fuel that avoids combustion and incomplete combustion of hydrocarbons. Incomplete combustion of hydrocarbons can produce CO emission if the fuel conversion is insufficient due to very lean conditions and/or slow combustion rate. At such conditions bulk quenching occurs, as the flame is extinguished before combustion is complete. Slip-fuel emissions occur due to bulk quenching and also trapped fuel-CO is in e.g. the cylinder crevices between the piston and cylinder wall. As the piston expands, the CO is released to the cylinder, but might not be converted if the conditions are insufficient in the exhaust.

CO emissions are namely sensitive to the air-fuel ratio of the combustion. As the process becomes increasingly richer the CO emissions will increase, due to lower access to oxygen and unburned CO will be emitted. Close to stoichiometric conditions the λ-value causes large differences in emissions. Sufficiently lean combustion produces the lowest emissions and CO emissions are almost constant over a large span.

Very lean conditions can reduce the temperature to a level where the combustion is incomplete and hence emissions will increase.

Ignition timing has an effect on the maximum temperature in the cylinder and can hence affect the combustion efficiency. As the θ_{ign} is retarded, more of the combustion will occur after the top dead center. This will lower the maximum temperature and time available and thus limit combustion of CO and emissions will increase.

Pressure charging have relatively low effect on emissions in comparison and are primarily applied for performance optimization.

NO_x emissions

Emissions of nitrogen oxides originate from oxidized fuel-N and atmospheric nitrogen. As fuel-N is not of significant importance in this project, it will not be discussed further. The main source of NO_x is oxidation of atmospheric nitrogen at very high temperatures. The reaction strongly depends on temperature, but also on oxygen-availability and thus reduction should focus on lowering the maximum temperature and/or utilize an oxygen starved environment.

The air-fuel ratio affects the maximum temperature of the cylinder, as the combustion mixture is diluted to a minimum around stoichiometric. The temperature is highest at λ-values around 0.9, but as there is no oxygen available, no significant amounts of NO_x are produced. The emissions are highest at a λ-value of around 1.1, where the availability of excess oxygen offsets the slightly lower temperature. From this value the emissions decline both towards richer and leaner combustion.

The ignition timing has a large impact on NO_x, as it affects the maximum temperature. Ignition timing is often used for NO_x control, lowering the performance slightly for reduced emissions. Retarding the θ_{ign} will cause significant emission reductions as the temperature is lowered.

Pressure charging has a significant effect on emissions. The maximum temperature increases during this process and will therefore increase NO_x emissions.

Emission regulation in Denmark

In order to evaluate the engine operation, the emissions are compared to the regulation limits set by the Danish government. The permitted emissions from product gas and biogas engines where until recently in line with those of natural gas. However, as these biofuels have different CO emission mechanisms, the permitted emission levels were increased for support. The permitted limits for CO and NO_x are given in Table 3. There is currently no emission limits for unburned hydrocarbons (UHC).

Fuel	**NO_x**	**CO**
Natural gas	507	507
Product gas	507	3000
Biogas	507	1200

Table 3 - Permitted emissions of CO and NO_x in Denmark [mg/nm^3 @ 5% O_2].

The emission levels are given at a reference condition: 5 vol% O_2, 0^{o}C and 1.013 bar. The conversion to 5% oxygen at reference, C_r, from the measured, C_m, at oxygen level, O_m, is done by:

$$C_r = \frac{21-5}{21-O_m} \cdot C_m$$

Engine performance and knocking

Engine performance depends on a host of parameters. Those relevant here is fuel conversion efficiency, mean effective pressure, heat loss, and use of pressure charging.

The electric efficiency, η_e, is defined by the electric output from the generator, P_{el}, and thermal input of fuel, Q_{in}, based on LHV and mass fuel mass flow, m_{fuel}:

$$\eta_e = \frac{P_{el}}{Q_{in}} = \frac{P_{el}}{m_{fuel} \cdot LHV_{fuel}}$$

Pressure charging allows more air, and thus more fuel, to be introduced into the cylinder and increases work and possibly efficiency depending on fuel, engine etc. This cause an increase in fuel conversion efficiency, as the combustion process is enhanced by increased temperature and pressure. The increased friction is offset by the increased work. Pressure charging is namely limited by knocking in the engine and NO_x emission, which depends on the engine design and the fuel. Fuel with high knocking resistance and low NO_x can hence use this measure to a larger extend. Knocking resistance is often correlated to the ignition characteristics of the fuel and its ability to lower the maximum temperature through e.g. dilution with inert. In general, fuel with smaller molecules (CH_4, CO, CH_3OH, H_2) have high knocking resistance. This makes product gas and biogas suitable for pressure charging with regards to knocking.

Heat losses to the cylinder walls and the exhaust are of significance, as they represent a loss of energy that could have done work on the piston. Heat losses are affected by the maximum temperature of the combustion and the ignition timing. Combustion temperature is at a maximum around stoichiometric conditions and the power output is at a maximum at λ-values between 0.9 - 1.0, as the effective mean pressure is highest in this interval. However, as the combustion temperature is high, the heat losses are equally high which lowers the efficiency. In general, leaner combustion is more efficient due to better fuel conversion and heat control.

The ignition timing controls the amount of fuel that is combusted before and after the piston reaches top dead center. Thus, retarding the θ_{ign} will cause an increase in exhaust temperature, but also lower the losses to the cylinder walls, due to a lower temperature. Because of this, an optimum can be found where the ignition timing is optimal and the power and efficiency is slightly increased. Retarding the θ_{ign} will lower the maximum temperature and will thus reduce the tendency to knocking and NO_x.

Three-way catalytic converter

As mentioned, emissions can be lowered to a minimum if the engine is operated near stoichiometric-lean conditions and a three-way catalytic converter is applied. The TWC is named because of its ability to remove all three emissions (CO, hydro carbons and NO_x). The converter is a commercial technology that is widely applied in automotive engines and has relatively low cost. The TWC utilises catalytic material to promote oxidation and reduction simultaneously in the slipstream, oxidizing unburned CO and hydrocarbons to CO_2 and H_2O and reducing NO_x to N_2 (using unburned CO, H_2 and hydrocarbons for reduction). In order for both processes to happen, the excess air in the engine exhaust has to be carefully controlled and kept within a narrow window. Usually an exhaust oxygen sensor is applied to achieve strict control. The operating window should be kept in the range of λ-values of 0.01, with the operation being slightly lean or rich. An example of the operating window for TWC is shown in Figure 2. The converter typically needs temperature around 250-300^{0}C to be effective.

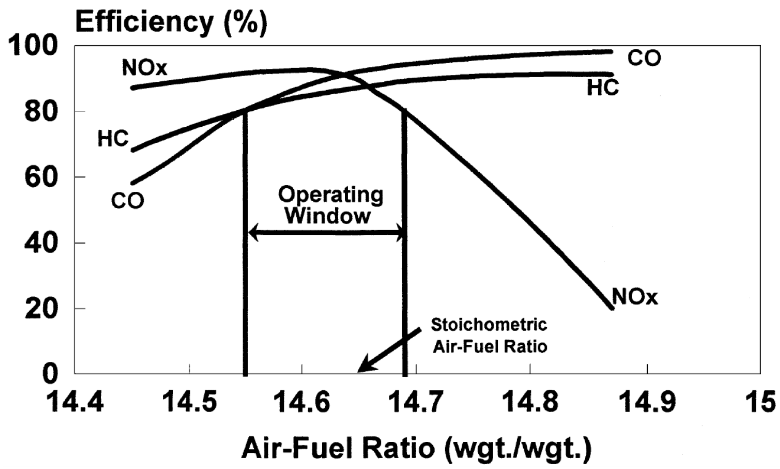

Figure 2 – Example of an operating window for a three-way catalytic converter [4].

This background section has presented the emission and performance mechanisms for engine operation. Biogas and product gas are seen to be promising fuel for high compression ratios and pressure charging, namely due to their composition and large inert fractions. Reductions in power compared to natural gas are expected, because of lower heating values, but might be offset by higher knocking resistance (as higher boost pressures can be applied) and operation closer to stoichiometric conditions. Emissions near stoichiometric operation are expected to be relatively high, but especially NO_x might be lower due to cooling from inert gas. Emission increase can be countered to adjusting ignition timing and air-fuel ratio of the engine before entering the TWC.

Materials and methods

This section presents the experimental setup used and the conditions of the research.

Experimental setup

The experimental setup is focused on a container, which contains the engine, system controls and auxiliary equipment. The setup acts a CHP engine, with connection to the electric grid and a cooling system simulating district heating production. The container is connected to external gas and air supply, including the natural gas grid, a manual gas mixer for cylinders and a direct line to the TwoStage gasifier. The manual gas mixer is adjusted according to the desired gas composition and the gas composition is measured before the gas enters the container. The engine exhaust is coupled to the TWC and the exhaust is led to the outside environment. A PI-diagram of the setup is shown in Figure 3.

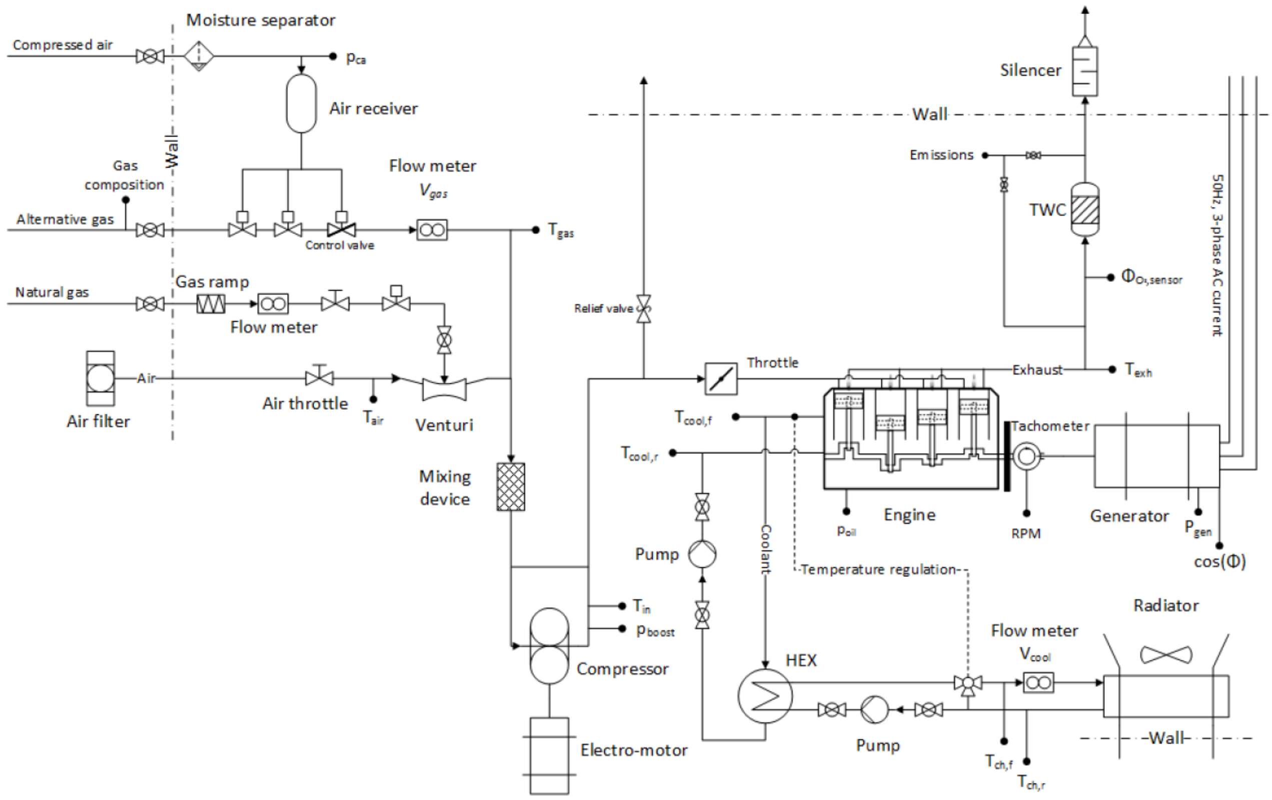

Figure 3 – PI-diagram of the engine setup with gas intake, cooling system, generator and exhaust.

Picture 1 Engine set-up

Intake and exhaust systems

The intake system is designed with two separate trains: natural gas and alternative gas. The engine is started and warmed with natural gas before switching to mixed or real product gas. The alternative gas flow is in the range of 0 - 45 nm^3/h and is adjusted by the exhaust oxygen sensor, while the air (and air-fuel ratio) is adjusted manually on a valve. The engine is naturally aspired, but has a compressor installed between the mixing device and engine to apply a boost pressure of 0 - 2.5 bar. Natural aspiration bypasses the compressor.

The hot exhaust from the engine is lead through the TWC. It is a commercial and mass-produced catalyst designed for cars. The catalyst is heated by the exhaust gas and is operated between 350-450^oC depending on operation. Gas outtakes are placed before and after the converter in order to measure gas compositions. The exhaust is lead through a silencer before being discharged to the outside environment.

Engine system

The gas engine is designed to be mounted with two different compression ratios. Both engine configurations are four-stroke, four-cylinder 1.86 L spark ignition Lister Petter models. The applied compression ratio in this project is 9.5:1. The engine runs at a regulated constant speed of 1500 rpm, to match the generator grid frequency. The electric power production is in the range of 0 - 20 kW_e. The engine ignition timing is controlled digitally through the control system and is varied between 24^o and 0^o CADBTDC.

The engine is internally cooled by simulating a district heating network. The engines cooling circuit is connected via a heat exchanger to a second cooling cycle that is cooled by forced convection the outside environment. The temperature is measured before and after the forced convection and hence the heat production can be calculated. The heat production is in the range of 0 - 120 kW_{th}.

Tests

The experimental setup has been modified and preliminary tested in this project from mid-2012 to mid-2015. Tests with real TwoStage product gas have been performed in December 2014 and January 2015. Tests with synthetic gas mixes have been performed from May 2015 to August 2015. All tests were initiated after stable engine operation and thermal stability with natural gas. Experimental results are taken as averages over 2 - 5 min of operation. An overview of performed tests is shown in Table 4.

Fuel	Air-fuel ratio	Ignition timing	Pressure charging
TwoStage gas*	√	√	√
Biogas**	√	√	
LT-CFB gas**	√		

Table 4 – Overview of experimental tests.*Live gas from the Viking gasifier. **Synthetic gas mixed from cylinders.

The test with LT-CFB gas was only done for varying air-fuel ratio at constant ignition timing (set at 18^o CADBTDC). The test with biogas was carried out without a gas flow meter, and hence the efficiency cannot be determined.

Results

Gas compositions

While the gas compositions were constant for the Biogas and LT-CFB product gas, because they were from cylinders, the test with real TwoStage product gas fluctuated slightly. An example of the gas composition during the live test is shown in Figure 4. The engine tests were carried out with the three gases that are listed in Table 5.

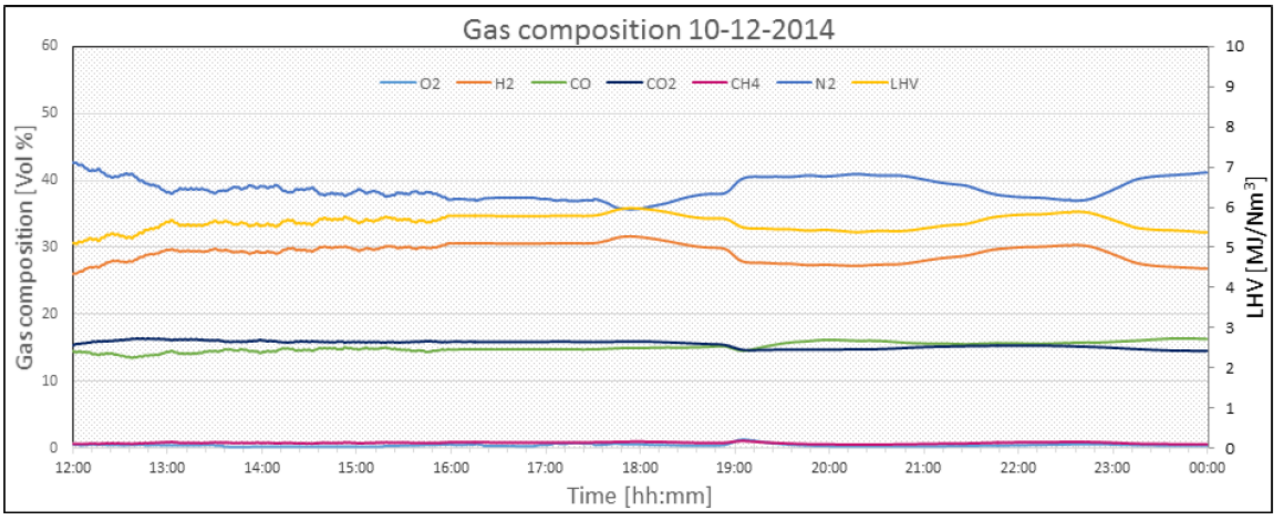

Figure 4 – Example of fluctuations in gas composition during test with real TwoStage product gas.

Gas	H_2 [vol%]	CH_4 [vol%]	CO [vol%]	CO_2 [vol%]	N_2 (rest) [vol%]	LHV [MJ/nm^3]
LT-CFB	10.5	5.8	13.1	20.0	50.6	4.82
Biogas	0.0	70.0	0.0	30.0	0.0	25.06
TwoStage*	27	1	15	15	42	5.5

Table 5 - *Values taken as averages due to fluctuations

Engine performance

The performance is measured by power output and electric efficiency. The performance for the gases is shown in Figure 5, Figure 6, Figure 7, Figure 8, and Figure 9.

The power outputs of the gases are seen to increase with decreasing λ as expected. The power is approximately 20-25% higher at a λ of 1.03 compared to high end values for all the gases, showing the incitement to operate near stoichiometric (Figure 5 and Figure 7). The main difference between the tests, are that the biogas power output is significantly higher than that of the product gases. This is namely due to the fivefold higher heating value of the gas, that increases the power with 20 - 30%. This is due to the lower oxygen demand of the product gases, which causes the fuel-air mix to have a relatively high value.

As a function of ignition timing, it is seen that there are different optimums for the biogas and TwoStage gas. The biogas optimum is between 18-22^o CADBTDC (Figure 6), while the TwoStage gas optimum is between 5 – 12 CADBTDC (Figure 8). This is an interesting result, as retarded ignition timing closer to top dead center will lead to a decrease in NO_x formation. The ignition is generally not seen to impact the power production as much as λ, ranging from 10-15% power differences at various ignition timings.

The electric efficiency of the engine on product gases is ranging from 24 - 33%, which is in line with expectations based on previous engine tests with the setup on product gas[5]. The efficiency measurements done for the TwoStage gas is subject to high uncertainty. Based on previous tests[6], the efficiency with TwoStage gas is expected to peak between λ 1.0 and 1.5 and not increase continuously with increasing λ as in Figure 7. The efficiency with the LT-CFB gas is seen to be more reasonable and increases with lower lambda values (Figure 5). This effect is also seen elsewhere[6], as an increased amount of inert shifts the optimum closer to stoichiometric.

Tests with pressure charging are carried out for TwoStage product gas. The power is seen to increase with increasing pressures as expected (Figure 9). The efficiency decreased till a certain pressure, where it flattened out to a near constant value. At max boost pressures of 900 mbar, the produced power is increased 80 - 110% and the efficiency decreases 5 - 7%[7].

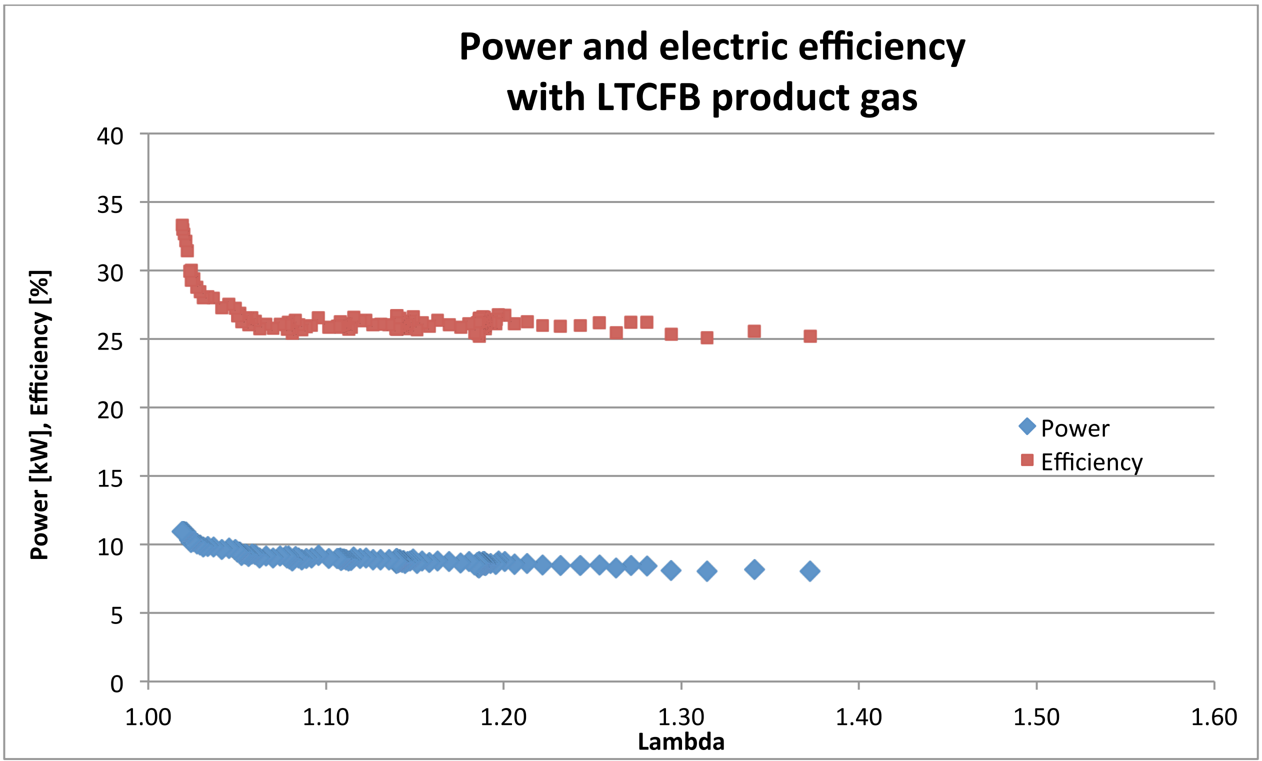

Figure 5 – Performance with LT-CFB product gas as a function of λ at ignition timing 18 CADBTDC

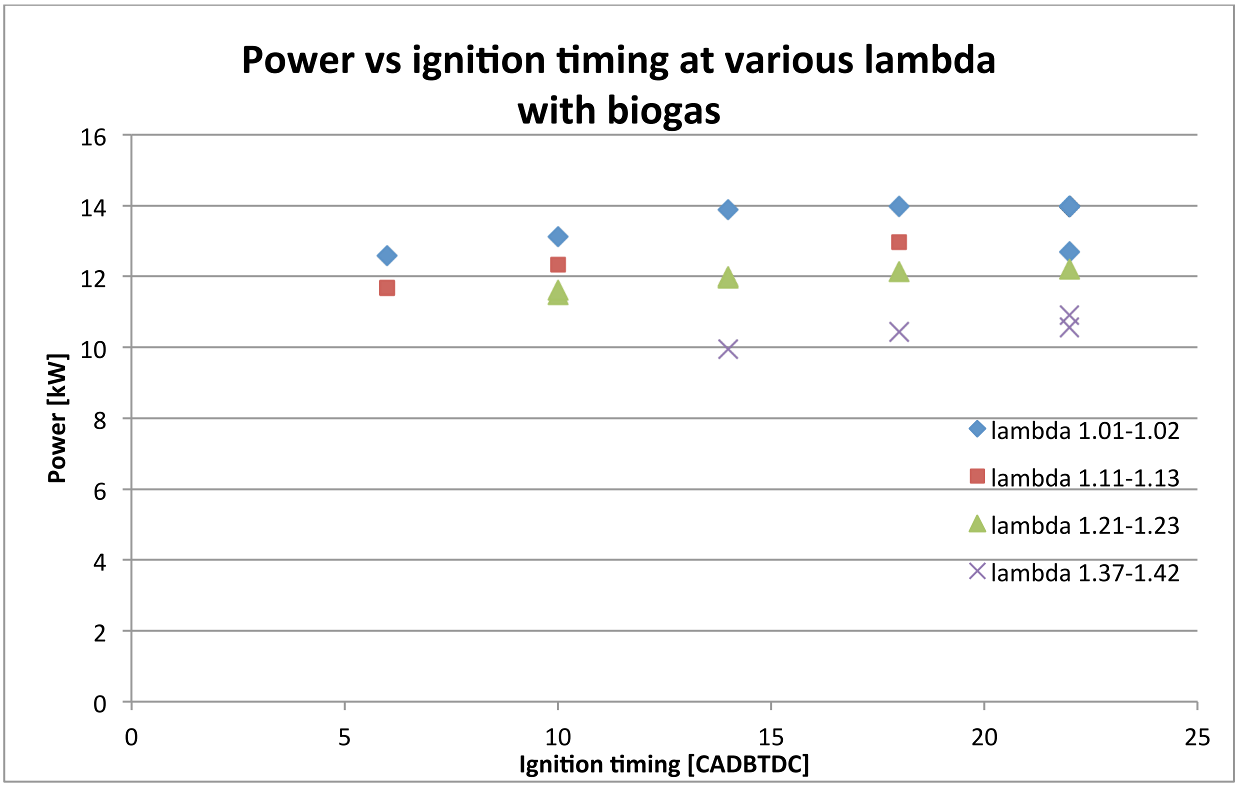

Figure 6 – Power output with biogas as a function of ignition timing and λ

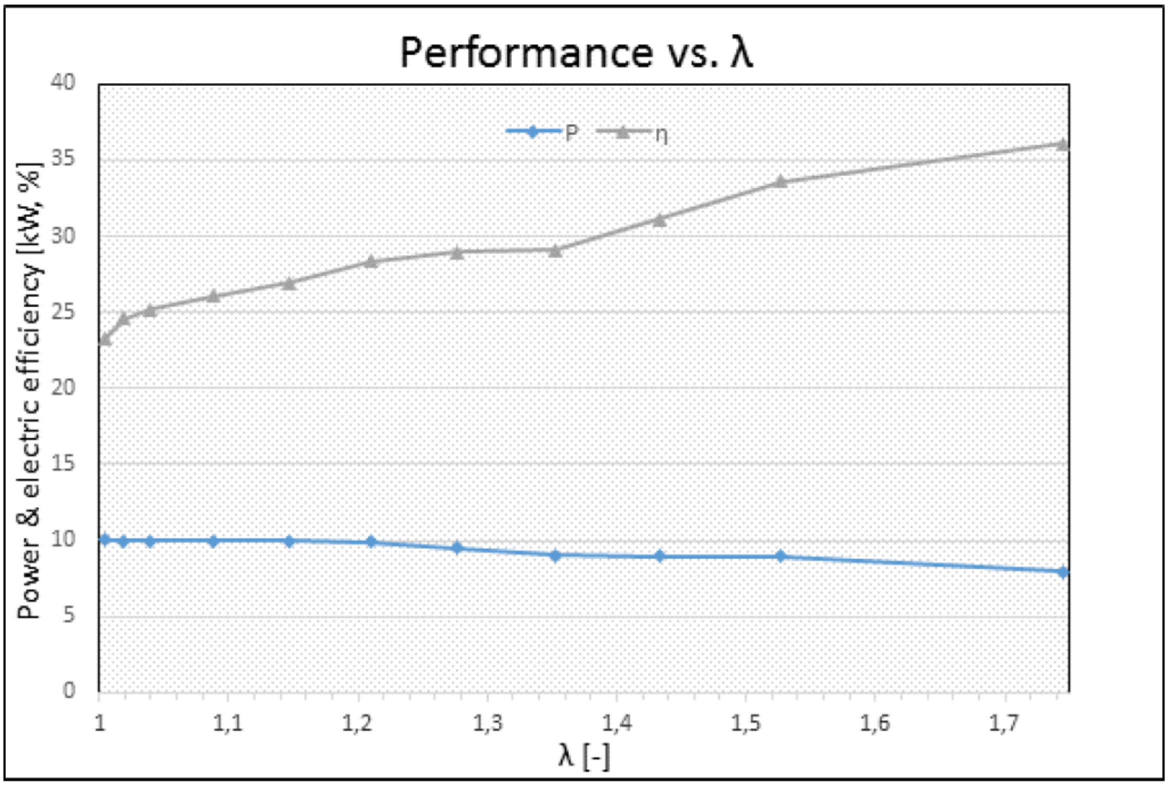

Figure 7 – Performance with TwoStage product gas as a function of λ

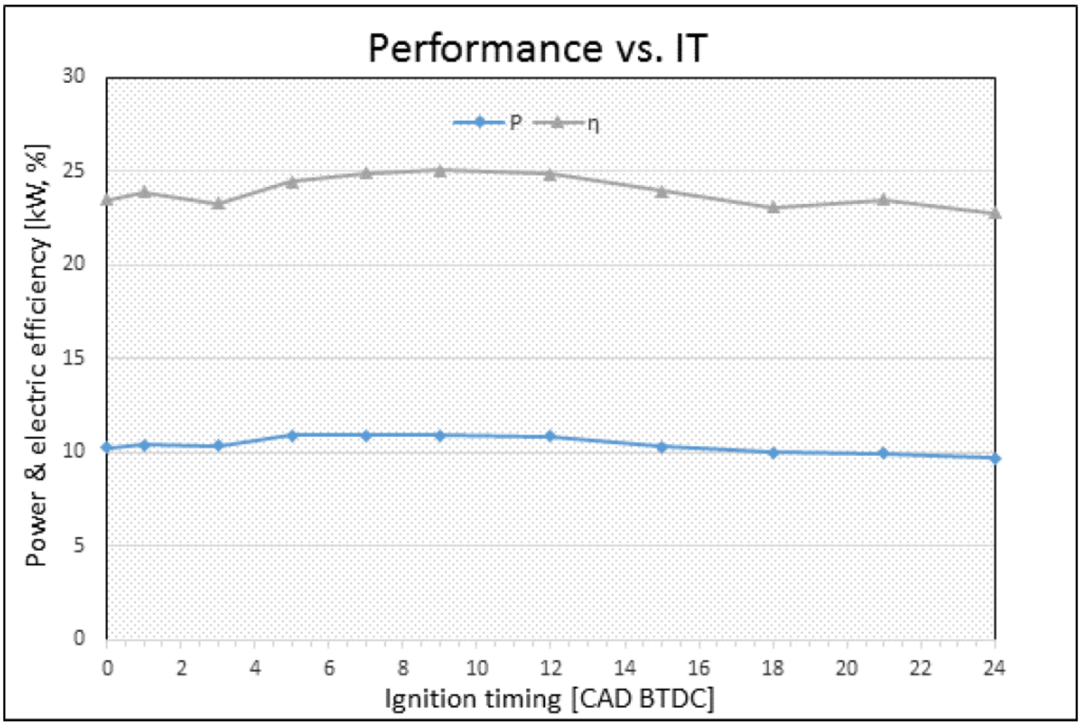

Figure 8 - Performance with TwoStage product gas as a function of ignition timing. λ is 1.03

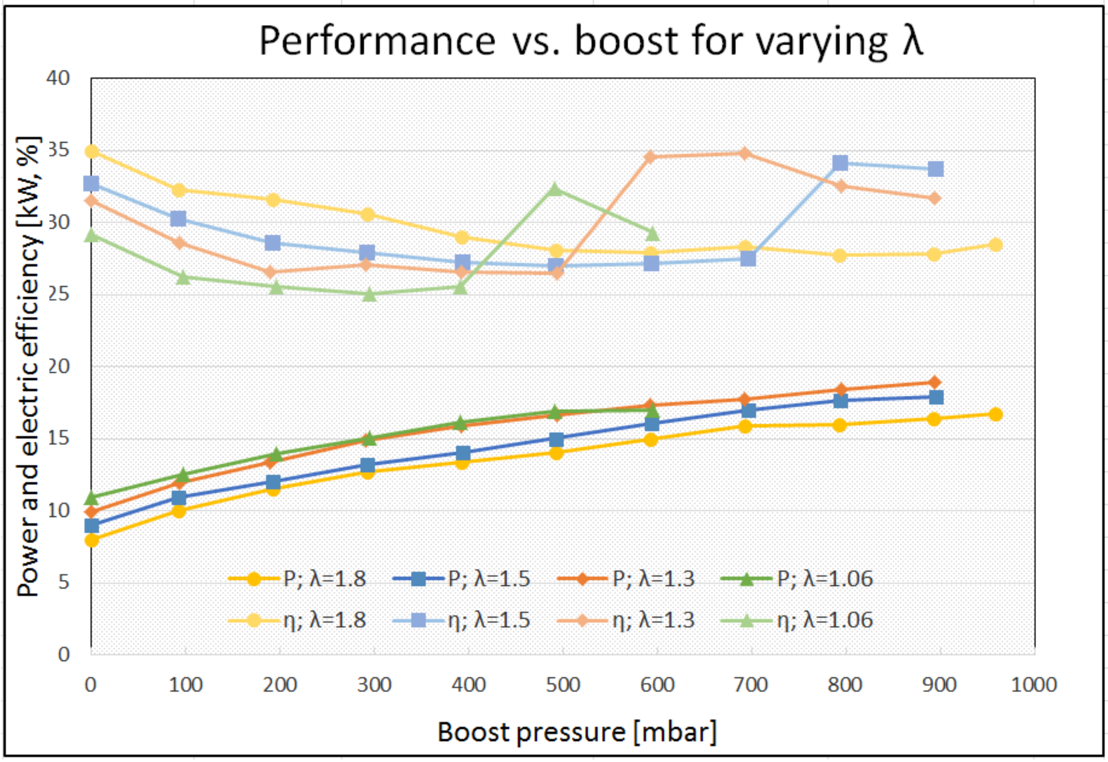

Figure 9 – Performance with TwoStage product gas with pressure charging

Emissions

The emissions measured are CO, NO_x and UHC. Hydro carbon measurements are carried out by Dansk Gasteknisk Center (DGC) for the test with biogas.

Emissions as a function of λ

Emissions as a function of λ are shown in Figure 10 to Figure 13,

CO emissions ranged between 400 – 3800 mg/nm^3 before the TWC. As explained earlier, this value is strongly dependent on λ, with high emissions at very low and high λ-values. The CO emissions are slightly lower for biogas, which is because the product gases contain fuel-CO, which inevitably will result in higher emissions from slip-fuel, see Figure 10 and Figure 11. While the range for LT-CFB gas emissions is low before the TWC, the CO emissions is seen to be significantly higher than TwoStage at λ-value of 1.2 (2000 compared to 800 mg/nm^3) (Figure 12, and Figure 13). This is thought to be the result of lower peak temperatures by increased inert dilution and possibly because of lower combustion properties with a lower content of highly combustible hydrogen.
Despite very different regulation for product gas and biogas, it can be generally stated that CO emissions can be kept sufficiently low by regulating the fuel-air ratio above stoichiometric conditions. However, emissions should be minimized and the TWC shows excellent ability to reduce emissions to a minimum.

NO_x emissions ranged between 80 – 3400 mg/nm^3, with peak values around λ-values of 1.1 as expected. The biogas produced significantly larger volumes of NO_x up to 3400 mg/nm^3, while the TwoStage gas reached up to1400 mg/nm^3, and the LT-CFB gas produced up to 100 mg/nm^3. The biogas produced by far the largest volume, which was expected with a much higher combustible fraction of fuel components – resulting in higher temperatures and hence more NO_x. Between the two product gases, there is remarkable difference, as the LT-CFB gas produces nearly no NO_x across λ and the TwoStage gas is above the regulated limit at λ-values lower than 1.3. This difference is mainly associated with lower temperatures caused by: 1) the higher inert content of LT-CFB gas and 2) the lower hydrogen content of the LT-CFB gas that has a high flame temperature.
NO_x emissions have a strict emission limit and only the LT-CFB gas can reach the required level below a λ-value of 1.3. LT-CFB gas shows remarkably low emissions. The TWC is not seen to affect the NO_x emissions significantly. At lower, lean λ-values the NO_x can thus not be controlled solely by the air-fuel ratio.

UHC emissions are measured for the biogas tests and show levels between 600 - 1400 mg/nm^3. The emissions are seen to follow the pattern of CO, with higher emissions at lower and very high λ-values. The TWC shows to be less effective for UHC, as only lower and mid-level reductions are seen.

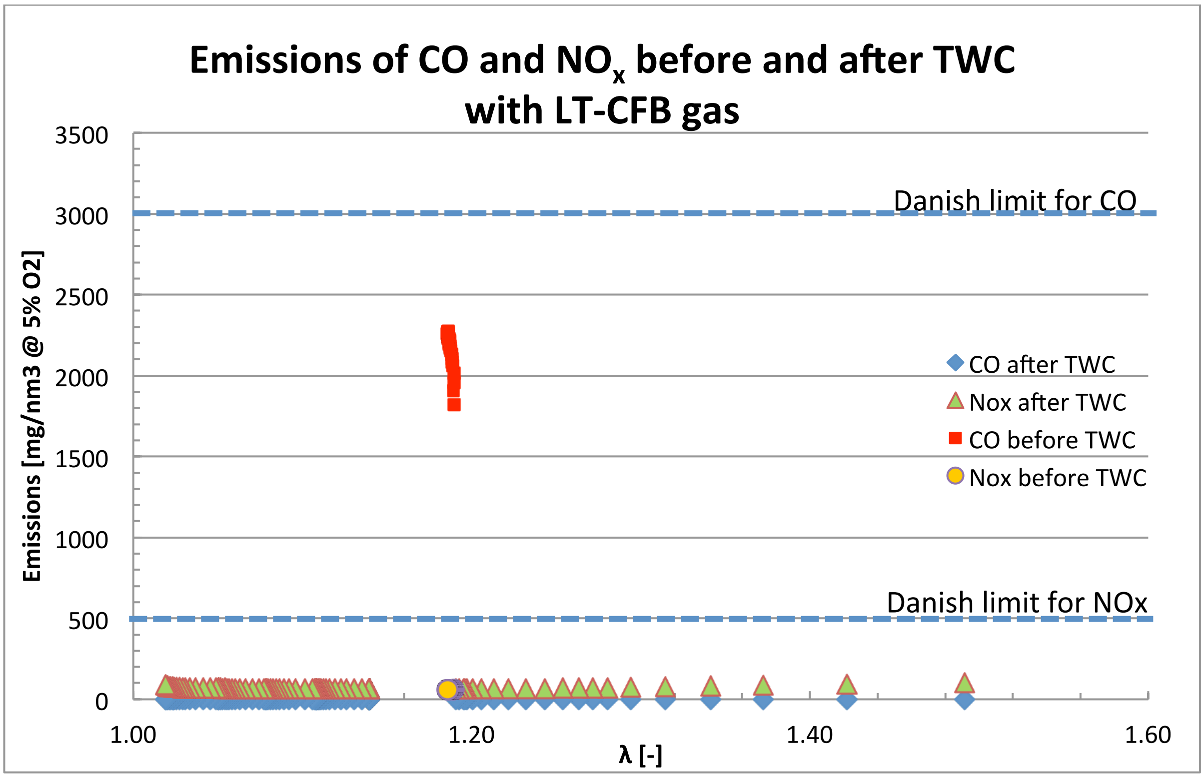

Figure 10 – Emissions of CO and NOx as a function of λ for LT-CFB gas at ignition timing 18 CADBTDC

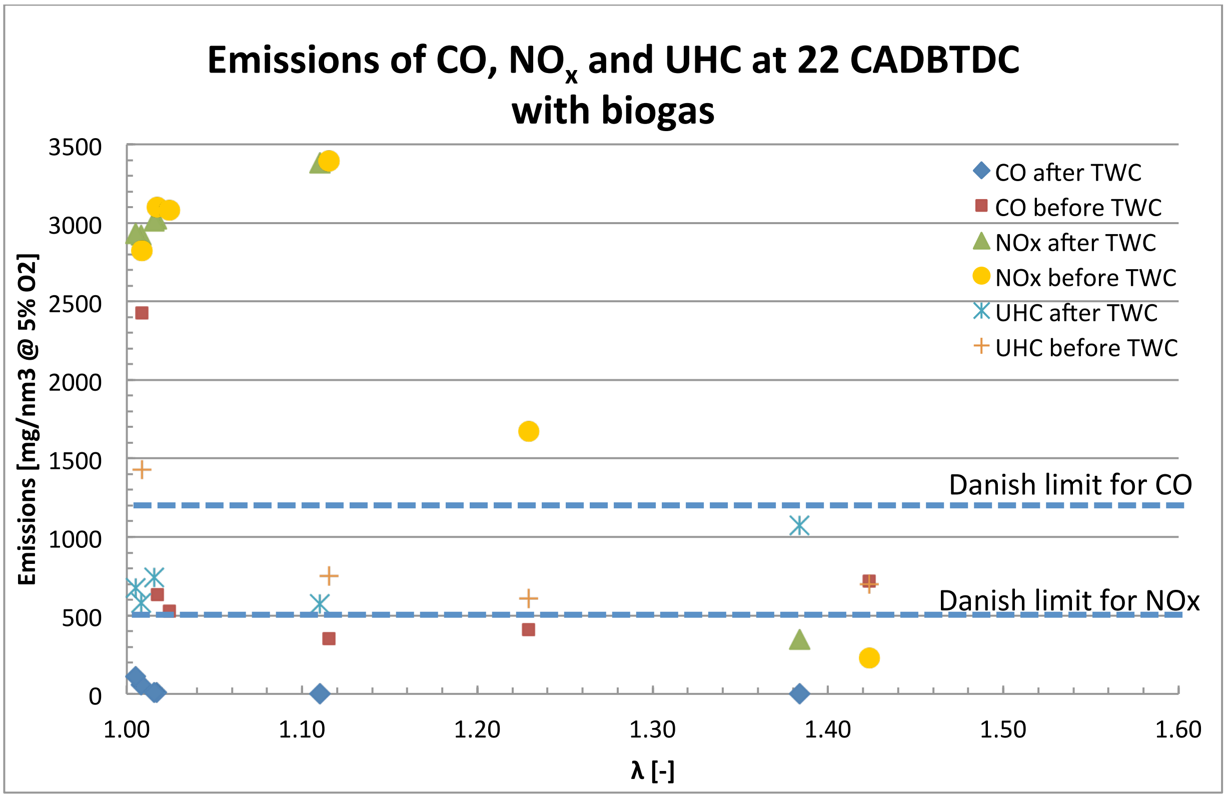

Figure 11 – Emissions of CO, NOx and unburned hydrocarbons (UHC) as a function of λ for biogas at ignition timing 22 CADBTD

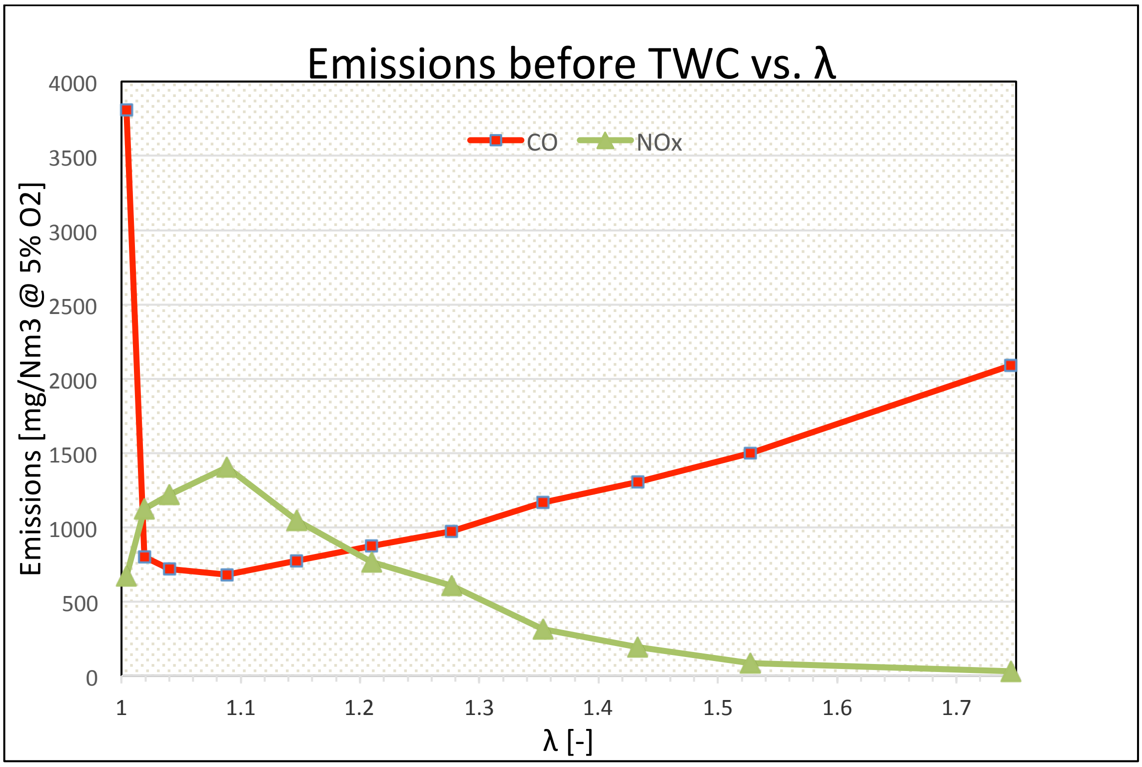

Figure 12 - Emissions of CO and NOx as a function of λ before the TWC for TwoStage product gas

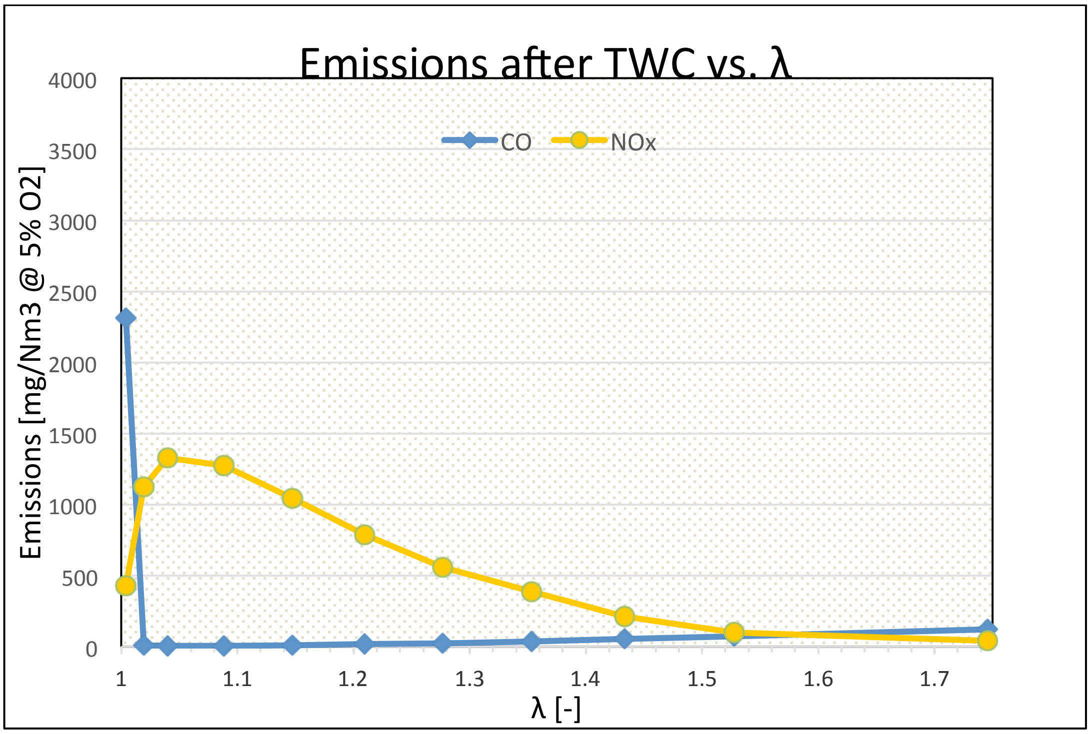

Figure 13 - Emissions of CO and NOx as a function of λ after the TWC for TwoStage product gas

Emissions as a function of ignition timing

Varying the ignition timing will cause a difference in the combustion temperature and adjust the time available for complete combustion. There also exists an optimum for igniting the fuel, in which the performance will be the highest, depending on the fuel and engine. Ignition timing tests are done for TwoStage product gas and biogas and are shown in Figure 14, Figure 15, Figure 16, and Figure 17.

CO emissions are generally less affected by varying ignition timing. For the TwoStage gas, CO emissions range between 700 – 900 mg/nm^3 (before TWC), while emissions for the biogas range between 0 - 400 mg/nm^3 (after TWC). CO emissions follow the pattern of the NO_x, with lower values at lower ignition timings (Figure 14 and Figure 15).
Ignition timing is therefore not an appropriate measure to handle CO emissions, but can reduce levels by retarding the timing. The TWC is again seen to effectively reduce CO emissions to a minimum.

As a temperature controller, ignition timing strongly affects NO_x formation, with reduced emissions at retarded timings. For the TwoStage gas, ignition timing is seen to reduce emissions up to 93%, down to 118 mg/nm^3. For biogas, NO_x reductions up to 62 and 69 % are seen, lowering the level to 1100 and 500 mg/nm^3 respectively.
The TWC shows only minor effect on the NO_x emissions. Thus the ignition timing is of great importance when reducing NO_x near stoichiometric. It is seen on Figure 17, that to reach levels below regulation it is necessary to operate the gas engine at a λ-value ≥1.21 and retarded ignition timing of ≤10 CADBTDC.

UHC emissions are again seen to follow CO emissions for the biogas, with very limited effect by the TWC. The ignition timing is however applicable for UHC reduction, up to 58% is seen.

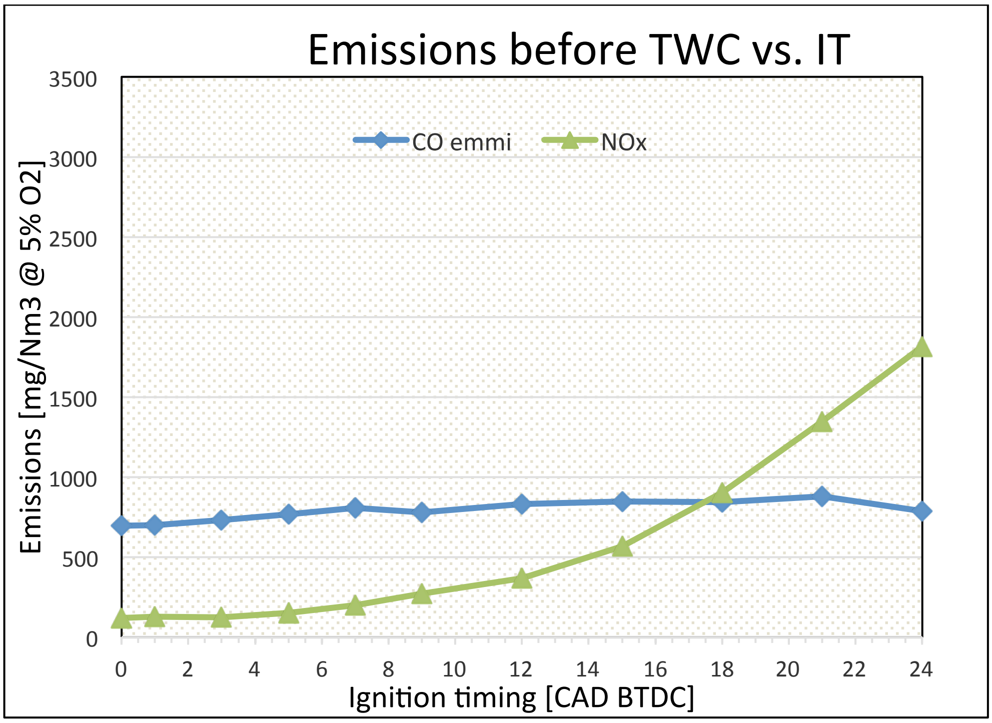

Figure 14 - Emissions of CO and NOx as a function of ignition timing before the TWC for TwoStage product gas at a λ-value of 1.03

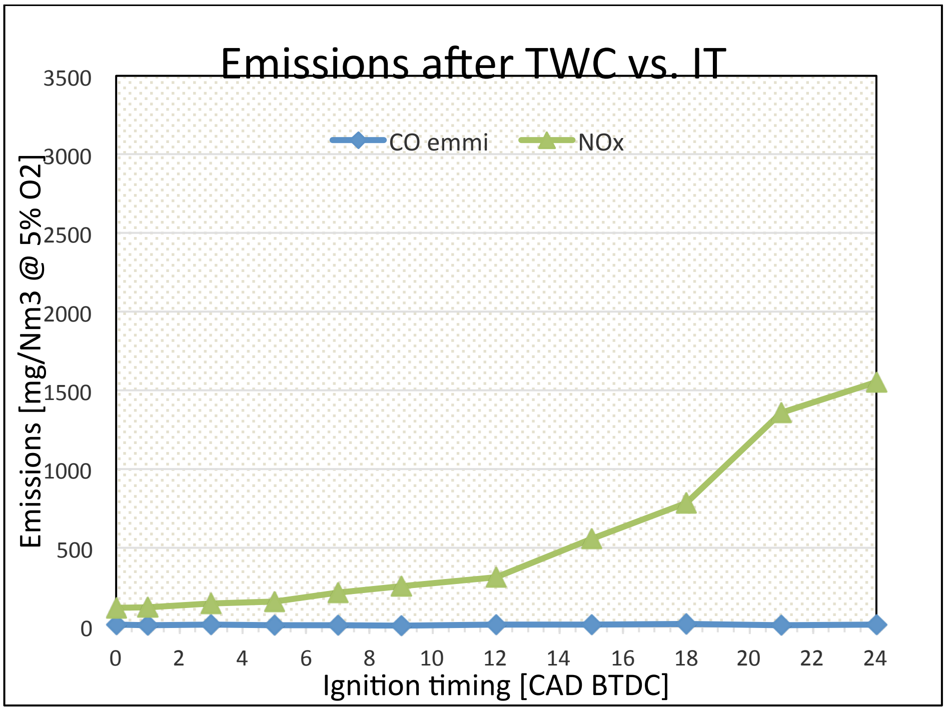

Figure 15 - Emissions of CO and NOx as a function of ignition timing after the TWC for TwoStage product gas at a λ-value of 1.03

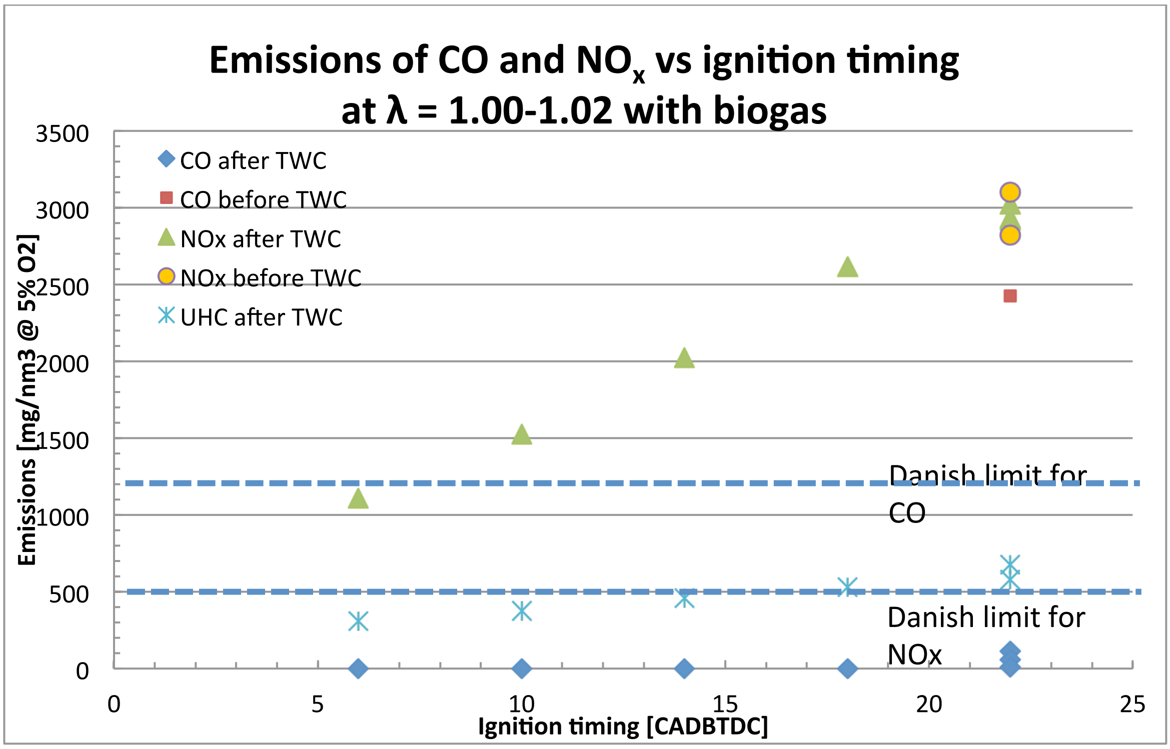

Figure 16 –Emissions of CO and NOx as a function of ignition timing for biogas at λ-values between 1.00 - 1.02

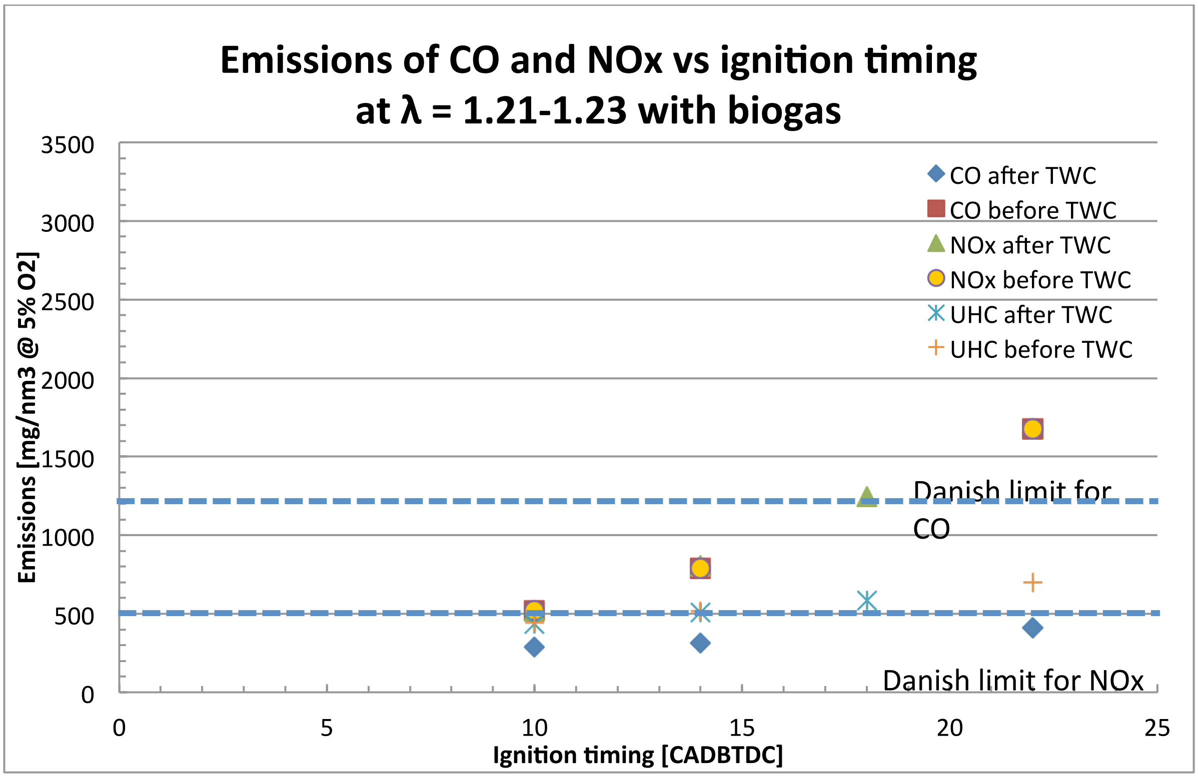

Figure 17 – Emissions of CO and NOx as a function of ignition timing for biogas at λ-values between 1.21 - 1.23

Emissions as a function of pressure charging

Pressure charging increases the amount of fuel and thus affects the maximum pressure and temperature in the cylinder. The results are performed with TwoStage product gas and are shown in Figure 18 and Figure 19.

As the boost pressure increases, the CO emissions are seen to slightly increase from 700 – 1000 mg/nm^3. The CO is generally less affected by pressure charging, as expected. The TWC shows excellent ability to convert CO in the exhaust to a negligible level (Figure 18).

NO_x emissions are strongly influence by the applied boost pressure. Boosting the engine up to 400 mbar, will increase the emissions from 1500 - 3500 mg/nm^3. Therefore pressure charging should be used with retarded ignition timing, higher λ and/or inert dilution of the fuel. The TWC shows very limited effect on the NO_x (Figure 19).

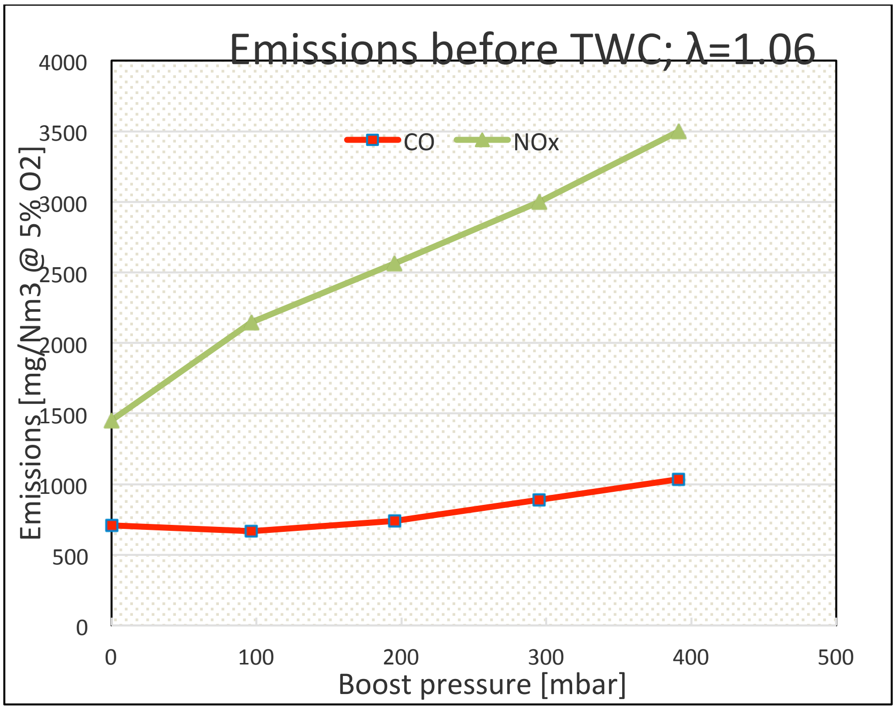

Figure 18 – Emissions from TwoStage product gas of CO and NOx as a function of boost pressure before the TWC at a λ-value of 1.06

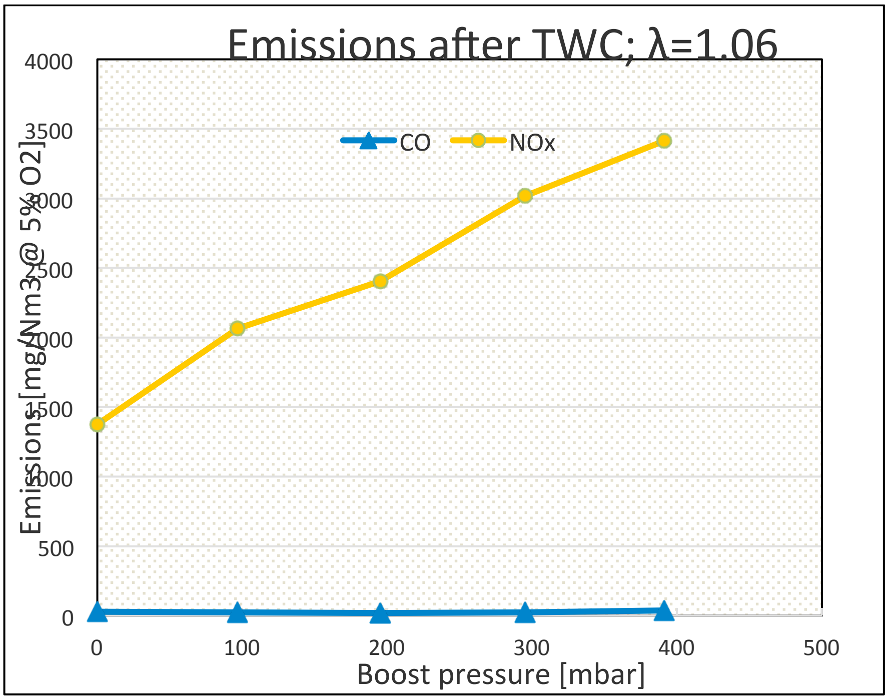

Figure 19 - Emissions from TwoStage product gas of CO and NOx as a function of boost pressure after the TWC at a λ-value of 1.06

Sub-stoichiometric operation

During the experimental work, it was attempted to operate the biogas at sub-stoichiometric conditions. Operating near the stoichiometric limit could prove effective for the TWC, enabling it to both reduce NO_x and oxidize CO. The air addition was reduced, but because no air flow meter was installed, the exact λ-value is unknown (λ is calculated using the exhaust oxygen percentage). It is estimated to be slightly below stoichiometric. During 9 minutes, the engine was operated sub-stoichiometric and the emission results are shown in Figure 20.

While the slightly lean operation at λ-values between 1.01 – 1.02 yielded close to 3000 mg/nm^3 NO_x, the oxygen-starved test shows very low emissions well below the regulated limit for biogas. After initial stabilization, the level drops to below 150 mg/nm^3. The CO emissions increase as expected, from 50-100 mg/nm^3 at slightly lean, to 300 – 1200 mg/nm^3 sub-stoichiometric after the TWC –still well below regulation.

The large NO_x reduction seen is namely due to the oxygen-starved environment, that inhibits formation, and assumedly the TWC, that have increased NO_x conversion due to an increased share of oxygen-absorbing CO. It is unknown to what extend the TWC affects the emissions as no measurements were made prior to the TWC due to the shortage of fuel gas.

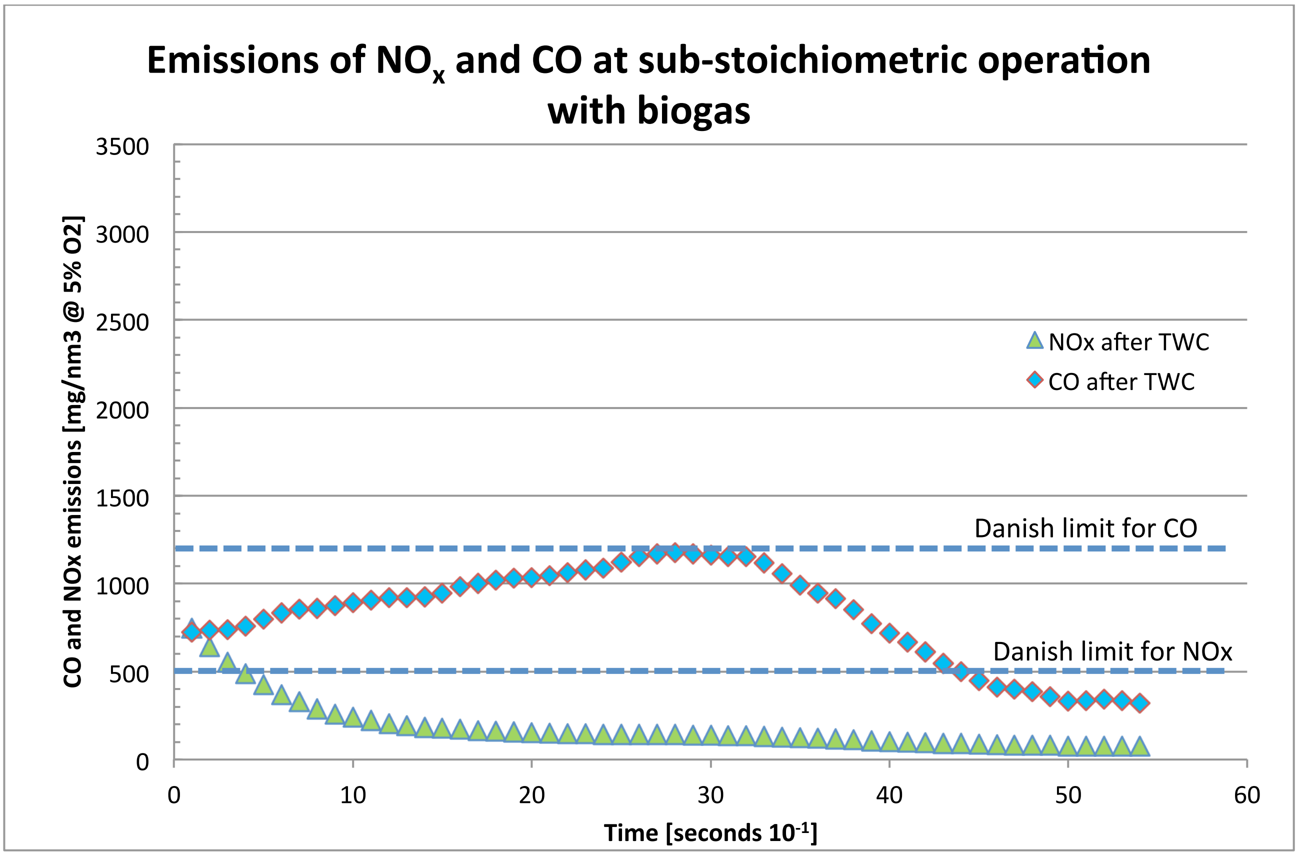

Figure 20 - Emissions of CO and NOx as a function of λ after the TWC, ignition timing is 22 CADBTDC

Three-way catalytic converter performance

Combining the data for the TWC across the tested gases, the conversion efficiency, η_T, based on emission levels before and after the catalyst, C, can be calculated:

$$\eta_T = 1 - \frac{C_{after}}{C_{before}}$$

The conversion efficiency of the TWC is shown as a function of λ and ignition timing in Figure 21 and Figure 22 respectively. The catalyst shows excellent ability to remove CO at a very high rate, above 94% across tests. Thus, it is shown that a simple and cheap TWC can easily reduce CO emissions in the large interval of operation that is used. Even at sub-stoichiometric conditions with a lot of CO, the catalyst showed the ability to reduce levels well below the regulated limit – and to a limit that is acceptable for natural gas. Hence, the catalyst can effectively be applied to engines, using biogas and product gas, to obtain CO emissions similar to known natural gas technology.

As seen in the previous results, the TWC is ineffective in reducing NO_x emissions at lean conditions. Across λ-values and ignition timings, the conversion efficiency is negligible or fluctuating between slightly positive and negative. As an average, the NO_x reduction is practically non-existing and the TWC acts solely as an oxidizing unit for CO and UHC. However, at sub-stoichiometric conditions the catalyst is assumed to display effective reduction, leading to very low levels of NO_x. This will however need further testing in order to be validated.

Hydro carbon conversion in the exhaust is seen to be close to negligible. As a function of λ, there is a weak correlation towards increasing conversion at higher λ-values. But varying the ignition timing shows that the conversion efficiency fluctuates, as with the NO_x, around 0%. Thus, the catalyst is not thought to promote any significant UHC oxidation and emissions will have to be dealt with alternately for biogas engines.

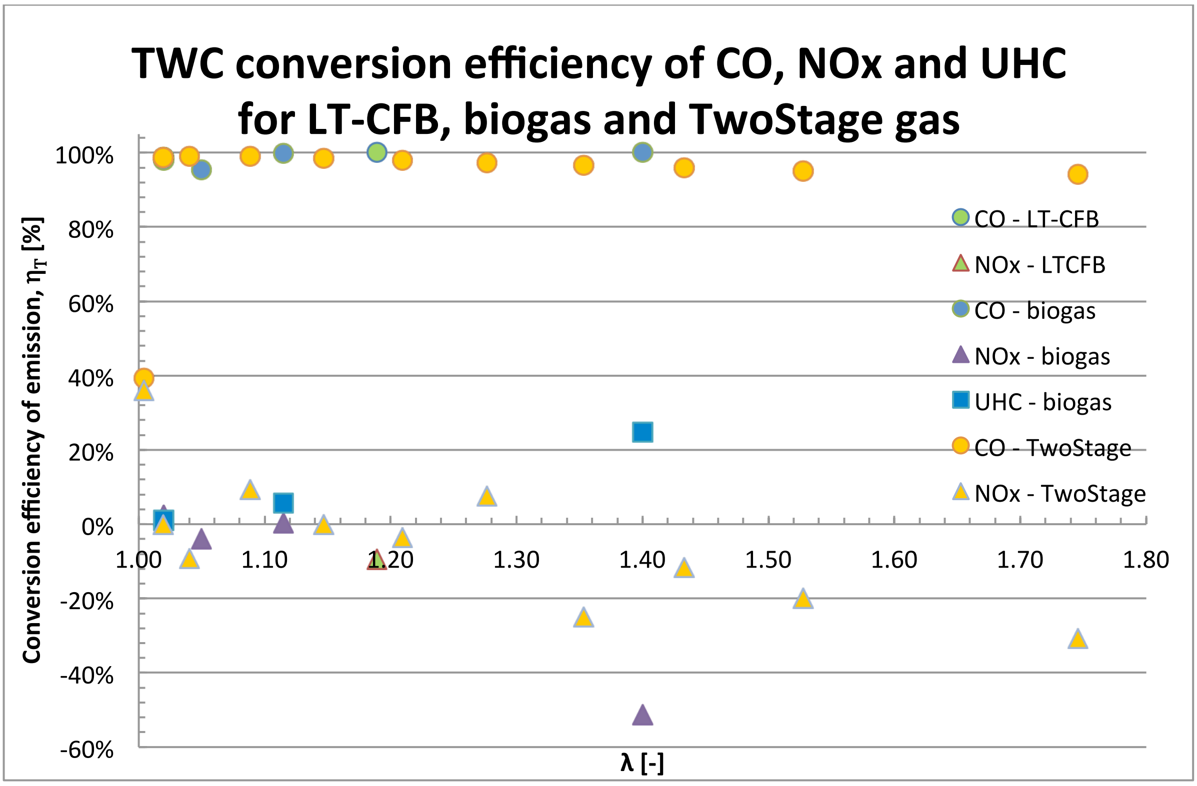

Figure 21 – Conversion efficiency of emissions in TWC vs λ for all gases

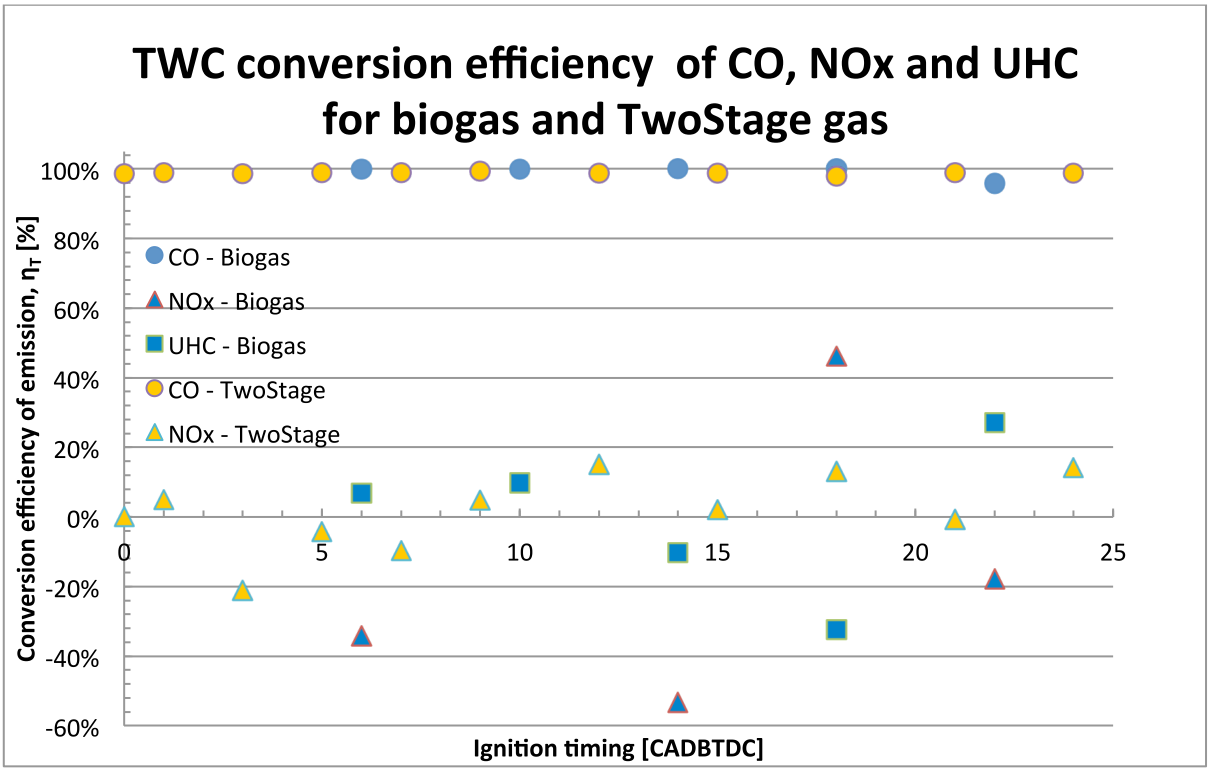

Figure 22 - Conversion efficiency of emissions in TWC vs ignition timing for biogas and TwoStage gas. λ-values vary between 1.01 – 1.75

Emission overview and evaluation

The emission results are summarized in Table 6, where the required λ and ignition timing for feasible NO_x levels are shown.

Gas	**λ-values** [-]	**Ignition timings** [CADBTDC]	**CO conversion in TWC** [%]	**NO_x after TWC** [mg/nm^3 @ 5% O_2]	**λ needed for NO_x limit** [-]	**Ignition timing needed for NO_x limit** [CADBTDC]
LT-CFB product gas	1.02 – 1.49	22	100	64 – 101	1.02 – 1.49	-
TwoStage product gas	1.02 – 1.75	0 – 24	98 – 99	41 – 1331	1.35 – 1.75	0 – 12 (at λ = 1.03)
TwoStage product gas	1.00	-	36	429	1.00	-
Biogas	1.00 – 1.42	6 – 22	99 – 100	346 – 3381	1.38 – 1.42	-
Biogas	< 1	-	-	76 – 748	-	-

Table 6 – Overview of results from emission tests

Future work

Further work with the engine setup will include continued testing of the fuels, for which it is interesting to:

- Utilize pressure charging of LT-CFB gas for increased power
- Further test sub-stoichiometric operation with biogas and TWC for low NO_x
- Apply a higher compression ratio and test the gases. Especially interesting is:
    - sub-stoichiometric biogas operation
    - LT-CFB operation with varying ignition timing
    - TwoStage gas and biogas with exhaust gas recirculation

Tests with the TwoStage gasifier is planned in the future, where the gasifier will be operated in oxygen- and steam-blown operation. This will cause the heating value and hydrogen content to be significantly higher.

Flame speed and heat release measurements and calculations should be included for detailed analysis. Activities, including further testing, are planned on being carried out over the course of 2015-2016.

Milestones and status

Due to a unforeseen and time consuming approval procedure for the engine set-up (these new strict and detailed approval instruction were not enforced for research set-up's when the project proposal was developed) the project has been delayed and the expenses for build up of the more complex engine test set-up has been significantly higher than foreseen in the budget. This has significantly affected the project output.

The project specific milestones are shown in Table 7. The low compression ratio engine was tested for all gases, but did not accomplish testing for biogas with pressure charging in M5. The high compression ratio engine has been made ready for operation, but no tests have yet been accomplished - testing is planned in the future. The data analysis was originally intended to feature advanced analysis e.g. a heat release model.

	Milestone	Status
M1	Test engine installed and ready	OK
M2	Low r_c test with product gas without turbo	OK
M3	Low r_c test with product gas with turbo	OK
M4	Low r_c test with biogas without turbo	OK
M5	Low r_c test with biogas gas with turbo	Not reached, to be investigated in student projects
M6	High r_c test with product gas without turbo	Not reached, to be investigated in student projects
M7	High r_c test with product gas with turbo	Not reached, to be investigated in student projects
M8	High r_c test with biogas gas without turbo	Not reached, to be investigated in student projects
M9	High r_c test with biogas gas with turbo	Not reached, to be investigated in student projects
M10	Data analysis	OK
M11	Final reporting	OK

Table 7 – Milestones and status. r_c denotes compression ratio of engine. All tests are done with the three-way catalytic converter. Turbo denotes pressure charging

Utilization of project results

Utilizing a three-way catalytic converter on the engine exhaust has shown to be very effective in reducing CO emissions. Testing has showed CO conversion efficiencies above 98% for most operations. Thus, applying such a system at current combined heat and power engines can reduce CO emissions to a minimum. Engine systems with gasification product gas and biogas can thus be evaluated equally to natural gas with regards to CO emissions if such a system is applied.

Treatment of NO_x emissions has shown to be a compromise of air-fuel ratios, ignition timing and inert content of the gas. Product gases have a distinct advantage compared to natural gas and biogas with regards to these emissions, as the large inert fraction effectively reduce NO_x levels. Especially LT-CFB gas with 70% inert showed very low levels. Hence, exhaust gas recirculation to increase the inert of TwoStage gas might be feasible, if pressure charging and/or higher compression ratios are applied. Adjusting the ignition timing was seen to be very effective for TwoStage gas, as the optimum value was found to be at low CADBTDC, and hence lowering the NO_x significantly. Alternatively, the TwoStage gas can be operated sub-stoichiometrically for reduced NO_x.[8]

Biogas suffers from a high NO_x penalty at lower λ-values. This usually causes biogas engines to operate as natural gas engines: with very lean operation to lower the temperature. This study has however shown, that sub-stoichiometric operation with a simple TWC can reduce emission levels well below regulation. Thus pressure charging and/or higher compressions ratios can be applied for additional power for these engines.

Dissemination

The project has currently included a master thesis "Efficiency and emissions optimization of biomass CHP gas engines", by Martin Holm, 2015.

Further publications are planned based on current and future work with the engine set-up. Especially tests with the oxygen-steam-blown TwoStage gasifier is expected to facilitate a scientific article.

Summary and Conclusions

A Lister Petter natural gas engine setup with four-cylinders, 1.86 L, and spark ignition was operated with biogas and two product gases. These fuels have some characteristics that make them very interesting for operation near stoichiometric operation: large inert fraction that lowers the maximum temperature; and better combustion properties with CO and H_2 present in large quantities. Near stoichiometric the engine has a higher power density, but emissions will increase. Hence a three-way catalytic converter (TWC) is applied and emissions are measured before and after. The biofuels were tested for performance and emission levels, by varying the excess air-fuel ratio, ignition timing and boost pressure.

Performance

The engine power differed across the fuels, due to differences in gas compositions. Biogas showed the highest power output, with 20 - 30% higher output. This relatively small difference is due to the large inert fraction and low hydrocarbon content of product gases, which lowers the oxygen demand and hence causes the heating value of the fuel-air mixture to be relatively high.

As the λ was lowered, the power increased as expected. Across tests the power increased up to 20 – 25% with the excess air-fuel ratio. The electric efficiency of the engine varied between 24 – 33% for the product gases. The efficiency showed high values at low λ for the LT-CFB gas, which is opposite to natural gas operation. Thus there is a clear incitement to operate product gas engine close to stoichiometric condition: increased power and efficiency.

Studying the ignition timing, it was seen that biogas had an optimum between 20 – 22 crank angle degrees before top dead center (CADBTDC), while the TwoStage gas optimum was 5 – 12 CADBTDC. This is an interesting result, as product gas timing can be adjusted for optimization, which also lowers the emissions of NO_x, CO and unburned hydrocarbons (UHC). Ignition timing was found to affect power production less than λ, ranging between 10 – 15%.

Pressure charging of TwoStage operation showed power increases up to 80 – 110% at boost pressures 900 mbar. The resulting drop in efficiency was 5 – 7%.

Emissions
Pressure charging the engine increased both CO and NO_x emissions. Nitrogen oxides were increased significantly, as a raise of 2000 mg/nm^3 were found for 400 mbar boost pressure. Thus pressure charging needs to be used parallel with the λ, retarded ignition timing and/or high inert content of the fuel. CO levels were only increased slightly with increased pressure.

The LT-CFB product gas showed excellent NO_x performance, with values well below regulation and close to negligible. Across air-fuel ratios, the gas showed very low emissions. Namely the large inert content of 70% is thought to regulate the temperature and hence reduce emissions. The LT-CFB is because of this, very suited for pressure charging and higher compression ratios that will increase NO_x as well as power and possibly efficiency.

The TwoStage product gas showed high CO concentration at stoichiometric conditions after the TWC, but did not surpass the regulated limit. If this high-power operation is to be maintained, NO_x management will have to be managed through either sub-stoichiometric conditions (which might increase CO across the regulated limit) or by retarding the ignition timing. At a λ-value of 1.03, it was shown that retarding the ignition timing to ≤12 CADBTDC will keep NO_x within regulation. This is particular convenient, as the optimal ignition timing is found to be between 5 – 12 CADBTDC. Therefore pressure charging and/or higher compression ratios might be applicable if the ignition timing is retarded below 12 CADBTDC. Alternately, TwoStage operation can be carried out at λ-value of >1.3, <1 or apply exhaust gas recirculation to increase the inert content.

Biogas had the lowest inert content at 30% and resembles natural gas the most. NO_x emissions are high at lean operation close to stoichiometric and either very lean burn around λ 1.4 or sub-stoichiometric operation is needed. Biogas can be operated at very lean conditions to reduce NO_x and apply the TWC for effective CO removal. It was shown that retarding the ignition timing to ≤10 CADBTDC and a λ ≥1.22 can reduce NO_x below the regulated limit. UHC was also seen to decline up to 56% by retarding the ignition timing. However, retarding the ignition timing will lower the power output, which is not desired. Further testing of biogas below stoichiometry presents the biggest potential, as higher compression ratios and pressure charging can be applied.

TWC evaluation
The catalyst displays the ability to remove CO at a very high rate, above 94% across tests. This study shows that a simple and cheap TWC can easily reduce CO emissions in the large interval of operation that is applied. At sub-stoichiometric conditions, where a lot of CO is emitted, the TWC reduced levels well below the regulated limit. Hence, the catalyst can effectively be applied to engines, using biogas and product gas, to obtain CO emissions similar to known natural gas technology.

The TWC is ineffective in reducing NO_x emissions at λ>1. Across lean air-fuel ratios and ignition timings, the conversion efficiency is negligible. However, at sub-stoichiometric conditions the NO_x emissions were reduced to a minimum, because of the reduced oxygen availability and assumedly also because of increased catalyst activity. This will however need further testing in order to be validated.

Hydrocarbon conversion in the TWC is seen to be close to negligible. As a function of λ, there is a weak correlation towards increasing conversion at higher λ-values. By varying the ignition timing, the emissions show that the conversion efficiency fluctuates around 0%. Thus, the catalyst is not thought to promote any significant UHC oxidation.

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[1] Ahrenfeldt, J., et al., *Development and test of a new concept for biomass producer gas engines*, Risø report, 2010

Compression ratio engine operation on biomass producer gas

[2] Ahrenfeldt, J. & Henriksen, U. B., *High compression ratio engine operation on biomass producer gas*, Society of automotive engineers of Japan, 2011

[3] Table values taken from Ahrenfeldt, J., *Characterization of biomass producer gas as fuel for stationary gas engines in combined heat and power production*, Ph.D. thesis, 2007.

[4] Farrauto, RJ. & Heck, RM., *Catalytic converters: state of the art and perspectives*, Catalysis today 51, p. 351-360, 1999.

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Technical University of Denmark

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Efficiency and Emissions Optimization of Biomass CHP Gas Engines

Master's thesis, Biomass Gasification Group
March 2015

DTU Chemical Engineering
Department of Chemical and Biochemical Engineering

Abstract

Biomass gasification in combination with gas engines is an interesting technology for small scale combined heat and power (CHP) plants. The Viking gasifier produces a gas highly suitable for engine operation.

Emission regulations provide motivation for optimising engine operation to achieve low emissions with the highest possible efficiency and power output. At the same time biomass producer gas has a high knocking resistance, making stoichiometric operation feasible.

In this project experiments have been conducted with a laboratory scale CHP gas engine setup operating on producer gas from the Viking gasifier. The performance and emissions were mapped with respect to air-fuel ratio, ignition timing and pressure charging and the effect of a three-way catalytic converter (TWC) as an exhaust after treatment device was investigated.

The capabilities of this setup in terms of air-fuel ratio control showed to be insufficient to maintain a stable stoichiometric operation making it difficult to obtain good results of emission conversion in the TWC.

The results demonstrated lean operation with low NO_x emissions and good CO conversion in the TWC (94-99%). Near stoichiometric operation NO_x emission was significantly lowered by retarding ignition timing without apparent loss in power or efficiency. Potential of both CO and NO_x reduction in the TWC was observed, but the air-fuel ratio was never stable enough to achieve good results.

Resumé

Biomasseforgasning i kombination med gasmotorer er en interessant teknologi for mindre decentrale kraftvarmeværker. Viking forgasseren producerer en gas, der er meget velegnet til motordrift.

Emissionsrestriktioner lægger grund til optimering af motordrift med henblik på at opnå lave emissioner med højest mulige virkningsgrad og effekt. Samtidig har produktgas fra biomasseforgasning en højere tolerance med hensyn til motorbankning, hvilket muliggør støkiometrisk motordrift.

I dette projekt er der udført eksperimenter med et gasmotor setup til kraftvarmeproduktion med produktgas fra VIking forgasseren som brændstof. Ydelse og emissioner er blevet kortlagt med hensyn til luft-brændstof forhold, tændingstiming og trykladning, og effekten af en trevejskatalysator er undersøgt.

Dette setups evne til at kontrollere luft-brændstof forholdet viste sig ikke at være tilstrækkelig til at holde en stabil støkiometrisk drifttilstand, hvilket gjorde det svært at opnå retvisende resultater for konvertering af emissioner i trevejskatalysatoren.

Resultaterne viste lav NO_x emission ved mager drift og god konvertering af CO i trevejskatalysatoren (94-99%). Ved nær støkiometrisk drift kunne NO_x emissionen formindskes betydeligt ved at forsinkr tændingen uden synligt tab af virkningsgrad eller effekt. Potentiale af både CO- og NO_x-reduktion kunne observeres, men luftbrændstof forholdet var aldrig stabilt nok til at opnå gode resultater.

Preface

This report describes the work and results of a project carried out in the period from August 15th 2014 to March 8th 2015 as the final assignment of a MSc program at the Technical University of Denmark (DTU). The project is done at the CHEC Research Centre under Department of Chemical and Biochemical Engineering with supervisors Senior Scientist Jesper Ahrenfeldt and Senior Researcher Ulrik Birk Henriksen and external supervisor Torben Kvist from Dansk Gasteknisk Center (DGC).

I would like to thank my supervisors for providing valuable knowledge and assistance whenever needed. I would also like to thank technicians Erik Hansen, Kristian Estrup and Freddy Christensen for their big contribution to the practical work in this project and the rest of the staff for creating a good working atmosphere.

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1 Introduction

Biomass plays an important role in the conversion from a fossil based to a sustainable and CO_2-neutral energy system. It is the single largest source of renewable energy today and is expected to play an even bigger role in the future power and heat generation[1].

Biomass gasification opens up opportunities for utilisation of biomass in a wider range of combined heat and power (CHP) processes by converting solid biomass into a combustible gas termed producer gas (PG).

Biomass gasification in combination with a gas engine operating on the producer gas is an interesting prospect for small-scale CHP plants. The fact that it is possible to convert existing natural gas fuelled gas engine CHP plants makes it very attractive.

Biomass producer gas engines are based on technology developed for natural gas, and are still subject to optimisation with regards to performance and emissions. Biomass PG is a fundamentally different fuel compared to natural gas (NG) and offers opportunities for different operation characteristics.

When optimising CHP engines the goal is to achieve a high thermal efficiency and thus low specific fuel consumption and maintaining a high power density, while keeping emission below the regulated limits.

The dominating approach for gas engines is lean burn operation with turbocharging. This strategy has been widely used and optimised. It has the advantage of low NO_x emission and high efficiency. The drawbacks are higher emissions from unburned fuel and poor exhaust gas condensation.

An alternative approach is near stoichiometric operation with a three-way catalytic converter. The lowest emissions levels can be achieved with this strategy as seen from the automotive applications. However, engine knock at these conditions is a limiting factor for compression ratio and turbocharging leading to lower power density and efficiency.

Biomass producer gas as a fuel has a higher knocking resistance due to low hydrocarbon content and high CO_2 content. Application of this strategy to biomass producer gas engines is therefore interesting to investigate.

1.1 Objectives

This project is based around a laboratory scale CHP engine test setup. The goals are to investigate the performance and emissions with application of three-way catalytic converter and pressure charging at different operating conditions with biomass producer gas from the Viking gasification plant as fuel.

- Present relevant theoretical background for biomass producer gas operated engines based on the literature.
- Perform the necessary practical and technical tasks to prepare the test setup.
- Experimental investigation of performance and emissions and feasibility of operation of gas engines near stoichiometric condition with three-way catalytic converter

2 Background

This section describes the fundamental concepts and background knowledge relevant to the experiments and analyses conducted in this project with the CHP engine operating on producer gas from the Viking gasification plant.

2.1 Thermal gasification

Thermal gasification is a thermochemical process in which solid carbonaceous matter is converted to a combustible gas. It can be divided into two steps, pyrolysis and gasification. During pyrolysis the fuel is heated and volatile components including tars are released and solid carbon (char) and inorganic matter (ashes) are left.

Gasification is the process where char is converted to gases by addition of a gasification agent, typically CO_2, H_2O or O_2.

The produced gas is called producer gas and consist mainly of CO, H2, CO_2, CH_4, tars, N_2 and H_2O. Tars are higher hydrocarbons that are condensible. For producer gas applications such as gas engines or fuel cells none or very low concentrations of tars and particles are allowed, whereas if the producer gas is to be combusted tars are not a problem, but inorganic components must be minimised [2].

Different types of fuels, reactors (fixed bed, fluid bed, entrained flow), operating temperatures and gas cleaning systems influence the gas quality and cold gas efficiency of a gasification plant.

2.1.1 Viking gasifier

The Viking gasifier is a two-stage down draft gasifier developed at the biomass gasification group at DTU. The two-stage process implies pyrolysis and gasification processes in two separate reactors with an intermediate-high tar cracking zone. The process is stable and easy to control and yields a high cold gas efficiency ($\approx 93\%$) and a very low tar content in the gas (< 1 mg/Nm^3) making it suitable for applications requiring a clean gas [3, 4].

The technology has been tested and validated at different scales. A fully automated 70 kW plant was built in 2002 for research and demonstration purposes. It is coupled with a gas engine-generator set that uses the produced gas to produce power to the grid. Table 1 shows the main parameters of the Viking demonstration plant.

Figure 1 shows a process flow diagram of the Viking gasifier. The biomass (wood chips) are fed to the pyrolysis reactor where it is transported through by a screw conveyor. The reactor is heated by exhaust from the engine in order to dry and pyrolyse the biomass.

The pyrolysis products are led to the gasification reactor where the gases are first partially oxidised to produce heat for the gasification process and then led through the char bed where the char is gasified.

The hot producer gas leaves the gasification reactor and goes through a system of heat exchangers and gas conditioning to heat up a part stream of the engine exhaust and the gasification air and further remove particles and tar.

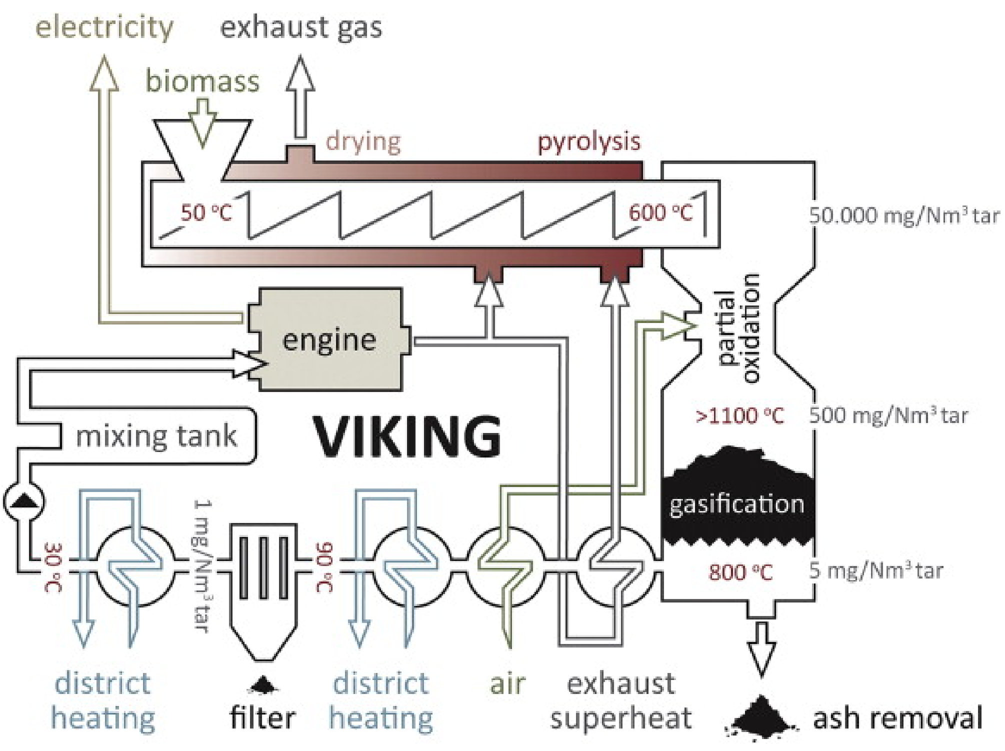


Figure 1: *Process flow diagram of the Viking gasifier [5].*

Table 1: *Key specifications of the Viking demonstration plant [3, 6].*

Description	Value	Unit
Feedstock	Wood chips	[-]
Moisture content	35-45	[%]
Thermal input	70	[kW]
Power output	17.5	[kW]
Thermal output	39	[kW]
Gasifier efficiency	93	[%]
Engine-generator electric efficiency	29	[%]
Overall electric efficiency	25	[%]f

Table 1 shows the specifications of the Viking demonstration plant and Table 2 lists an example of average composition of the Viking producer gas.

Table 2: *Viking producer gas composition [6].*

Component	Value	Unit
H_2	30	[Vol. %]
CO	20	[Vol. %]
CO_2	15	[Vol. %]
CH_4	1	[Vol. %]
N_2	34	[Vol. %]
LHV	6	[MJ/Nm^3_{dry}]

2.2 Internal combustion engines

An internal combustion engine (ICE) is essentially a heat engine that serves to produce mechanical work from chemically bound energy in the fuel. The energy is released by combusting the fuel inside the engine and work is transferred directly from the high pressure combustion product to mechanical components of the engine.

ICE's is an old and well proven technology and numerous engine designs and configurations have been developed over the years such as reciprocating/rotary engines, different cylinder arrangements, working cycles, fuel utilisation, fuel addition systems etc.

The two most common types are spark-ignition (SI) engines and compression-ignition (CI) engines. They are used in many different applications in transportation, domestic use, industry and power and heat generation, because of their stability, flexibility and good power-to-weight ratio.

The following will mainly focus on reciprocating four-stroke SI engines with special attention to producer gas operated engines.

2.2.1 Fundamentals

In a four-stroke cycle the piston makes four strokes (two revolutions of the crankshaft) per cycle yielding one power stroke. Figure 2 illustrates the process. Below the process of the four strokes is outlined.

(a) Intake stroke - Inlet valve is open and the piston moves from top dead center (TDC) to bottom dead center (BDC) and draws in air-fuel mixture.

(b) Compression stroke - Valves are closed and the piston moves up and compresses the air-fuel mixture. Shortly before the piston reaches TDC the compressed charge is ignited by the spark plug and combustion initiates.

(c) Expansion stroke - The high pressure caused by combustion pushes the piston down and power is transferred to the shaft.

(d) Exhaust stroke - The exhaust valve is open and the piston moves up pushing out the exhaust gases.

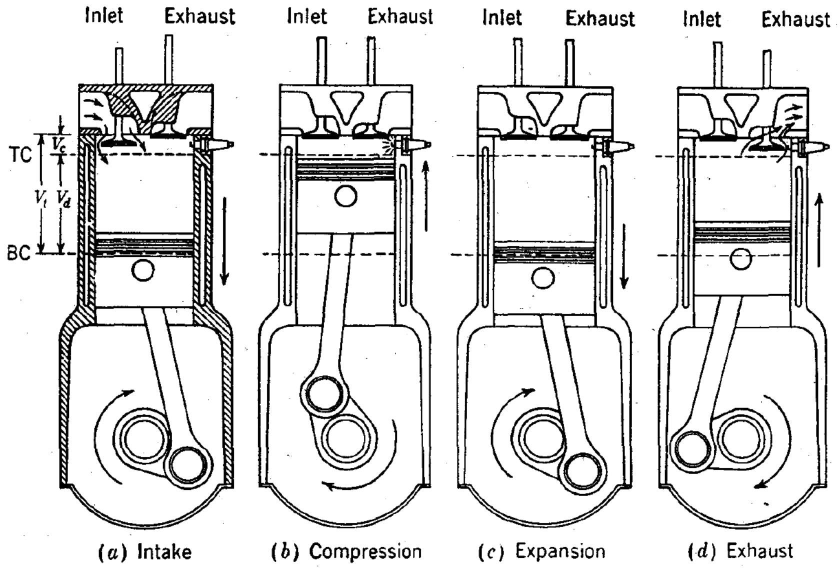


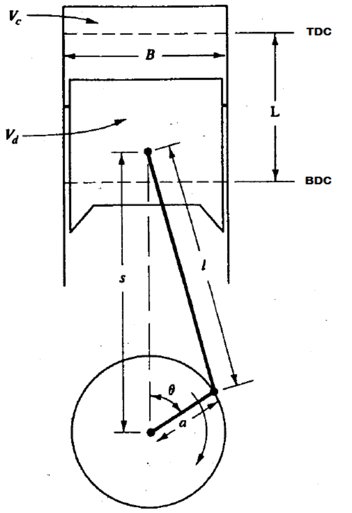
Figure 2: *The four steps in a four-stroke cycle. Illustration from [7]*

Figure 3a and Table 3 shows the most important geometries of an engine cylinder, piston and crank mechanism. The displacement volume (V_d) and compression ratio (ϵ) are given by:

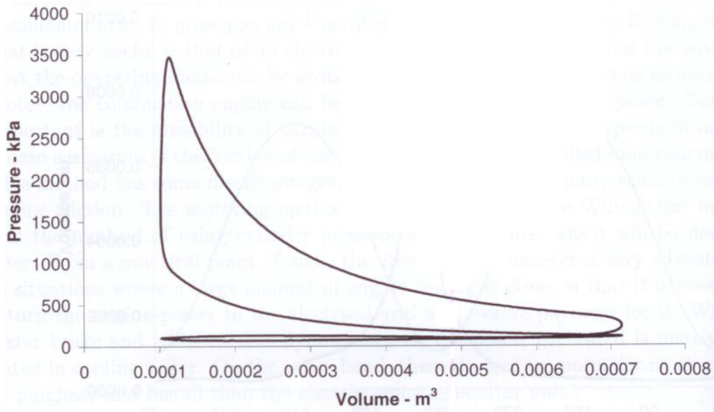
$$V_d = V_c + \frac{\pi}{4}B^2L \tag{1}$$

$$\epsilon = \frac{V_d + V_c}{V_c} \tag{2}$$

Figure 3b shows a p-V diagram for one SI engine combustion cycle where the cylinder pressure is plotted as a function of the volume. The work exerted on the piston (indicated work W_i) is the integral around this curve eq. (3). The indicated power (P_i), brake power (P_m), mechanical efficiency (η_m), indicated efficiency (η_i) thermal efficiency (η_e) and thermal input ($\dot{Q}_{in}$) are defined in eq. (4)-(9) on a per cylinder basis



(a) *Engine geometries [7].*



(b) *p-V diagram of four-stroke SI engine cycle [8].*

Figure 3

$$W_i = \oint p \cdot \mathrm{d}V \tag{3}$$

$$P_i = \frac{W_i N}{2} \tag{4}$$

$$P_b = P_i - P_f \tag{5}$$

$$\eta_m = \frac{P_b}{P_i} \tag{6}$$

$$\eta_i = \frac{P_i}{\dot{Q}_{in}} \tag{7}$$

$$\eta_e = \frac{P_b}{\dot{Q}_{in}} \tag{8}$$

$$\dot{Q}_{in} = \dot{m}_f \cdot LHV \tag{9}$$

where N is the engine speed in [rev/s], P_f is the friction power, $\dot{m}_f$ is the fuel flow and LHV is the lower heating value of the fuel. From analysis of the ideal Otto cycle it can be shown that η_i increases with ϵ [8]:

$$\eta_i = 1 - \left(\frac{1}{\epsilon}\right)^{\gamma-1}, \qquad \gamma = c_p/c_v \tag{10}$$

where c_p and c_v are the specific heat at constant pressure and volume of the cylinder gases respectively.

The electric efficiency is given by the measured generator power output (P) per unit thermal input effect ($\dot{Q}_{in}$)

$$\eta_p = \frac{P}{\dot{Q}_{in}} = \frac{P}{\dot{m}_f LHV} \tag{11}$$

Table 3: *Engine parameters.*

Parameter	Symbol
Cylinder bore	B
Stroke	L
Connecting rod length	l
Crank radius	a
Crank angle	θ
Clearance volume	V_c
Displacement volume	V_d

The air-fuel ratio (AF) of the combustion mixture is an important parameter for performance and emissions. It is defined as:

$$AF = \frac{m_a}{m_f} = \frac{\dot{m}_a}{\dot{m}_f} \tag{12}$$

Stoichiometric combustion means that there is exactly enough oxygen for complete combustion of the fuel. The reaction equation for stoichiometric combustion of an organic fuel with air is:

$$C_xH_yO_z + \left(x + \frac{y}{4} - \frac{z}{2}\right)(O_2 + 3.76N_2) \longrightarrow \; xCO_2 + \frac{y}{2}H_2O + \left(x + \frac{y}{4} - \frac{z}{2}\right)3.76N_2 \tag{13}$$

The stoichiometric air-fuel ratio (AF_{st}) is then calculated as:

$$AF_{st} = \frac{\left(x + \frac{y}{4} - \frac{z}{2}\right) \cdot 4.76 \cdot M_{air}}{M_{fuel}} \tag{14}$$

where M_{air} and M_{fuel} are the molar weights of air and the fuel respectively. AF_{st} is dependant on fuel compositions, hence the ratio of AF to AF_{st} (excess air equivalence ratio λ) is a more convenient parameter to use. It is defined as:

$$\lambda = \frac{AF}{AF_{st}} \tag{15}$$

For $\lambda > 1$ the mixture is lean (excess air) and for $\lambda < 1$ the mixture is rich (excess fuel.

2.2.2 Fuels

The fuel used in an engine has a big influence on operation, performance and emissions. Engines are therefore designed and optimised for a specific fuel type. However, producer gas engines as a technology in development are based on natural gas engines because the fundamental operation is similar.

Natural gas consists mainly of methane, while producer gas has a completely different composition (e.g. Viking PG Table 2) and different properties as an engine fuel. Table 4 compares natural gas and Viking producer gas in terms of LHV. Producer gas has a much lower LHV, but the stoichiometric air/fuel ratio is also lower meaning that the LHV of the stoichiometric mix (LHV_{st}) is only 20% below that of natural gas.

Table 4: *Comparison of Viking producer gas and natural gas [6].*

	Viking PG	NG	Unit
LHV	6	39	[MJ/Nm^3_{dry}]
AF_{st}	1.3	10.6	[Nm^3/Nm^3]
LHV_{st}	2.6	3.4	[MJ/Nm^3_{stmix}]

Another important parameter for a fuel is its knocking resistance. Knocking is caused by spontaneous ignition of parts of the premixed fuel charge due to high pressure and temperature causing very high pressures that can damage the engine. Knocking limits the compression ratio and pressure charging and thus efficiency and power density.

Producer gas has a very high knocking resistance compared to natural gas, because of the low hydrocarbon and high CO_2 content. This allows for operation with higher compression ratio (comparable to Diesel engines), higher pressure charging and stoichiometric operation.

A problem with producer gas as engine fuel has been high tar content. Tars can be a critical problem for engine operation as it can cause blockage and form deposits. The extremely low tar content in the Viking producer gas makes it suitable as an engine fuel. No significant problems were observed on the Viking gasifier engine after long term operation. [6].

2.2.3 Exhaust emissions

Exhaust emissions from ICE's contribute to air pollution and can have a negative environmental impact and adverse human health effects.

Many different factors have a say in the nature and amount of emissions formed during combustion from ICE's such as engine type, fuel, combustion chamber design, ignition timing, λ etc.

The emissions types focused on in this study are unburned hydrocarbons (UHC), carbon monoxide (CO), nitrogen oxides (NO_x)

2.2.3.1 Unburned hydrocarbons

Unburned hydrocarbons (UHC) originate from hydrocarbons in the fuel that pass through the engine unburned or partially burned. Several mechanisms are relevant to the formation of UHC [8].

Flame quenching - When the flame approaches the colder surfaces of the combustion chamber it is extinguished and HC near these surfaces remain unburned. However, most of these will subsequently mix with the still hot combustion gases and oxidise.

Crevices - Small volumes in the combustion chamber where unburned mixture enters under high pressure, but the flame cannot penetrate due to narrow entrance e.g. between piston, piston ring and cylinder wall. When the pressure decreases due to expansion the gases are released. Some HC will oxidise depending on the temperature of the combustion products.

Absorption and desorption - Oil film and deposits in the combustion chamber can absorb fuel vapour. During the expansion and exhaust stroke it is desorbed into the combustion products as the fuel vapour concentration here is low. Fuel vapour that is desorbed late in the cycle when the temperature is lower may add to UHC.

Bulk quenching - Due to rapid expansion and hence cooling of the unburned mixture flame extinction can occur before it has reached the whole volume. This mechanism can occur under certain operating conditions e.g. excessive dilution with exhaust gas recirculation (EGR) and near the lean combustion limit. Bulk quenching results in very high UHC concentration.

For engines operating on producer gas UHC is insignificant since the only HC in producer gas is small amounts of methane (CH_4). Instead the high CO content in the fuel can lead to unburned fuel-CO (UCO).

2.2.3.2 Carbon monoxide

For conventional HC based fuels CO emissions in the exhaust come from partial oxidation of HC in the fuel. The excess air equivalence ratio (λ) is the main influence factor of CO emission. For rich conditions CO emission increase rapidly with decreasing λ due to insufficient oxygen to complete combustion. At lean conditions low temperature can in some cases cause CO to be formed.

As mentioned above it is a different case for engines operating on producer gas. The low HC content in the fuel means that only little CO emissions originate from partial oxidation hereof. CO is present in the fuel and CO emissions will mainly originate from UCO. The UCO formation mechanisms are more or less the same as for UHC described above.

As with HC based fuels high CO emission is seen at rich conditions, but not so much due to partial oxidation of HC, but because the fuel-CO passes the combustion chamber unburned.

2.2.3.3 Nitrogen oxides

NO_x emission is a mixture of nitric oxide (NO) and nitrogen dioxide (NO_2). NO which is formed during combustion, account for the most of NO_x emissions, while NO_2 is formed by later oxidation of NO.

NO is formed by a reaction between nitrogen, usually from the combustion air, and oxygen

$$N_2 + O_2 \rightleftarrows 2NO \tag{16}$$

The reaction rate constant is strongly dependant on temperature. So higher temperature means that more NO_x will be produced as long as enough nitrogen and oxygen is available. The highest NO_x emissions will be slightly lean of stoichiometric condition where the temperature is highest and lower at leaner condition and richer condition where also oxygen is a limiting factor.

The ignition timing influences the maximum pressure and temperature during combustion and thus also NO_x emission. retarding the ignition timing towards TDC lowers the maximum pressure and temperature because a larger part of the combustion duration take place after TDC during expansion.

Some fuels including producer gas may contain nitrogenous compounds such as ammonia (NH_3), which when combusted lead to fuel-NO_x which may constitute a significant amount of the total NO_x emissions [6].

2.2.4 Emissions regulation

The current regulated limits in Denmark for stationary gas engines are shown in Table 5. The CO emission limit for producer gas is significantly higher than for natural gas. In many countries (including DK until recent years) the CO emission limit for producer gas engines is that given for natural gas. These regulations are based on CO emission formed by partial oxidation of HC, however, as described previously CO emission formation mechanisms for producer gas engines

(CO containing fuel) are completely different making it difficult to comply with these regulations [9].

The values of emission concentrations are standardised according to guidelines from the Danish Environmental Protection Agency [10] to be expressed in milligrams per normal cubic meter at a reference oxygen content of 5 vol.% [mg/Nm^3 @ 5% O_2]. Conversion of CO and NO_x are given by

- 1 ppm CO = 1.25 [mg/Nm^3] CO (at 273 K, 101,3 kPa)
- 1 ppm NO_2 = 2.05 [mg/Nm^3] NO_2 (at 273 K, 101,3 kPa)

NO_x is calculated based on the mass of NO_2.

$$C_{ref.} = \frac{21 - O_2\%_{(ref.)}}{21 - O_2\%_{(meas.)}} C_{meas.} \tag{17}$$

where $C_{ref.}$ is emission concentration at the reference O_2 content, $C_{meas.}$ is measured emission concentration, $O_2\%_{(ref.)}$ is reference O_2 content $O_2\%_{(meas.)}$ measured O_2 content.

Table 5: *Danish regulated limits for NO_x and CO emissions for new stationary engines with a capacity of 0.12-5 MW_{th} [11]. Values are converted from [mg/Nm^3 @ 15% O_2] listed in the reference to [mg/Nm^3 @ 5% O_2].*

Fuel	NO_x	CO
Natural gas	507	507
Producer gas	507	3000
Biogas	507	1200

2.2.5 Emissions control

Increasingly stringent emissions regulations calls for constant improvement of emission control. The aim of emission control is to keep emissions below the regulated limit set by the authorities for the application in question, while keeping investment and O&M costs as low as possible, for which high power density and efficiency are the key factors.

2.2.5.1 Lean burn and stoichiometric operation

The excess air equivalence ratio is an important factor regarding emissions as described earlier. Lean burn operation has the advantage of low NO_x emission and high thermal efficiency, but with higher emission from unburned fuel. Practically all stationary gas engines utilise this operation mode often in combination with turbocharging to get a higher power density.

Stoichiometric operation mode with a three-way catalytic converter has the potential of very low overall emissions as it is seen from automotive SI engines. For natural gas engines additional means are required to reduce engine knock if a high power density and efficiency is to be maintained. As mentioned producer gas has a high knocking resistance and this operation mode is interesting for producer gas engines.

2.2.5.2 Exhaust gas recirculation

Exhaust gas recirculation (EGR) is a method used to control NO_x emissions. A part of the exhaust gases are reintroduced to the the intake manifold and mixed with the unburned mixture. The inert combustion gases dilute the unburned mixture and lower the flame temperature without altering the λ value.

The advantages of this when used in proper amounts are lower NO_x emissions, higher thermal efficiency due to lower heat loss and better knock resistance (higher ϵ and/and boost pressure possible). The use of EGR can also cause immediate disadvantages such as unstable combustion due to slower burn rate and an increase in unburned fuel and lower power density of the engine. However, this is often compensated for and even improved by the higher possible ϵ and/and boost pressure.

2.2.5.3 Three-way Catalytic Converter

Catalytic converters are devices used in engine exhaust systems to remove pollutants from the exhaust gas. They utilise catalytic materials to enhance reactions converting harmful emissions in the exhaust into harmless components such as CO_2, H_2O and N_2.

Oxidation catalysts are used to oxidise UHC and CO to CO_2 and H_2O. A surplus of oxygen must therefore be present in the exhaust either by operating the engine in lean condition or introducing air into the exhaust.

A three-way catalysts (TWC) is, like an oxidation catalyst, capable of oxidising HC and CO, while simultaneously reducing NO_x. This requires the exhaust stream to be maintained close to stoichiometric ratio. TWC's can thus only be used with SI engines since CI engines always operate under lean conditions. The following will focus on the characteristics and functionality of the TWC.

Structure and materials

The TWC is made of a ceramic or metallic monolith (honeycomb) support structure with a thin layer of catalysed washcoat deposited on the walls, all held inside a metal housing (see Figure 4). The exhaust stream is led through parallel channels (cells) in the monolith structure which offers good mass transfer and low pressure drop compared to previous bead bed designs. With modern technology cell densities up to 1200 cells per square inch and an open frontal area close to 90% can be achieved [12].

The washcoat is constituted of a carrier, typically alumina (Al_2O_3), which is a porous material with an internal surface area of 100-200 m^2/g, impregnated with the active catalytic materials [7].

The main active catalytic materials used are noble-metals; platinum (Pt), palladium (Pd) and rhodium (Rd). Pt is mainly active in CO and HC oxidation and Rd in NO_x reduction, while Pd mainly is used to substitute Pt and/or Rd as a means of cost reduction.

Operating exactly at stoichiometric condition all the time is not practically feasible. In reality the engine will oscillate between slightly rich and lean of stoichiometric. Therefore oxygen storage components (OSC) such as CeO_2 and ZrO_2 is often used. These components are able to store oxygen when running lean and release it when running rich [13].

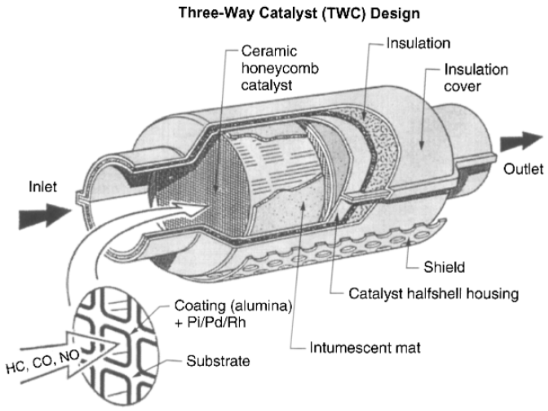


Figure 4: *Schematic of typical auto catalyst design [12].*

Reactions

When operating near stoichiometric condition there will be enough reducing gases to reduce NO_x and enough oxygen to oxidise CO and HC. These are the main reactions taking place in a TWC [14]

Oxidation

$$CO + \frac{1}{2}O_2 \longrightarrow CO_2 \tag{18}$$

$$H_2 + \frac{1}{2}O_2 \longrightarrow H_2O \tag{19}$$

$$C_xH_y + \left(x + \frac{y}{4}\right) O_2 \longrightarrow xCO_2 + \frac{y}{2}H_2O \tag{20}$$

NO_x reduction

$$2CO + 2NO \longrightarrow 2CO_2 + N_2 \tag{21}$$

$$C_xH_y + \left(2x + \frac{y}{2}\right) NO \longrightarrow xCO_2 + \frac{y}{2}H_2O + \left(x + \frac{y}{4}\right) N_2 \tag{22}$$

$$H_2 + NO \longrightarrow H_2O + \frac{1}{2}N_2 \tag{23}$$

Steam reforming

$$C_xH_y + xH_2O \longrightarrow xCO + \left(x + \frac{y}{2}\right) H_2 \tag{24}$$

Water-gas shift

$$CO + H_2O \longrightarrow CO_2 + H_2 \tag{25}$$

Air-fuel ratio

As mentioned above the engine must be operated at an AF close to stoichiometric in order to efficiently remove all three pollutants. However, the window of operation greatly depends on the catalyst state and formulation, engine operating conditions and fuel type. Figure 5 shows the efficiency of NO_x, HC and CO removal as a function of AF for a gasoline engine. It is suggested in [7] that the window is about 0.1 AF units wide, which is $\approx 7 \times 10^{-3}$ in terms of λ, while other sources such as in Figure 5 show up to 1.5 AF $\approx 1 \times 10^{-2}$ in terms of λ.

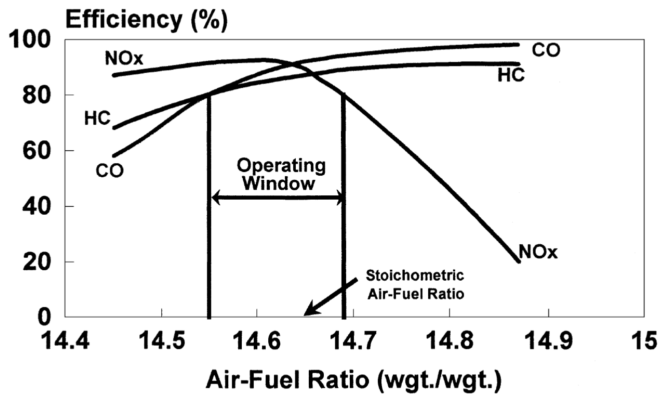
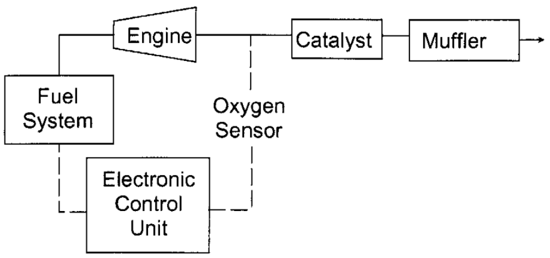


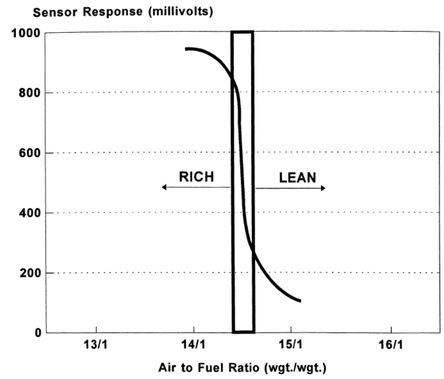
Figure 5: *Example of TWC conversion efficiency as a function of AF. [13]*

Control

In order to keep λ within the desired window either a sophisticated carburetor or injection system with closed-loop control must be implemented [7]. An oxygen sensor located in the exhaust stream immediately before the TWC continuously sends a signal that indicates whether the engine is running rich or lean to a control unit that that adjusts the fuel addition accordingly (Figure 6a).



(a) *Schematic of closed-loop AFR control system [12].*



(b) *Oxygen sensor response [13].*

Figure 6

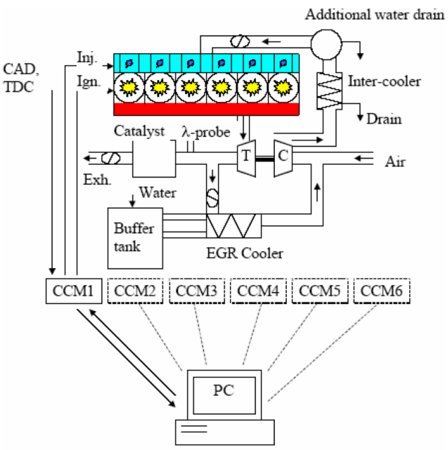
Figure 6b shows the voltage response of an oxygen sensor typically used in AF control systems. The frequency of the signal and time lag in the electronic system results in a λ perturbation around the setpoint, with a frequency and amplitude. Generally higher frequency and lower amplitude yields higher conversion rates in the TWC. The nature of this perturbation also has an impact on how wide the acceptable λ window is [12].

Studies with gas engines and TWC

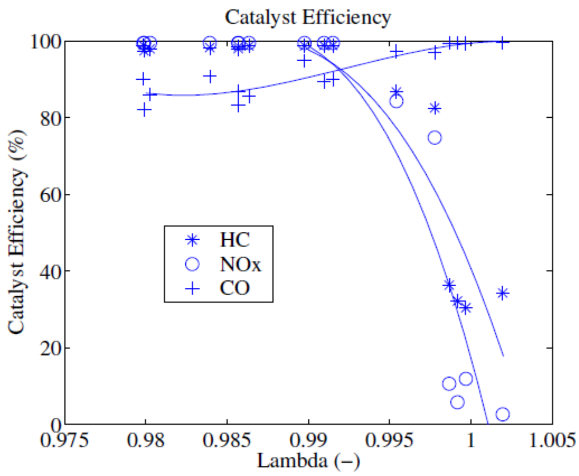
The vast majority of the literature concerning TWC's are based on gasoline operation. The performance of the TWC greatly depends on the properties and components in the exhaust and thus the fuel utilised. No reported studies of stoichiometric operation with producer gas engines has been found, but a few exist for natural gas.

Studies of a test setup of a natural gas engine with stoichiometric operation and TWC with EGR and turbocharging has been carried out at Lund Institute of Technology (LTH) [15, 16].

Figure 7a shows a schematic of their test setup. They use a closed-loop control system with port injection. EGR is also used for improved knocking resistance, efficiency and NO_x reduction. They do not use a binary lambda probe with an oscillating system. Instead they keep the lambda value almost constant (using a wider range lambda probe). This means that the oxygen storage ability of the TWC is not used resulting in a sensitive NO_x/CO conversion trade-off and thus a narrower window of operation. They achieved 99.9% NO_x reduction and 90-97% HC and oxidation, while keeping $\lambda \pm 0.01$ of optimum (see Figure 7b). The operation was optimised for NO_x reduction which meant that higher CO emission was seen.



(a) *Schematic of test setup at LTH [16]*



(b) *Catalyst efficiency as a function of λ with 30% EGR [15].*

Figure 7

At ETH Zurich a similar study has been made. They have successfully optimised a natural gas CHP engine for stoichiometric operation with TWC, EGR and turbocharging [17]. Figure 8 shows a schematic of their test setup. Not many details are provided, but it shows that they use a carburetor system.

They obtain engine efficiencies of 40-42% and NO_x emission levels <2 mg/Nm^3 @ 5% O_2 and CO and UHC emission levels <75 mg/Nm^3 @ 5% O_2.

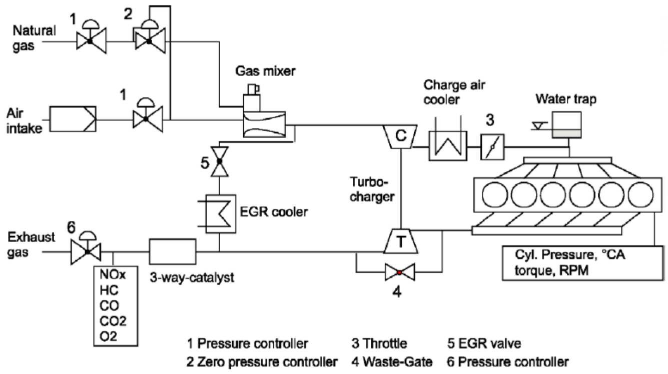


Figure 8: *Schematic of test setup at ETH Zurich [17].*

2.2.6 Pressure charging

The main purpose of pressure charging is to increase the power output for a given size engine. This is done be pressurising the intake air or mixture with a compressor, hence increasing the density allowing more fuel to be introduced to the cylinder per cycle resulting in power increase.

The two most common ways of achieving this is supercharging and turbocharging. A supercharger is a compressor mechanically driven by the engine shaft. A turbocharger uses the hot exhaust gases to drive a turbine connected to a compressor. A turbocharger has the advantage that no power is consumed from the engine shaft.

Besides increasing the power output pressure charging may improve the efficiency of the engine. This is due to a higher mechanical efficiency, since power increases without increasing the friction significantly [8]. The increased pressure can also improve the combustion.

When the intake air or mixture is compressed the temperature increases, which works against increasing the inlet density. Therefore the air or mixture is often cooled between the compressor and engine using an intercooler.

3 Experimental

This section describes the experimental aspects related to this project including a description of the test setup, measurements, an overview of the practical work in preparing the test setup, operation description and overview of the tests carried out.

3.1 Test setup

The test setup is built as a lab scale CHP engine system with the purpose of testing system performance and emissions with gaseous fuels e.g. producer gas. The setup is located at DTU Risø Campus outside building 321. It is however conveniently built as a transportable unit in a 20ft container so that live tests can be carried out in combination with fuel producing plants at different locations.

On the outside there is a number of inputs and outputs to and from external sources required for operation. The inside of the container is designed with separate engine- and control rooms housing all the main components (see Figure 9).

The engine room contains the engine with inlet and exhaust system, generator, cooling system and the main control panel. The control room contains the secondary control panel, gas- and emissions analysis equipment and computers for control and DAQ. Figure 10 shows the PI-diagram for the setup. Below the main components, subsystems and operation principles are described.

3.1.1 Intake and exhaust systems

The intake system is designed to accommodate two different fuel operating modes, NG and PG, and to operate the engine either naturally aspirated or with forced induction. It consists of an air intake and two gas trains, one for NG and one for PG.

The pressure is reduced by a a constant pressure regulator in the NG gas train to just above atmospheric pressure and added through a simple carburetor. The air passes through a venturi tube. This creates a pressure difference dependant on the volume flow. NG is sucked through small holes in the venturi due to the

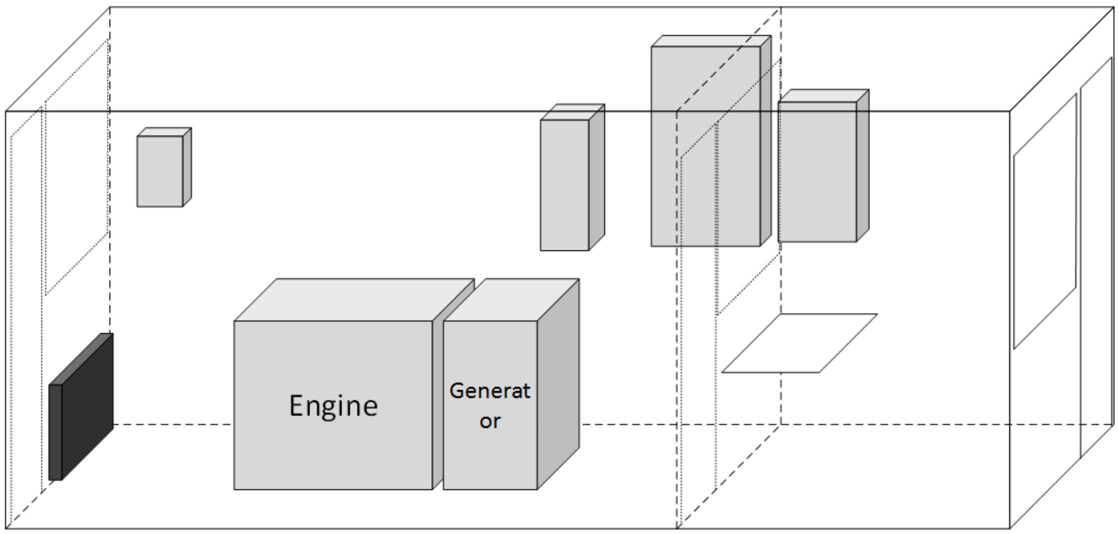


Figure 9: *Schematic of container housing the test setup.*

pressure difference. In order to be able to reduce the λ value in NG mode a throttle in the form of a sliding valve is fitted on the air intake before the venturi.

PG is added through a T-piece after the venturi. A control valve on the PG train adjusts the amount of fuel added by feedback from an O_2 sensor in the exhaust and a set value. In both modes the air-fuel mixture is led through a mixing device to ensure a homogeneous mixture.

A compressor powered by an electro-motor is installed between the mixing device and the engine in order to be able to provide a boost pressure. The power consumed by the compressor is taken directly from the power produced by the generator. A parallel bypass line is used when the engine is naturally aspirated. When the compressor produces a boost pressure the intake charge is automatically sucked through the compressor and a one-way valve in the bypass line prevents backflow.

The exhaust system consists of a pipe leading the exhaust through a TWC and a silencer before discharged to the surroundings. Sampling taps are placed before and after the TWC to make it possible to measure emissions from either.

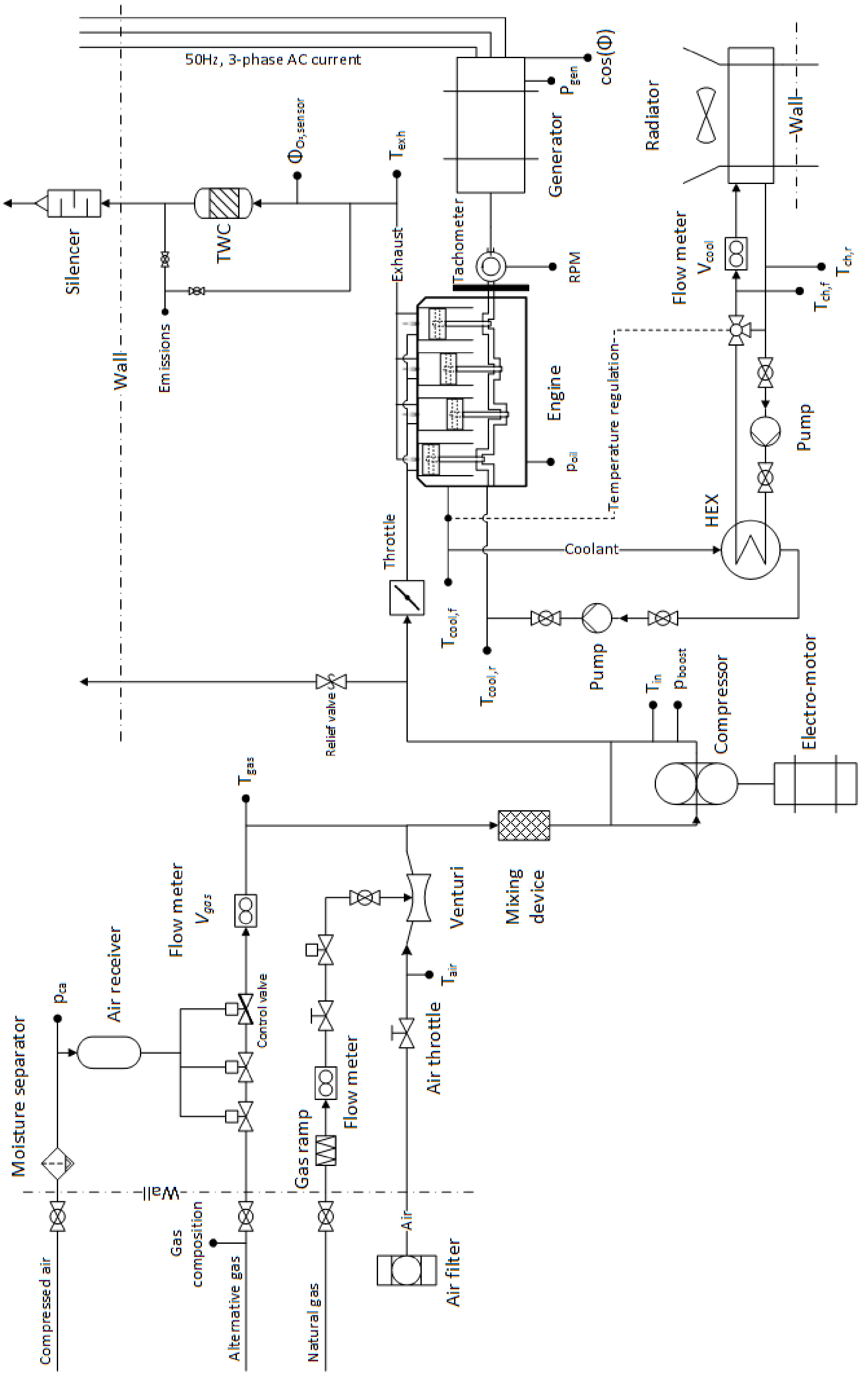


Figure 10: *PI-diagram of the gas engine setup.*

3.1.2 Engines

The engine used with the test setup in this project is a natural gas engine with a compression ratio of 9.5:1. A second engine is available to be used with the same setup. It is Diesel engine with a compression ratio of 18.5:1 rebuilt to gas operation. Both are four-stroke, four-cylinder 1.86 L SI engines produced by Lister Petter.

3.1.3 Generator

The generator used in the setup is a Leroy Somer four-poled synchronous generator with a nominal effect of 19.2 kW. When synced to the 50Hz grid the gen-set runs at 1500 rpm.

3.1.4 Cooling system

The cooling system is made to simulate a central heating system (CHS) in order to be able to calculate the overall system efficiency (power + heat). It consists of an engine cooling circuit and a CHS circuit connected by a heat exchanger. A three-way valve in the CHS circuit bypasses the cooling fan (*'consuming'* the heat) until at set temperature is reached in the return flow of the engine circuit. By measuring the flow and temperatures before and after the fan the consumed energy can be calculated (see Figure 10).

3.1.5 Control system

A programmable logic controller (PLC) governs the control strategy of the system. It controls the electricity supply and electronic valves and is thus capable of shutting down the engine and fuel supply in case an automatic alarm or emergency button is triggered. It receives signals from temperature-, pressure-, engine speed-, lambda-, O_2 sensor etc. and adjusts operation according to these. A DEIF Generator Paralleling controller is in charge of synchronizing the generator to the electricity grid.

3.1.6 Safety

The setup has been approved for operation with natural gas by the gas distribution company HMN and for operation with producer gas by the Danish Safety Technology Authority. For and overview of safety aspects see Appendix B.

3.2 Data acquisition system

Relevant measurements are performed by various sensors and analysers in the system and acquired through National Instruments (NI) DAQ devices and software (LabView). Table 6 gives an overview of sensors and collected data.

Table 6: *Overview of measurements logged with the DAQ system.*

Type	Data	Symbol	Range	Signal	Channel
Temperature	Air	T_{air}	0 – 1200 °C	0.04 – 0.2 V	14
	Producer gas	T_{gas}	0 – 1200 °C	0.04 – 0.2 V	8
	Inlet	T_{in}	0 – 1200 °C	0.04 – 0.2 V	7
	Cooling forward	$T_{cool,f}$	0 – 600 °C	0.04 – 0.2 V	5
	Cooling return	$T_{cool,r}$	0 – 600 °C	0.04 – 0.2 V	6
	Exhaust	T_{ex}	0 – 1200 °C	0.04 – 0.2 V	9
Flow	Alternative gas	$\dot{V}_{gas}$	0 – 45 Nm^3/h	0.04 – 0.2 V	13
	Cooling CHS	$\dot{V}_{cool}$	0 – 4 m^3/h	0.04 – 0.2 V	12
Gas composition	Methane	ϕ_{CH_4}	0 - 20 vol. %	0.04 – 0.2 V	-
	Hydrogen	ϕ_{H_2}	0 - 50 vol. %	0.04 – 0.2 V	-
	Carbon monoxide	ϕ_{CO}	0 - 30 vol. %	0.04 – 0.2 V	-
	Carbon dioxide	ϕ_{CO_2}	0 - 30 vol. %	0.04 – 0.2 V	-
	Nitrogen	ϕ_{N_2}	100 - rest vol. %	0.04 – 0.2 V	-
Emissions	Oxygen from sensor	$\phi_{O_2,sensor}$	0 – 21 vol. %	0.2 - 1.0 V	0
	Oxygen from analyser	ϕ_{O_2}	0 - 25 vol. %	0.04 – 0.2 V	15
	Carbon monoxide	C_{CO}	0 - 10000 ppm	0.04 – 0.2 V	16
	Nitrogen oxide	C_{NO}	0 - 2000 ppm	0.04 – 0.2 V	17
	Nitrogen dioxide	C_{NO_2}	0 - 2000 ppm	0.04 – 0.2 V	18
Other	Power generated	P_{gen}	0 – 20 kW	0.04 – 0.2 V	1
	Heat effect	$\dot{Q}_{CHS}$	0 – 120 kW	0.04 – 0.2 V	11
	Boost pressure	p_{boost}	0.0 – 2.5 bar	0.04 – 0.2 V	4
	Frequency converter out	f_{fc}	0 – 50 Hz	0.04 – 0.2 V	10
	Phase	$\cos(\phi)$	0.0 – 1.0	0.04 – 0.2 V	2

Figure 11 shows the interface of the DAQ software, where measurements can be monitored live.

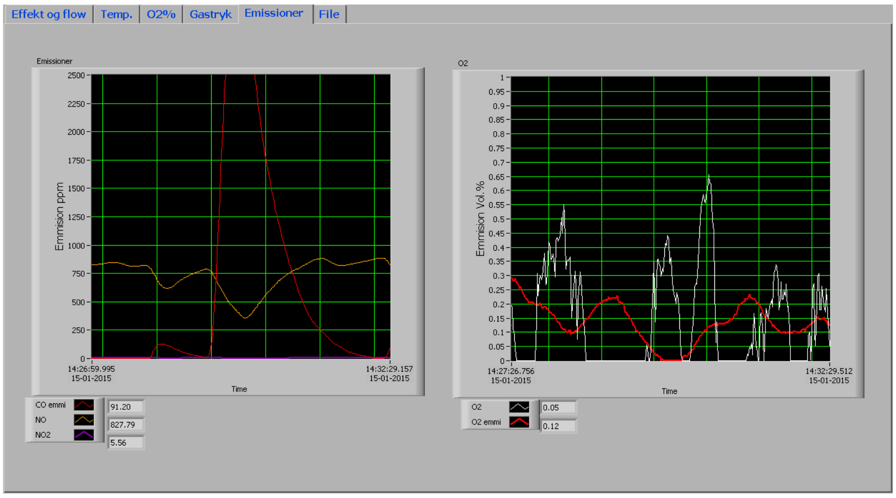


Figure 11: *LabView interface.*

3.3 Operation

This section provides a quick overview of operation with th test setup. For a step-by-step control description see Appendix A.

Operation of the test setup is relatively straight-forward as many processes are automated. Figure 12 shows the PLC software interface. It provides an overview of system alarms and status and user inputs for O_2 level, ignition timing and boost pressure. Figure 13 show the DEIF software interfaces that shows information about the generator. The setup has three operation modes:

Standby System is in standby and engine is stopped. Selecting this mode stops the engine if it is running and resets alarms.

Natural gas For operation on natural gas. The engine can only be started in this mode.

Producer gas Can be selected when engine is running on natural gas to switch to producer gas.

In both natural gas and producer gas mode pressure charging can be activated in either manual or automatic mode. In manual mode the compressor speed is adjusted manually with a rotary knob and in automatic mode the desired boost pressure is set in the PLC interface.

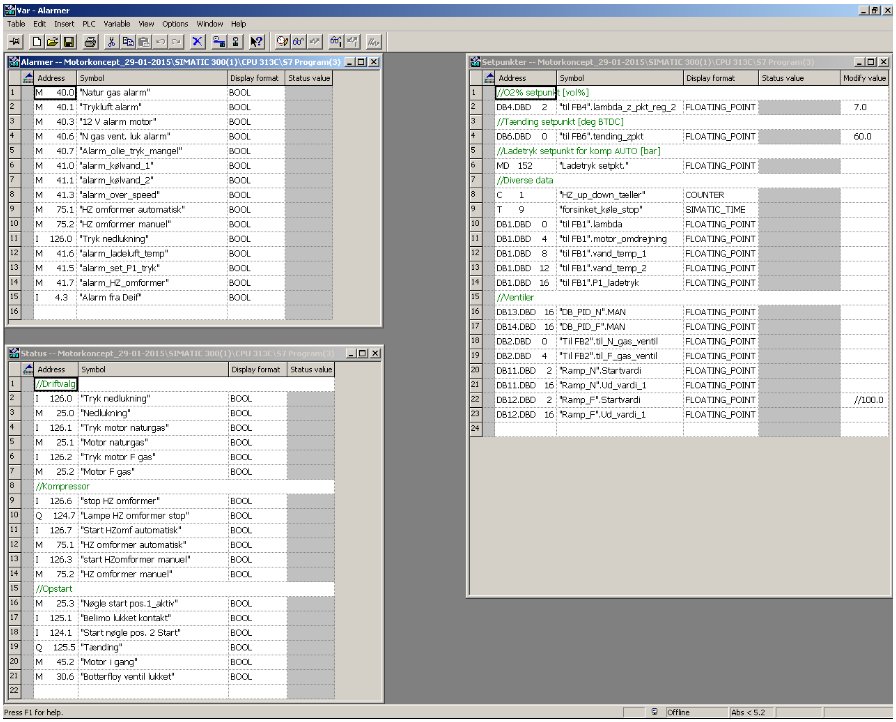


Figure 12: *PLC software interface.*

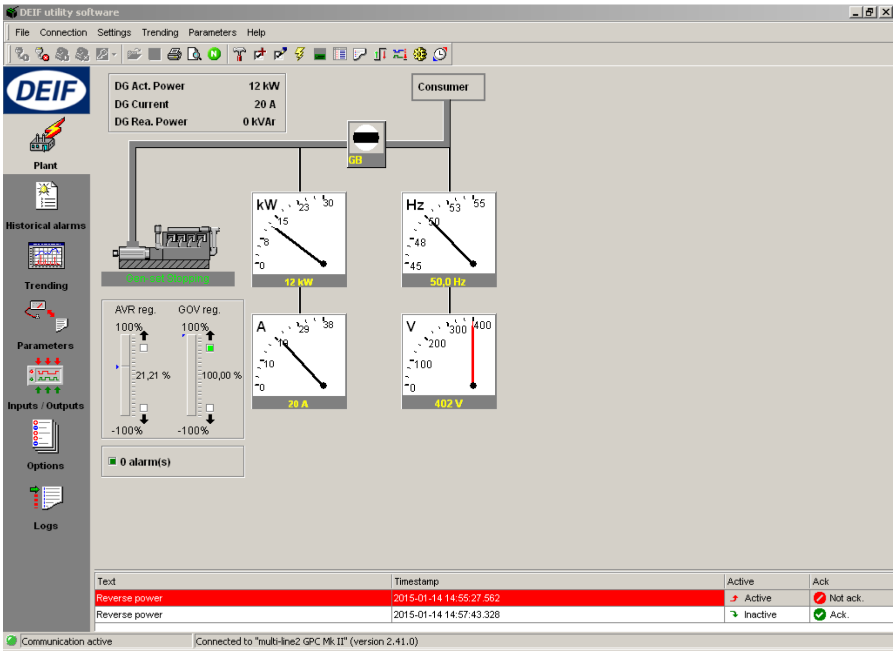


Figure 13: *DEIF remote interface.*

3.4 Preparation of test setup

As a part of this project various practical and technical work was done on the test setup in order to prepare for testing and solve problems occurring during tests. The work was done with help from technicians Erik Hansen, Kristian Estrup, Freddy Christensen and supervisors Jesper Ahrenfeldt and Ulrik Birk Henriksen. This section describes the main issues that was worked on.

Producer gas control valve The control valve on the producer gas fuel line was defect and blocked startup of the engine. It was sent in for repair and reinstalled.

Cooling system During preliminary tests the engine heated up rapidly and the temperature alarm on the cooling system triggered after a short period of operation. The whole cooling system was inspected to ensure optimal performance. The filters before the pumps were cleaned several times. Especially the engine cooling cycle carried a lot of dirt. The heat exchanger was cleaned, cooling fluid was changed and the system was ventilated. Also a new fuse for the radiator fan was installed as the old one started giving problems.

Measurement wiring All the wiring and signals of the different measurement points were gone through, tested and labelled.

Alarms The system alarms and emergency stops were tested and limits revised.

PLC program The PLC program was updated to solve various issues and improve user interface.

Emission analysis system The emission analysis equipment was sent in for calibration at DTU. See calibration report in Appendix D.

Gas analysis Measurements were taken for a reference gas and used to calculate correction factors for non calibrated measurements.

Throttle on air intake To be able to test stoichiometric operation with NG a throttle in form of a sliding valve was installed on the air intake.

TWC A TWC was purchase and installed in the exhaust system.

CO leak During producer gas operation CO leaking was detected in the engine room. A lot of work was put in detecting the leak. Parts of the intake system was pressure tested several times and all joints and gaskets were carefully inspected and several minor leaks were sealed. However, the main source turned out to be ventilation of the engine crankcase which was led directly out in the room. After fixing this no CO was detected in the room.

Compressor During operation with pressure charging the compressor started making jarring noises. The noise disappeared after disassembling and reassembling of the compressor. The problem was believed to originate from the compressor clutch.

3.5 Tests

Performance and emissions measurements have been taken during tests with the engine setup. Tests with Viking producer gas were carried out in the periods of Dec. 10.-12. 2014 and Jan. 14.-16. 2015. Some of the procedures and important things to notice in relation to the tests are describes in the following.

3.5.1 Viking producer gas composition

The gas composition from the Viking gasifier is not always the same. Variations e.g. of moisture content in the feedstock results in fluctuating gas composition.

During tests the producer gas from the Viking gasifier was continuously measured. Figure 14 shows examples of the gas composition from the test days together with the calculated LHV.

As it shows concentrations of the different components can vary significantly. These variations will have an influence on engine operation, performance and emissions. The LHV of the fuel naturally also varies, it is seen that it is strongly dependant on the H_2 concentration. This additional variable must be kept in mind when analysing results.

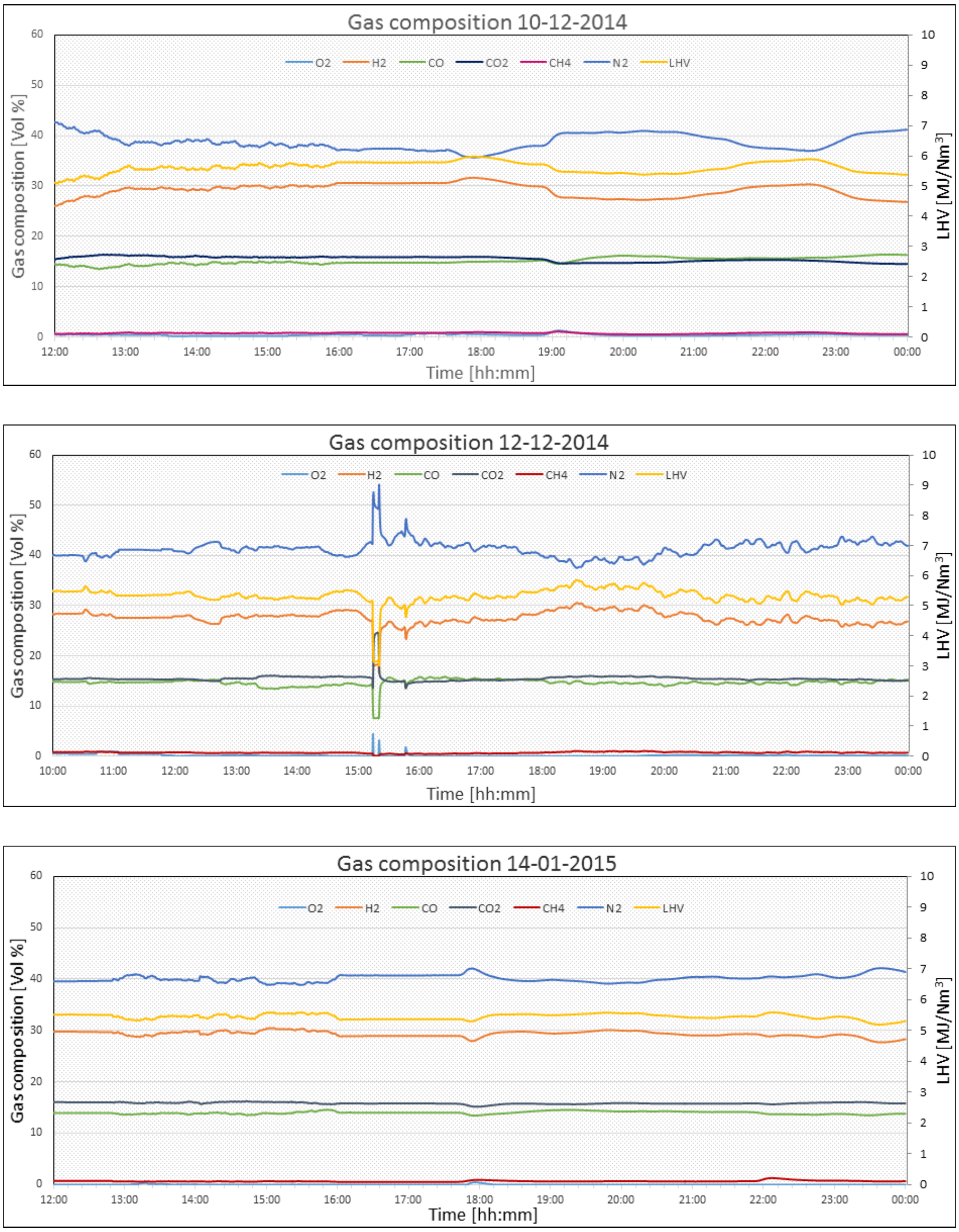


Figure 14: *Composition measurements of Viking producer gas and LHV from different test days.*

3.5.2 Engine measurements

To test the performance and emissions of the engine setup various parameters was measured and logged continuously (see Table 6). The influence on performance and emissions of three different operational parameters was investigated, λ, ignition timing and boost pressure. Besides this the influence of the TWC was tested by measuring exhaust emissions both before and after the TWC.

The data collected from the tests have been processed and analysed and are presented in section 4. The results are expressed as mean values of suitable time frames to compensate for fluctuations.

3.5.2.1 Excess air equivalence ratio

Figure 15 gives an overview of the course of the test with variable λ performed on December 10th 2014 through some of the key measurements. It shows raw measurements of O_2 concentration in the exhaust, electric power output, fuel flow and emission of CO and NO as functions of time.

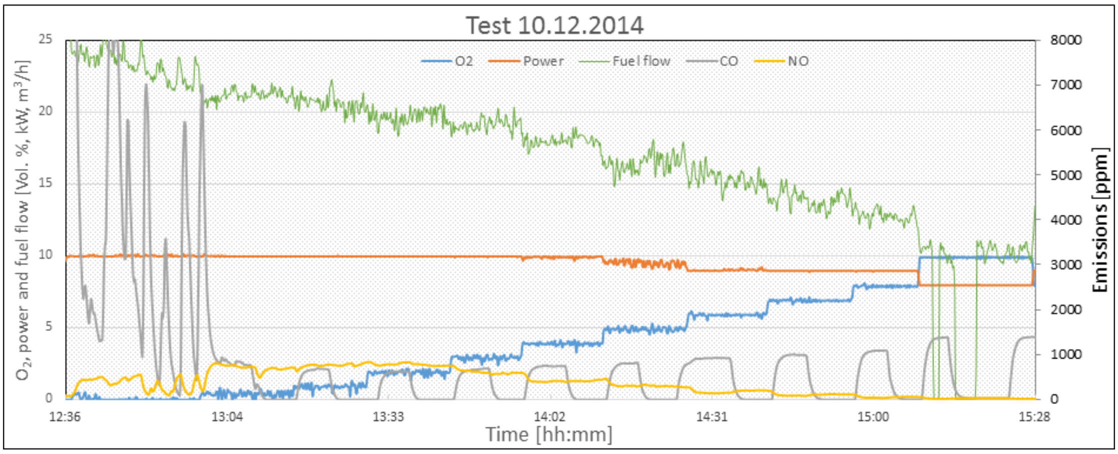


Figure 15: *Various raw measurements during test on 10.12.2014.*

The O_2 concentration was increased stepwise and left long enough for other parameters to settle. For each step emission measurements were switched between before and after the TWC. This is seen very clearly on the CO measurements.

To the right at high O_2 it is seen that the fuel flow measurements show sudden drops to zero. This is because the flow is near the lower measurement limit of the flowmeter. These values are omitted in the further analysis.

3.5.2.2 Stoichiometric operation

As discussed in section 2.2.5 in order for a TWC to work efficiently the λ value must be kept within a certain window close to stoichiometric condition.

As seen to the left in Figure 15 the emissions fluctuate a lot when the engine is operating near stoichiometric condition. Figure 16 gives a closer look at what is happening. It shows exhaust concentrations of O_2, CO and NO. Notice that the axes are scaled differently and O_2 has been scaled by a factor 10^3 for better visualisation.

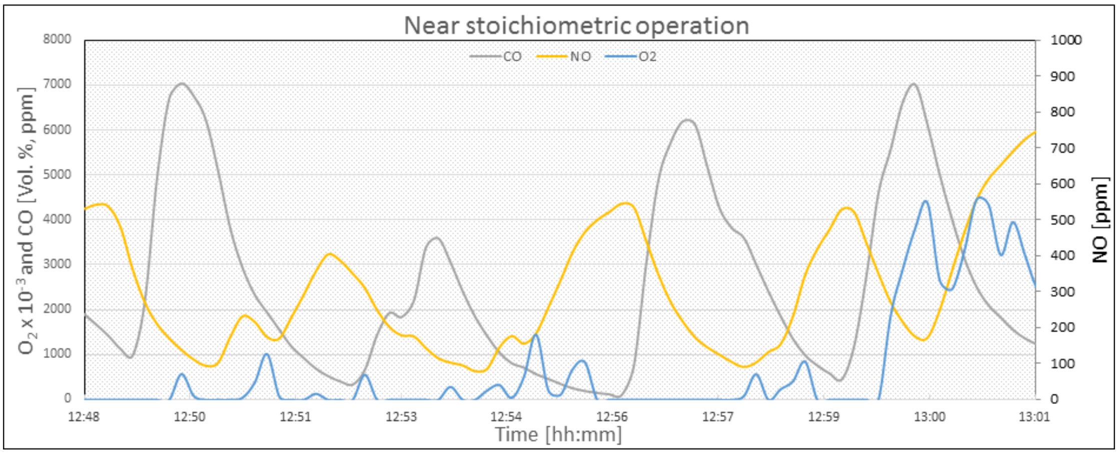


Figure 16: *O_2, CO and NO emissions during near stoichiometric operation. Notice the scaling of axes.*

Even though the O_2 value is set to zero it varies significantly over time. When $O_2 = 0$ on the graph in practise it means that $\lambda \leq 1$. It shows that CO emission peaks when O_2 is low (rich operation) and decrease when there is excess of O_2 (lean operation). The exact opposite trend shows for NO (the CO and NO curves are shifted slightly to the right of the O_2 curve due to lag caused be the sample line to the emission analysis equipment).

At rich operation UCO cause high CO emission and there is no oxygen present for the TWC to oxidise the CO. On the other hand less NO_x is produced and the reducing environment promotes NO_x conversion in the TWC. At lean operation these mechanisms are reversed.

This proves the potential of both CO and NO_x conversion with the TWC, however at this point the control of λ is not accurate enough for this setup to keep a simultaneous good conversion of both emission types.

This means that good indicative emission measurement at stoichiometric operation with TWC have not been possible in this project.

3.5.2.3 Ignition timing

Figure 17 shows key measurements during test with variable ignition timing on December 12th 2014. The ignition timing was retarded in steps of 3 crank angle degrees (CAD) starting at 24 CAD before top dead center (BTDC) and ending at 0 CAD BTDC.

The O_2 level was set slightly lean of stoichiometric where NO_x emission was highest and reasonably stable operation could be achieved. However, the conditions still occasionally shifted to rich, which caused spikes in CO emission. These are left out in the emission results.

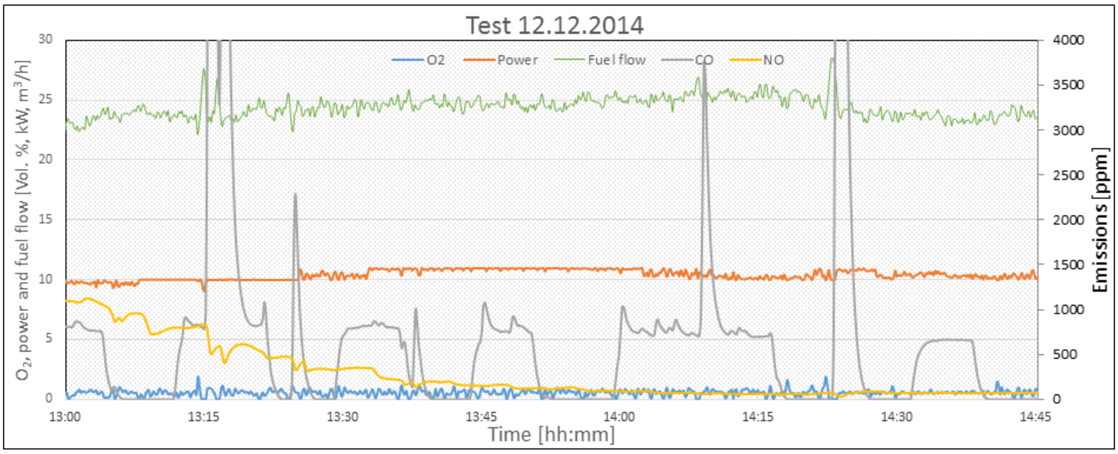
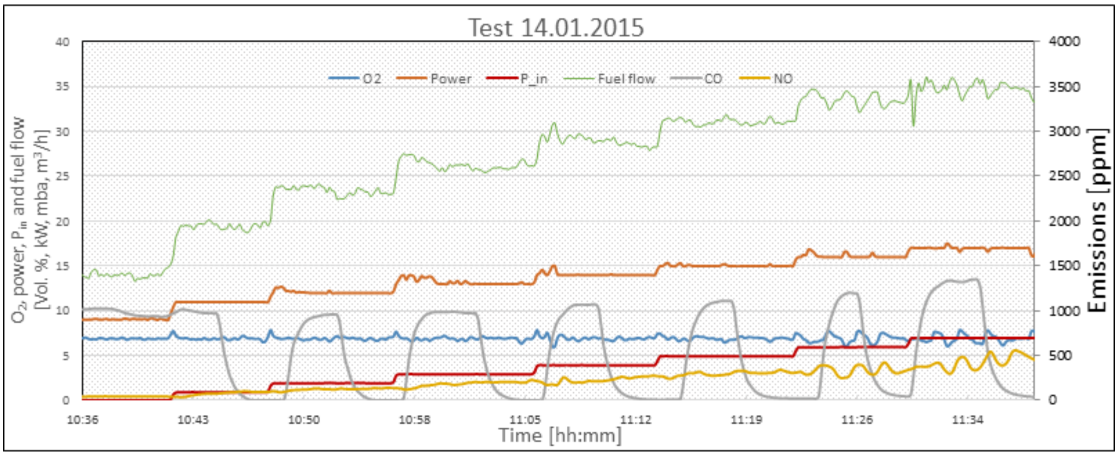


Figure 17: *Various raw measurements during test on 12.12.2014.*

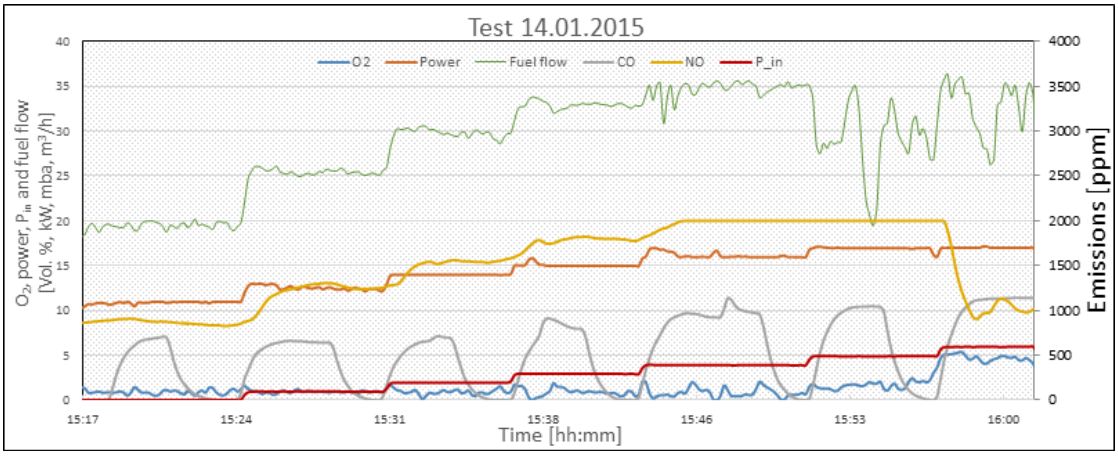
3.5.2.4 Pressure charging

Figure 18 shows key measurements during test with pressure charging on January 14th 2015 for two different λ values. The boost pressure was increased in steps of

100 mbar from 0-1000 mbar or until the fuel supply was insufficient and emissions measured before and after the TWC. This was done at four different λ values, $\lambda = 1.8, \lambda = 1.5, \lambda = 1.3, \lambda = 1.06$. At the highest levels of boost pressures especially at low λ values it is seen that the fuel flow stops increasing (as the maximum supply is reached) and becomes very unstable and O_2 level starts increasing as a consequence hereof.



(a) *Various raw measurements during test on 14.01.2015 at* $\lambda = 1.5$.



(b) *Various raw measurements during test on 14.01.2015 at* $\lambda = 1.06$.

Figure 18

4 Results

This section presents the results obtained from the tests carried out in this project.

4.1 Performance

Figure 19 and 20 shows the electric power output and the electric efficiency of the engine and generator as functions of λ and ignition timing ($\lambda = 1.03$) respectively.

The power output decreases 20% from $\lambda = 1.0$ to $\lambda = 1.75$. The efficiency drops near stoichiometric operation. Because of unstable operation the conditions will fluctuate between slightly rich and lean when operating close to stoichiometric. The efficiency will naturally decrease because of the excess fuel left unburned during rich operation.

The efficiency increases slightly up to $\lambda \approx 1.4$ from 25% to 29%. From here the efficiency increases quite rapidly to 36% at $\lambda = 1.75$. From previous results the efficiency is expected to decrease slightly in this area [18]. Uncertainties and issues with the lower range volume flow measurements of the producer gas are likely to have disturbed these results. This is described further in section 4.3.

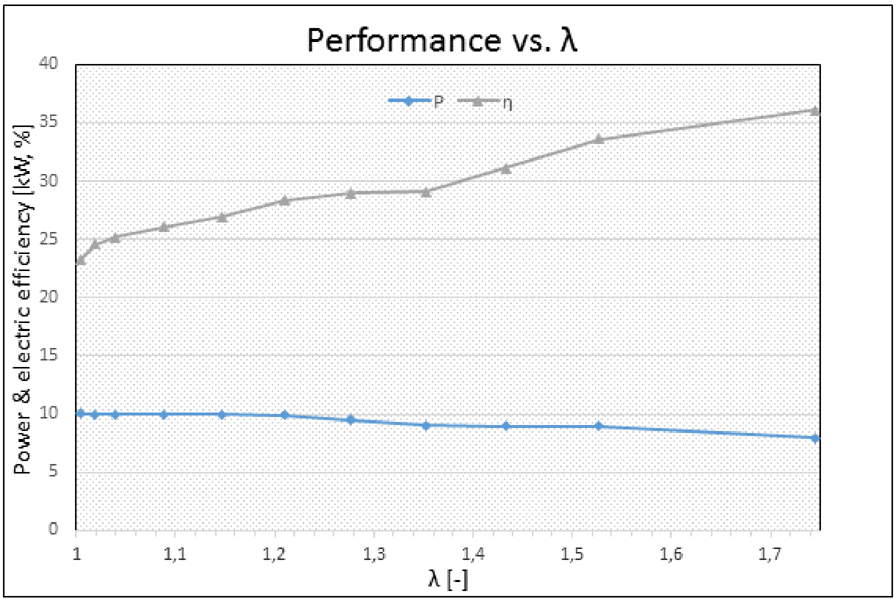


Figure 19: *Power and electric efficiency as a function of λ.*

The power output and efficiency is almost constant for for the entire range of ignition timing. A slight increase in both is seen between 5-12 CAD BTDC.

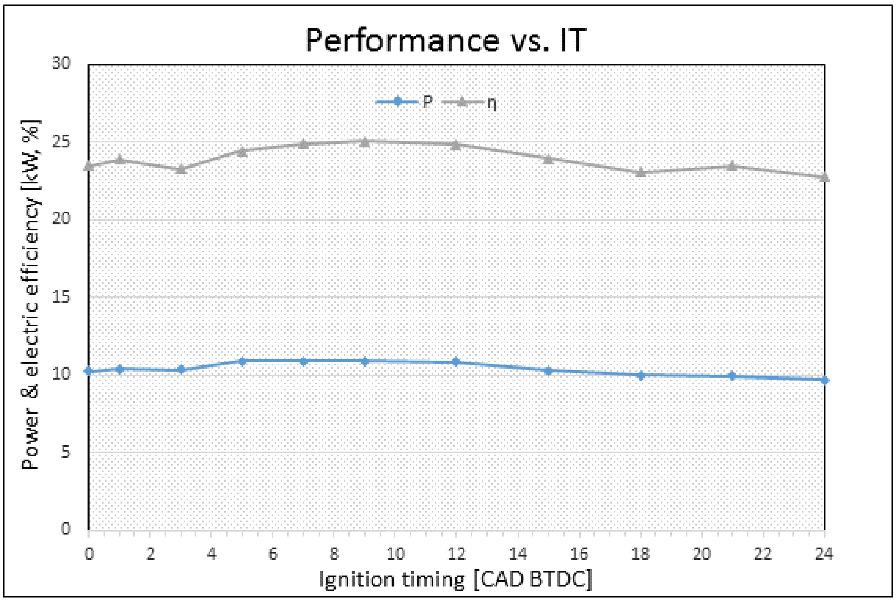
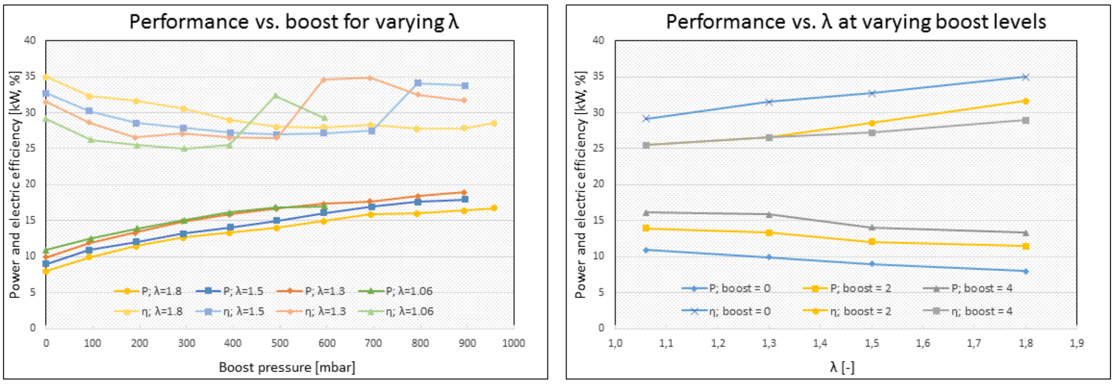


Figure 20: *Power and electric efficiency as a function of ignition timing at* $\lambda = 1.03$.

Figure 21a shows the engine performance as a function of boost pressure for different values of λ, while Figure 21b shows the same but as a function of λ. The power output increases with the boost pressure and decreases with λ as expected.

As the boost pressure increases the supply of producer gas becomes insufficient especially at lower λ values. This results in unstable fuel flow and engine operation. Figure 22a shows a sudden drop when the maximum flow is reached and the boost pressure is further increased. Figure 22b shows that the λ value increases at those points because more air is sucked in, but the fuel flow cannot keep up. Only boost pressures up to 400 mbar are therefore shown in Figure 21b.

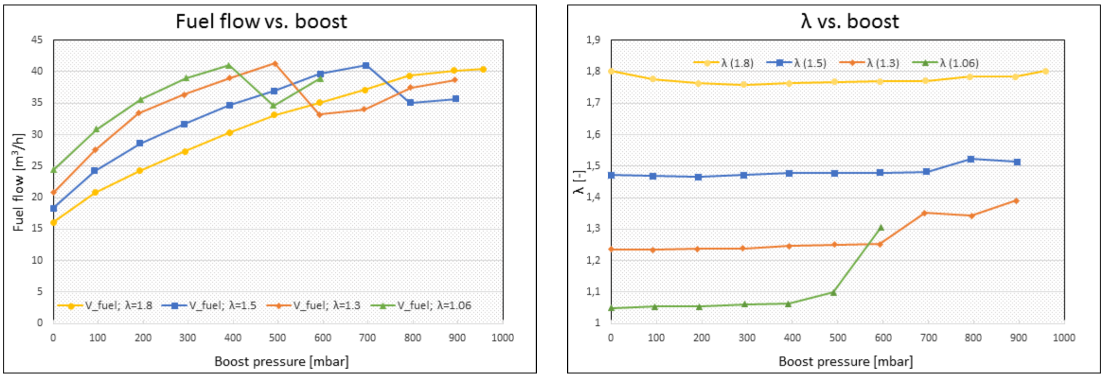
The efficiency decrease with the boost pressure. Mostly at the beginning and then it becomes more constant. The values at zero boost pressure seem abnormally high, between 29% and 35%. They should be similar to the ones in Figure 19. The efficiency seems to slightly increase with λ.



(a) *Power and electric efficiency as a function of boost pressure for different levels of λ.*

(b) *Power and electric efficiency as a function of λ for different levels of boost pressure.*

Figure 21



(a) *Fuel flow as a function of boost pressure.*

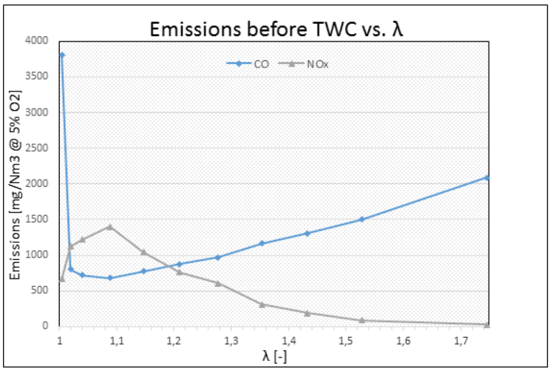
(b) *λ values as a function of boost pressure.*

Figure 22

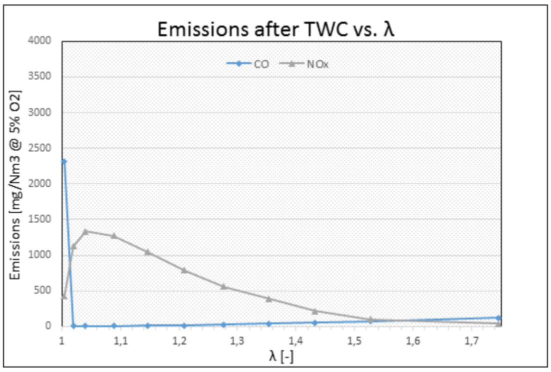
4.2 Emissions

4.2.1 Excess air equivalence ratio

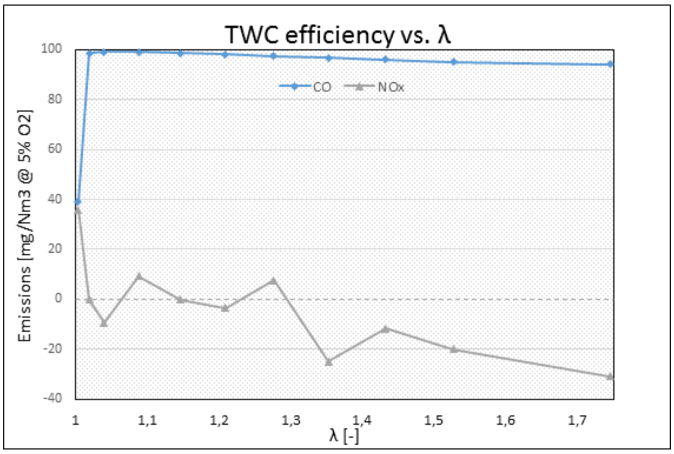
Figure 23 shows the emissions of CO and NO_x as a function of λ before the TWC (23a), after the TWC (23b) and the conversion efficiency of the TWC (23c) corresponding to the performance results in Figure 19.



(a) *CO and NO_x emission as a function of λ before the TWC.*



(b) *CO and NO_x emission as a function of λ after the TWC.*



(c) *Conversion efficiency of the TWC.*

Figure 23

The dependency of emissions on λ is seen very clearly in 23a. The CO emission is very high close to stoichiometric due to occasionally rich condition leads to unburned CO. It then drops very quickly to a minimum at $\lambda \approx 1.1$ and then increases steadily for leaner conditions due to lower temperatures and mechanisms described in section 2.2.3.

The NO_x emission has its maximum at $\lambda \approx 1.1$ as expected. It decreases quite fast at leaner conditions.

The CO emission after the TWC is very low except for near stoichiometric conditions. periods of rich conditions and thus lack of oxygen in the exhaust to oxidise CO contributes to a higher average CO emission. At lean conditions very good CO conversion is achieved.

At lean conditions NO_x emission is more or less constant before and after the TWC. This is expected since the TWC need a reducing environment to convert NO_x. Some conversion of NO_x is seen at stoichiometric conditions.

Conversion efficiencies of 39% for CO and 36% for NO_x at stoichiometric conditions are not impressive compared to what can be expected. However, these results do not represent the full potential of emission reduction with the TWC. The inability to keep the λ value within a sufficiently narrow window around stoichiometric causes the low efficiencies.

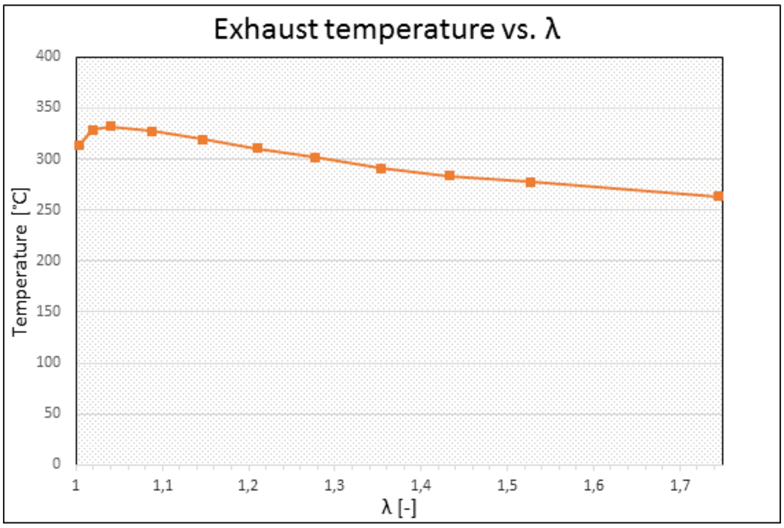
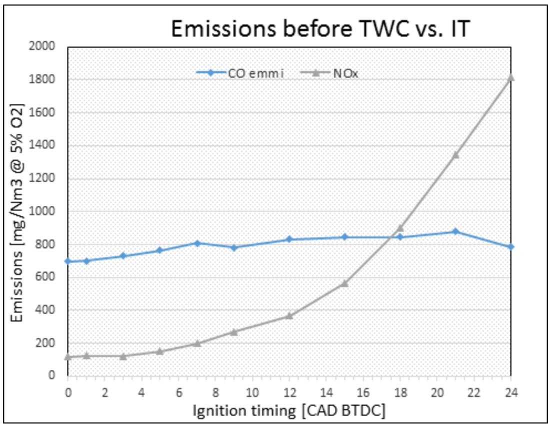


Figure 24: *Exhaust temperature as a function of λ.*

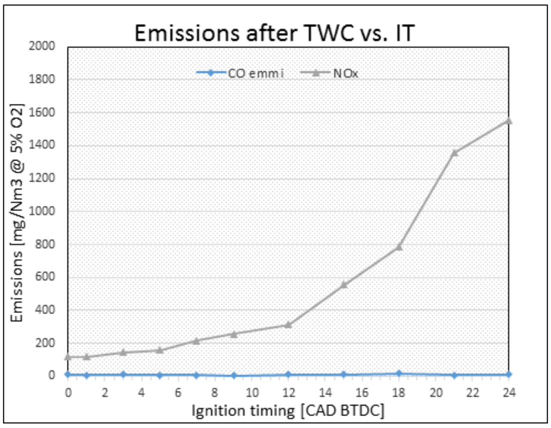
Figure 24 shows the exhaust temperature as a function of λ. It peaks at slightly lean conditions and decreases with increasing λ. The low temperature does not seem to affect the TWC efficiency.

4.2.2 Ignition timing

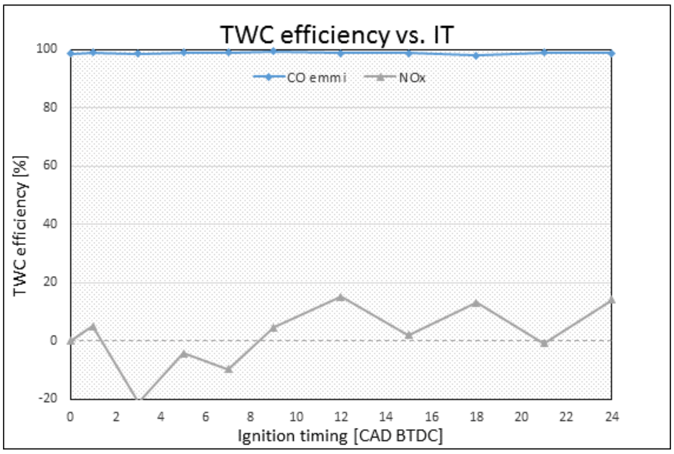
Figure 25 shows the emissions of CO and NO_x as a function of ignition timing at $\lambda = 1.03$ before the TWC (25a), after the TWC (25b) and the conversion efficiency of the TWC (25c) corresponding to the performance results in Figure 20.



(a)



(b)



(c)

Figure 25

The values of CO emission before the TWC are similar to that for corresponding λ value in Figure 23a. No significant dependency on ignition timing is seen.

The NO_x emission show a very strong dependency on ignition timing. It is high for advanced ignition timing and decreases rapidly for retarded ignition timings. It is decreased by a factor 15 when the ignition timing is retarded from 24 to 3

CAD BTDC. This shows that at these conditions ignition timing can be optimised to control NO_x emission without significant decrease in efficiency or increase in CO emission.

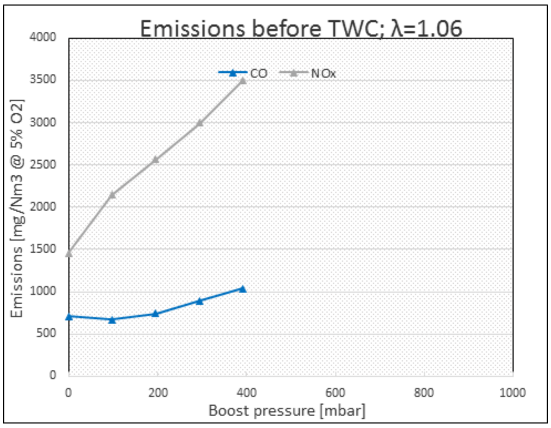
4.2.3 Pressure charging

Figure 26 and 27 shows the emission of CO and NO_x before and after the TWC as a function of boost pressure for different values of λ, while Figure 28 shows the TWC conversion efficiencies.

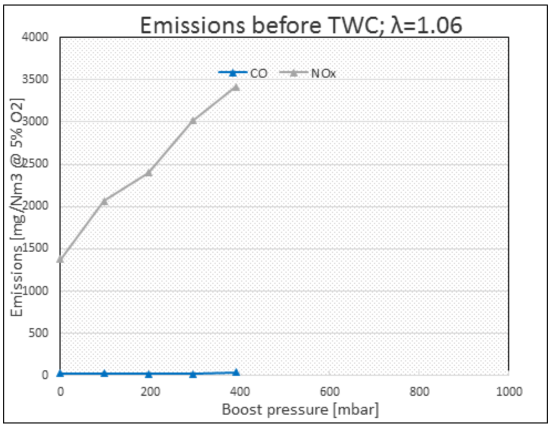
The results show that emissions follow the trend from the non pressure charged results in Figure 23a with respect to λ, which are increasing CO and decreasing NO_x with increasing λ. CO emission has a slightly increasing trend for increasing boost pressure. The higher pressure may increase the effect the crevice mechanism leading to higher UCO.

For NO_x it shows increased emission with increasing boost pressure. It increases at higher rates at lower values of λ where the NO_x levels are also higher. The higher pressure causes higher flame temperatures and NO_x.

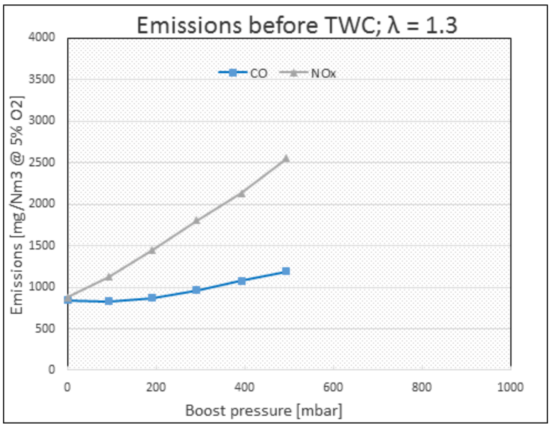
The efficiency of the TWC follows the same trend as for the non pressure charged tests, very good conversion of CO and no sign of any NO_x conversion. The efficiency values for NO_x are quite fluctuating especially for the higher values of λ since the lower NO_x levels make the efficiency more sensitive to uncertainties.



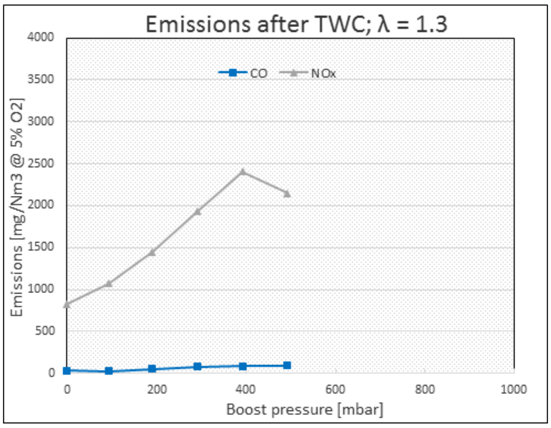
(a)



(b)



(c)



(d)

Figure 26

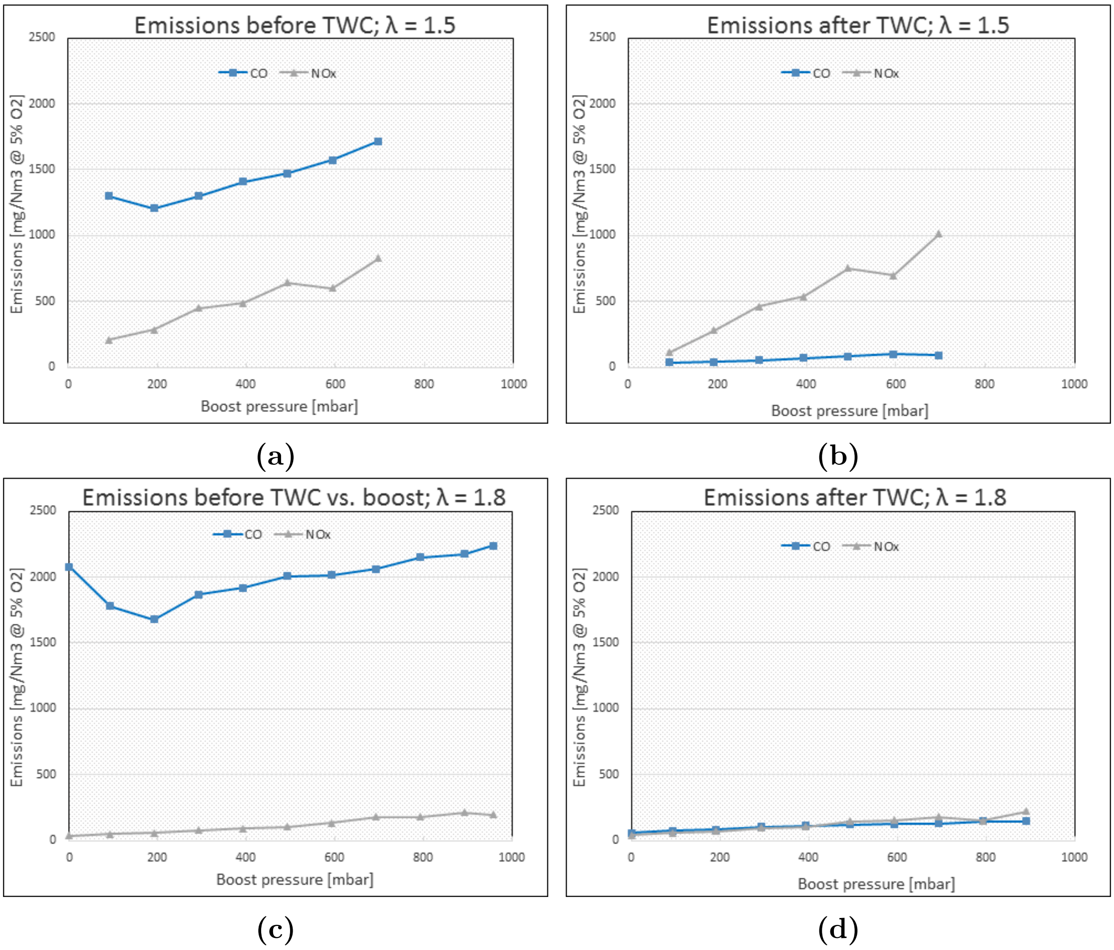
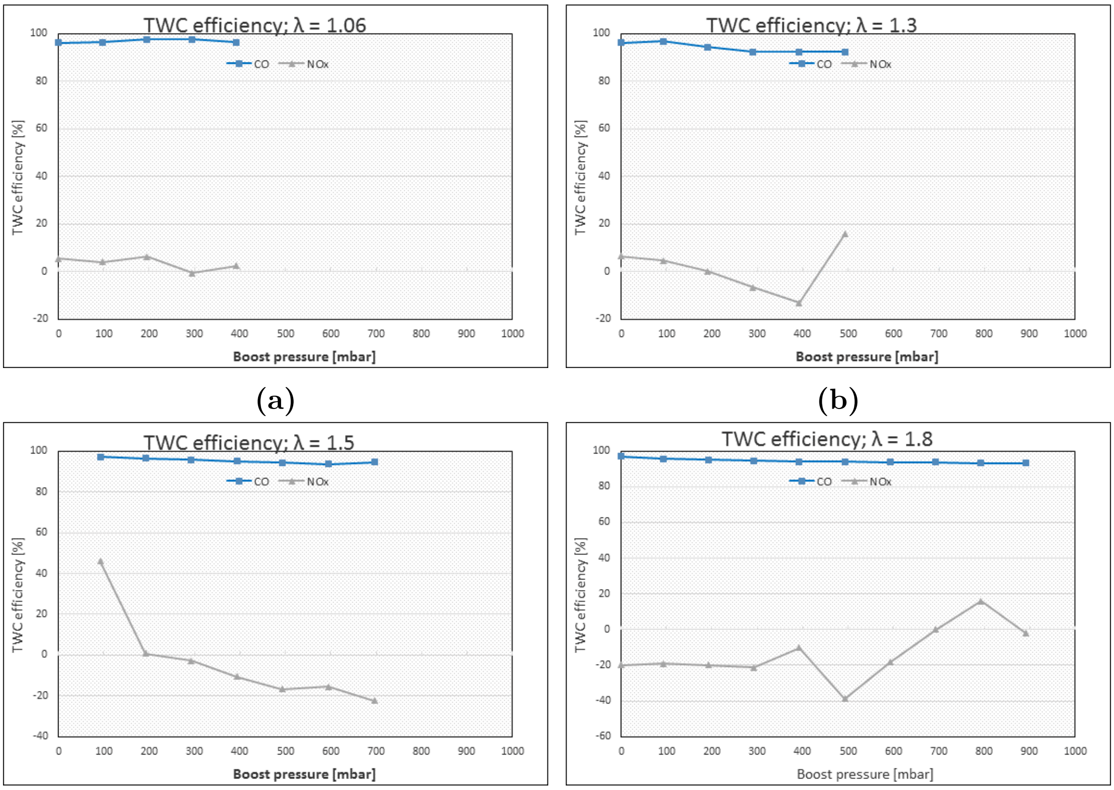

Emissions before TWC; λ = 1.5
CO NOx
Emissions [mg/Nm3 @ 5% O2]
Boost pressure [mbar]
(a)
Emissions after TWC; λ = 1.5
CO NOx
Emissions [mg/Nm3 @ 5% O2]
Boost pressure [mbar]
(b)
Emissions before TWC vs. boost; λ = 1.8
CO NOx
Emissions [mg/Nm3 @ 5% O2]
Boost pressure [mbar]
(c)
Emissions after TWC; λ = 1.8
CO NOx
Emissions [mg/Nm3 @ 5% O2]
Boost pressure [mbar]
(d)

Figure 27



(a) (b)

(c) (d)

Figure 28

4.3 Uncertainties

This section describes the most important uncertainties related to the measurements.

4.3.1 Flow measurements

The efficiency results based on flow measurements of the producer gas are subject to high uncertainty. The producer gas flow was measured with a vortex flowmeter. The initial measurements was found to show too low values especially in the lower range flows.

Reference measurements were made using a gas meter was used to correct the initial measurements, but low range flows still seem too low. Different factors may cause the high uncertainty:

- The lower range limit of the flowmeter is around 12.5 m^3/h. Flows measurements close to this limit are subject to increased uncertainty.
- Reference measurements were made with air, which have different properties than producer gas.
- The moisture content in the gas was not taken into account in the data analysis, however low temperatures ≈ 5°C means very little moisture in the gas.

4.3.2 Emission range

The equipment used for emission measurements were ABB A2020 Limas11 UV gas analyser (NO and NO_2) and Uras26 infrared analyser (CO and O_2). The measurement ranges are given in Table 11. During certain operating conditions the emissions exceeded these ranges. CO at rich operation and NO at near stoichiometric operation with pressure charging.

The analysers have a lower accuracy outside their ranges, but the measurements are still thought to be reasonably accurate.

4.3.3 Producer gas composition

As described in section 3.5.1 the varying gas composition is an extra variable that is not always taken into account in the results. The LHV of the producer gas and λ value are calculated for each time step, but e.g. in the emission results this variable is not accounted for.

4.3.4 Time steps

The time period between change in operating conditions e.g. λ value or boost pressure, influences the measurements. Some parameters need time to stabilise or have a delay like the emissions caused by the length of the sample line. The longer periods also allow better averaging of fluctuations, but is of course more time consuming.

5 Conclusion

A laboratory scale biomass CHP gas engine setup has been reconditioned and updated in preparations for experimental testing. A three-way catalytic converter has been purchased and installed.

The test setup has been operated and tested with biomass producer gas from the Viking gasifier. Tests showed that the control of λ of this setup is not accurate enough to keep λ in a window narrow enough for efficient conversion of CO and NO_x in the TWC. To achieve this a more sophisticated control system need to be implemented. It did show potential of reduction of both CO and NO_x. As conditions fluctuated between lean and rich CO and NO_x was reduced alternately.

Performance results showed no significant dependence of and efficiency on ignition timing. Higher power output, but generally lower efficiency with increasing boost pressure at different λ values and increasing efficiency with increasing λ value at different boost pressures.

At lean operation NO_x emission is very low. At $\lambda = 1.8$ and boost pressure close to 1 bar NO_x emission is ≈ 200 [mg/Nm^3 @ 5%O_2] well below the regulated limit. CO emission is high due to unburned fuel, but still below the Danish regulated limit at $\lambda = 1.8$ and boost pressure just under 1 bar. The trend, however, shows that it will increase at leaner conditions. The TWC functions as an oxidation catalyst at lean operation and shows very good conversion efficiency for CO (94-99%)

Near stoichiometric operation the NO_x emission is high, but it was shown that by retarding ignition timing it could be lowered significantly without significant decrease in power and efficiency. CO emission is low on the lean side of stoichiometric, but becomes very high on the rich side. Some degree of conversion of both CO and NO_x was shown in the TWC, but due to aforementioned conditions no real indicative results were obtained.

6 Further work

This section presents suggested further work regarding modifications to the test setup and relevant tests.

6.1 Test setup

In-cylinder pressure sensor Installation and setup of an in-cylinder pressure sensor would enable heat release analysis and other detailed analyses such as mechanical stress, cycle-to-cycle variations etc.

Exhaust throttle In real applications a turbocharger would be used for pressure charging and not an electrically powered compressor. Installation of a throttle in the exhaust would make it possible to simulate pressure drop in the exhaust from turbocharging.

Air-fuel ratio control Based on tests conducted in this project a more accurate control of λ has to be implemented to test and optimise stoichiometric operation with TWC.

Recirculation of crankcase ventilation At the moment the ventilation air from the crankcase is discharged to the atmosphere. Instead it could be fed back to the engine inlet to avoid emissions.

Flowmeter calibration To produce reliable performance results in the future the producer gas flowmeter needs to calibrated.

6.2 Testing

Stoichiometric operation with TWC This approach still needs to be tested with a better air-fuel ratio control. TWC efficiencies, sensitivity to λ, and optimisation of performance.

High compression ratio To achieve better performance tests with the high compression ratio engine should be conducted.

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Appendices

A Control description

This guide describes the step-by-step procedures for start-up, operation of various modes, data acquisition (DAQ) and shut-down of the biomass CHP gas engine at BGG.

The system has three basic modes; standby, natural gas (NG) and producer gas (PG). In addition pressure charging can be applied in either fuel mode and exhaust emissions measurements can be acquired from either before or after the three-way catalyst (TWC).

System start-up

- Check that all emergency stop buttons are disabled.
- Turn on main power switch (side of main panel).
- Turn system key on main panel: OFF → ON → REBOOT. Key will automatically return to ON. System is now in standby mode.
- Turn on system computer. Run applications:
  - Simatec Manager (Programmable Logic Controller interface (PLC)). File: Motorkoncept_29_01_2015
    - Open and monitor variable tables: Alarmer, Status, Setpunkter.
  - DEIF (Generator Paralleling Controller (GPC) remote interface)
    - Connect to GPC
- Make sure that all necessary fuel line and compressed air valves are open.

DAQ system

- In case emission and gas analysers are off; power on and wait for warm-up.
- Turn on computer for DAQ. Run LabVIEW application. File: EC.lvproj
- Open exhaust sample valve
- Connect sample input hose (Make sure there is always flow through).
- Press Run button to start DAQ.

- To switch between emission measurement before and after the TWC adjust the two sample line valves in the engine room.
- Press Stop button to end DAQ.

Operation

Standby mode

In this mode the system is in standby and the engine is not running. This mode is active by default at system start-up.

- Press STANDBY/RESET to reset alarms.
- Press STANDBY/RESET to stop engine operation and return to standby mode.

Natural gas mode

Engine always starts from this mode.

- Press NATURAL GAS to activate natural gas mode.
- Turn start key: ON → START (Hold until engine starts and then release). The GPC will now adjust the engine speed to synchronise the generator frequency with the mains electricity grid and then connect to it.

From producer gas mode

- Press NATURAL GAS to switch to natural gas mode.

Producer gas mode

This mode can only be activated from the natural gas mode.

- Wait until the "F-gas klarmelding" turns on.
- Press PRODUCER GAS to switch to producer gas mode.

Pressure charging

Pressure charging can be applied in both fuel modes. There are three different modes; Off, manual operation and automatic operation.

Off mode

In this mode pressure charging is deactivated, and the engine will be naturally aspirated. This mode is activated by default at system start-up.

- Press KOMP. OFF to activate this mode from any of the other pressure charging charger modes.

Manual operation

This mode can be activated from either off or automatic operation mode.

- Press KOMP. MANUAL to activate manual operation mode.
- Adjust the rotary knob KOMP HZ to control the compressor speed.

Automatic mode

This mode can be activated from either off or manual operation mode.

- Press KOMP. AUTO to activate automatic operation mode.
- Set the desired charge pressure in "Setpunkter" window - "Ladetryk setpkt." in PLC software.
  - Type value in [bar] positive pressure and press "Modify variable" button.

Variable parameters

The ignition timing and oxygen concentration can be set to desired value in the PLC interface variable table: "Setpunkter"

- Enter O_2 value in [vol. %] under "O2% setpunkt" Press "Modify variable"
- Enter ignition timing in[$\frac{1}{3}$CAD BTDC] under "Tænding setpunkt" Press "Modify variable"

System shut-down

- Press STANDBY/RESET to turn off engine.
- Close fuel line and compressed air valves.
- Disconnect sample hose and close sample line valves

B KT Safety assesment

Department of Chemical Engineering, DTU

Safety Assessment of experimental facilities

<table>
<tr><th colspan="5">Setup identification</th></tr>
<tr><td colspan="2">Name of the setup</td><td colspan="3">CHP gas engine test setup</td></tr>
<tr><td colspan="2">Location</td><td colspan="3">DTU Risø campus b. 321 (outside)</td></tr>
<tr><td colspan="2">Research group</td><td colspan="3">CHEC</td></tr>
<tr><td colspan="2">Project manager</td><td colspan="3">Jesper Ahrenfeldt, Ulrik Birk Henriksen</td></tr>
<tr><td colspan="2">Safety assessment carried out by</td><td>Martin Soon Ho Holm</td><td>Date:</td><td>27-01-2015</td></tr>
<tr><td rowspan="2">Local safety group</td><td>Signature</td><td></td><td>Date:</td><td></td></tr>
<tr><td>Signature</td><td></td><td>Date:</td><td></td></tr>
<tr><td>Control</td><td>Signature</td><td></td><td>Date:</td><td></td></tr>
</table>

Description	
Purpose	The purpose of the setup is to test the performance and emissions of a lab scale CHP four-stroke SI engine running on different gases and in different configurations with three-way catalytic converter, super charging and compression ratio.

Description (include PI diagram, drawings etc)

The setup is built as a transportable unit in a 20ft container with separate engine- and control room.

Two different engines are available for the setup. Both are four-stroke, four-cylinder 1.86L engines produced by Lister Petter. One is a natural gas engine with a compression ratio of 9.5:1 and the other is a Diesel engine with a compression ratio of 18.5:1 rebuilt to gas operation. The generator used in the setup is a Leroy Somer four-poled synchronous generator with a nominal effect of 19.2 kW. When synced to the 50Hz grid the gen-set runs at 1500 rpm.

There are two gas supplies, one for natural gas and one for alternative gas, e.g. biomass producer gas, and one air intake. When running on natural gas the fuel is added in a venturi tube. When running on alternative gas the gas is added in a T-piece after the venturi tube. In both cases the air-fuel mixture is led through a mixing device to secure a homogeneous mixture before entering the combustion chamber. A compressor driven by an electro-motor is parallel coupled between the mixing device and engine making it possible to pressure charge the engine.

The cooling system is made to simulate a central heating system in order to be able to calculate the overall system efficiency (power + heat). It consists of an engine circuit and a heating circuit connected by a heat exchanger. A three-way valve bypasses the cooling fan until at set temperature is reached in the return flow of the engine circuit.

A PLC governs the control strategy of the system. It controls the electricity supply and electronic valves and is thus capable of shutting down the engine and fuel supply in case an automatic alarm or emergency button is triggered. It receives signals from temperature-, pressure-, engine speed-, lambda-, CO sensor etc. and adjusts operation according to these. A DEIF Generator Paralleling controller is in charge of synchronizing the generator to the electricity grid.

The setup has been approved for operation with natural gas by the gas distribution company HMN and for operation of producer gas by the Danish Safety Technology Authority.

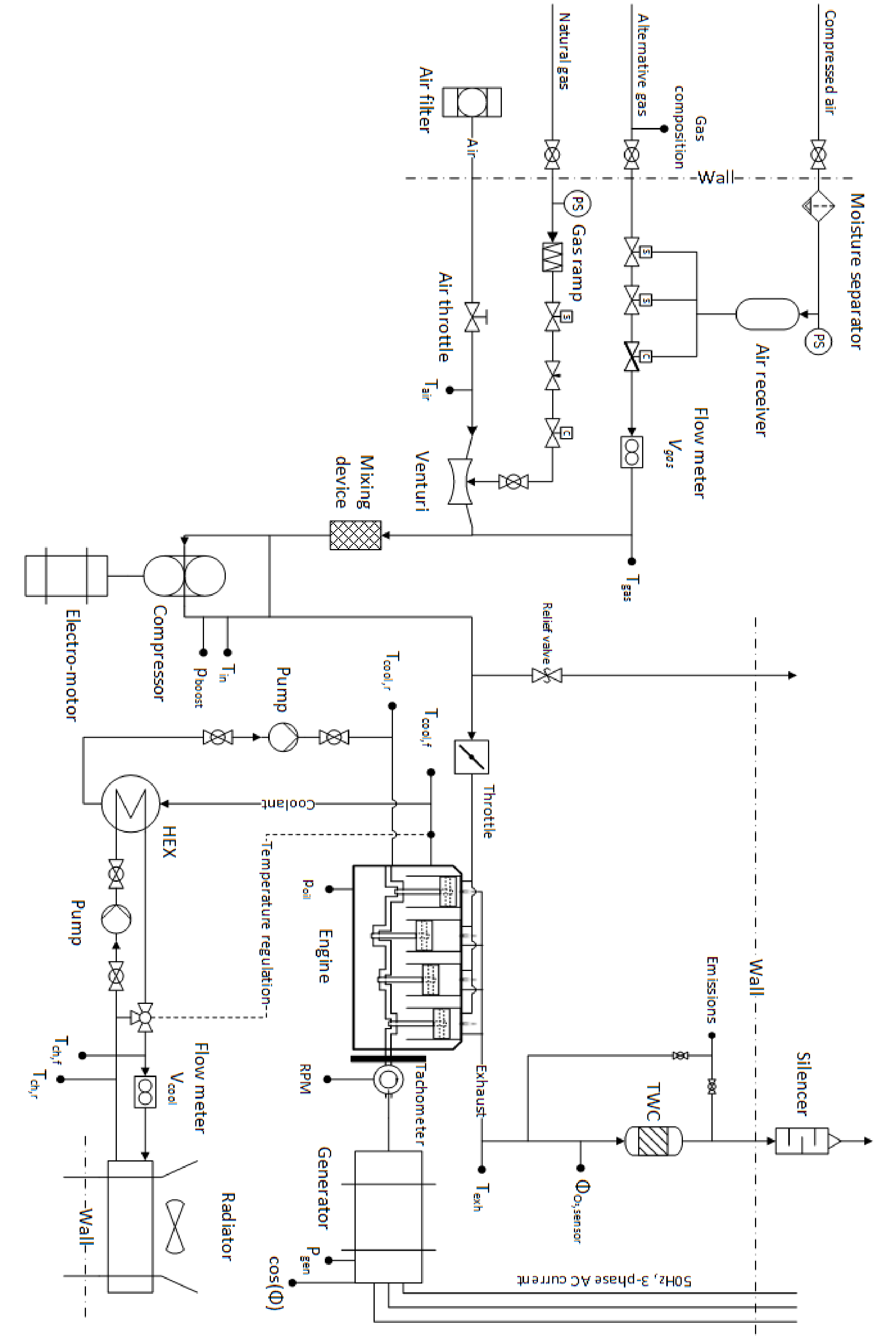


Figure 1: System diagram

Safety Assessment of experimental facilities

Operating conditions and main specifications	
Reactor or main vessel size	1.86 L engine displacement volume.
Normal operating temp/max temp	2000 K
Normal operating pressure/max pressure	100bar
Other relevant specifications	Test duration: max. 5 h Unmanned max. 1 h

Discharge to environment from Materials and Chemicals involved			
	Component(s)	Amounts – discharged to where?	CHEM-APV approved ?
Gases	Exhaust gas (CO_2, H_2O, O_2, N_2, CO H_2)	~ 47 L/s. varying concentrations. Outside environment through exhaust pipe.	
Liquids	Coolant	Small amounts can be spilled during maintenance	
Solids	NO, NO2	Varying concentrations. Outside environment through exhaust pipe.	
Dust			
Odors			
Consumption from building supply			
Gases	Natural gas (gas grid), H_2, N_2, CO, CO_2, CH_4 (bottled), producer gas from gasifier, compressed air	Combusted in engine, pneumatic valves.	
Liquids			
Solids			
Electrical power	Cooling fan, super charger, cooling pumps etc.	Auxiliaries: < 2kW Super charger: 0kW to 7.5kW Generator: -6kW to -12kW	

Main operational risks associated to the setup Please describe the main risks of this setup and what has been done to minimize these risks. Use ”Analysis of deviations from normal operation” below, as a work tool to identify and take measures against risks.		
	Risk	**Minimized by**
1	CO leakage	Fixed acoustic and visual CO alarm with automatic system shut-down. Personal CO alarm is worn during operation. Ventilation – continues after shut-down
2	Engine/compressor blow-out	Longitudinal engine placement – Engine rotation across the room. In case of an explosion, fragments are likely to travel towards the sides of the container and not towards the control room. Separate control room with break-proof window. Pressure relief valve on fuel/air pipe just before engine. Emergency stop buttons. Alarms with automatic shut-down - Coolant temperatures limit 95°C - Engine revolution limit 2500 rpm - Engine oil pressure switch - Charge air pressure limit 1bar - Charge air temperature limit 600°C
3	Hot components	Exhaust pipe is shielded to avoid contact.
4	Moving parts: Compressor belt Engine/generator	Shielded

When the Safety Assessment is completed and acknowledged by the local safety group, please send it to the KT safety manager including relevant appendices

Department of Chemical Engineering, DTU
Safety Assessment of experimental facilities

Analysis of deviations from normal operation

Use the following form as a tool to identify and evaluate risks. Add elements if necessary.

Setup: **Section:**		**Person:**	**Date:**
A. Problem/deviation	**B. Unwanted consequences of deviation**	**C. Measures taken to avoid unwanted consequences**	**D. Further actions to be taken to reach an acceptable situation**
1.Flow 1.1.Gas *1.1.1.Too much* *1.1.2. Too little* 1.2. Fluids *1.2.1.Too much* *1.2.2. Too little*	 Coolant Overheating	 Temperature alarm 95°C	
3.Temperature 3.1 Gas *3.1.1. Too high* *3.1.2.Too low* 3.2 . Fluids 3.2.1 *Too high* 3.2.2. *Too low*	 Coolant, oil Overheating	 Temperature alarm 95°C	
4. Pressure 4.1. Gas *4.1.1. Too high* *4.1.2. Too low* 4.2. Fluids *4.2.1 Too high* *4.2.2. Too low*	 Fuel/air mix Compressor/engine failure Oil Overheating	 Relief valve, pressure alarm 1 bar Temperature alarm 95°C	
5.Materials/chemicals (The considerations on materials can also be dealt with on APV's)			
5.1. Poisonous, Radioactive	CO	CO alarm 50ppm	

Setup: **Section:**		**Person:**	**Date:**
A. Problem/deviation	**B. Unwanted consequences of deviation**	**C. Measures taken to avoid unwanted consequences**	**D. Further actions to be taken to reach an acceptable situation**
or carcinogenic 5.2.Handling 5.3. Leakage 5.4. Disposal			
6. Ventilation 6.1. Cut-off 6.2 Reappearance	Engine overheating	Coolant temperature alarms 90℃	
7. Electrical power *7.1. Cut-off* *7.2. Reappearance*	None Sudden restart of engine/compressor	System going on standby at power cut-off	
8. Erroneous valve setting Exhaust sample valve Closed	Damage to emission analysis equipment	Put up a sign	
?			
?			
?			

C Gas engine announcement

BEK nr 1450 af 20/12/2012 (Gældende) Udskriftsdato: 2. marts 2015

Ministerium: Miljøministeriet
Journalnummer: Miljømin., Miljøstyrelsen, j.nr. MST-52100-00022

Senere ændringer til forskriften
Ingen

Bekendtgørelse om begrænsning af emission af nitrogenoxider og carbonmonooxid fra motorer og turbiner

I medfør af § 7, stk. 1, nr. 1, 2, 6 og 8, § 7 a, stk. 1, § 80, stk. 1, og § 110, stk. 3, i lov om miljøbeskyttelse, jf. lovbekendtgørelse nr. 879 af 26. juni 2010, som ændret ved lov nr. 446 af 23. maj 2012 og lov nr. 1149 af 11. december 2012, fastsættes:

Kapitel 1

Område

§ 1. Bekendtgørelsen fastsætter emissionsgrænseværdier, krav til kontinuerte målinger for NO_x m.v. for motorer og turbiner i faste installationer med en nominel indfyret termisk effekt på mindst 120 kW pr. motor eller turbine.

Stk. 2. Ved bestemmelse af den indfyrede effekt regnes med brændslets nedre brændværdi.

Stk. 3. Bekendtgørelsen omfatter motorer og turbiner til forbrænding af

1) naturgas,
2) LPG (Liquified Petroleum Gasses),
3) forgasningsgas,
4) biogas,
5) dieselolie,
6) gasolie,
7) fuelolie og
8) vegetabilsk olie.

Stk. 4. Bekendtgørelsen finder ikke anvendelse på:

1) Motorer og turbiner på anlæg med en samlet nominel indfyret termisk effekt på 50 MW og derover, som er omfattet af bekendtgørelse om begrænsning af visse luftforurenende emissioner fra store fyringsanlæg.
2) Motorer og turbiner til nødsituationer med færre end 500 driftstimer om året.
3) Motorer og turbiner på platforme på havet.

Kapitel 2

Definitioner

§ 2. I denne bekendtgørelse forstås ved:

1) Motor: En forbrændingsmotor, der fungerer efter ottoprincippet, og som anvender elektrisk tænding eller, når der er tale om dual-fuel-motorer eller dieselmotorer, kompressionstænding til forbrænding af brændstof.
2) Turbine: En roterende maskine, der omdanner termisk energi til mekanisk arbejde, og som hovedsageligt består af en kompressor, en termisk anordning, hvori brændslet oxyderes med henblik på at opvarme arbejdsmediet, og en turbine.

3) Motorer og turbiner til nødsituationer: Motorer og turbiner, der alene sættes i drift i tilfælde af havarier på produktionsanlæg eller ved udfald på transmissionsnettet.
4) Driftstimer: Det tidsrum udtrykt i timer hvor fyringsanlægget er helt eller delvis i drift og udleder emissioner til luften bortset fra opstarts- og nedlukningsperioder.
5) Ny motor eller turbine: Motor eller turbine der anmeldes eller godkendes og sættes i drift, efter denne bekendtgørelse er trådt i kraft.
6) Bestående motor eller turbine: Motor eller turbine der er anmeldt eller godkendt før den 7. januar 2013, og som er sat i drift senest 1 år efter dette tidspunkt.
7) AMS: Automatisk Målende System.

Kapitel 3

Bestemmelse af et anlægs samlede indfyrede termiske effekt

§ 3. Når røggasser fra to eller flere særskilte motor- eller turbineanlæg udledes gennem en fælles skorsten, anses en sådan kombination af anlæg for at være et enkelt motor- eller turbineanlæg, og deres samlede kapacitet betragtes under ét i forbindelse med beregningen af den samlede nominelle indfyrede termiske effekt.

Stk. 2. Hvis to eller flere særskilte motor- eller turbineanlæg installeres således, at røggasserne herfra, under hensyntagen til både tekniske og økonomiske forhold, efter godkendelses- eller tilsynsmyndighedens vurdering kan udledes gennem en fælles skorsten, anses en sådan kombination af anlæg for at være et enkelt fyringsanlæg, og deres samlede kapacitet betragtes under et i forbindelse med beregning af den samlede nominelle indfyrede termiske effekt.

Stk. 3. I forbindelse med beregning af den samlede nominelle indfyrede termiske effekt fra en kombination af fyringsanlæg som omhandlet i stk. 1 og 2 medregnes ikke særskilte fyringsanlæg med en nominel indfyret termisk effekt på under 120 kW.

Kapitel 4

Grænseværdier

§ 4. Nye motorer og turbiner skal overholde emissionsgrænseværdierne i bilag 1, tabel 1, for motorer og tabel 3 for turbiner.

Stk. 2. Bestående motorer og turbiner skal overholde emissionsgrænseværdierne i bilag 1, tabel 2, for motorer og tabel 4 for turbiner senest den 7. januar 2021.

Stk. 3. Tilsynsmyndigheden kan bestemme, at projekter til udvikling af ny teknologi fritages midlertidigt fra overholdelse af emissionsgrænseværdierne i bilag 1.

§ 5. Emissionsgrænseværdier for motorer og turbiner, der benytter to eller flere brændselstyper, beregnes som angivet i bilag 1.

Kapitel 5

Anmeldelse

§ 6. Den, der installerer en motor eller turbine, skal indgive anmeldelse til tilsynsmyndigheden, når den enkelte motor eller turbine har en indfyret termisk effekt over 120 kW, og anlæggets samlede indfyrede termiske effekt er mindre end 5 MW. Dog finder reglerne i bekendtgørelse om godkendelse af listevirksomhed anvendelse, hvis anlæggets samlede indfyrede termiske effekt er større end 1 MW, og motoren eller turbinen anvender biogas eller forgasningsgas fra forgasningsanlæg, der anvender biomasseaffald, som defineret i bekendtgørelse om biomasseaffald.

Stk. 2. Anmeldelsen skal være tilsynsmyndigheden i hænde, senest otte uger før anlægget tages i brug.

§ 7. Anmeldelsen skal indeholde dokumentation for, at emissionsgrænseværdierne i bilag 1, tabel 1, henholdsvis tabel 3, kan overholdes.

Stk. 2. Dokumentation skal foreligge i form af målinger, der er udført på en teknisk identisk motor eller turbine, samt eventuel rensningsteknologi.

Stk. 3. Målinger skal udføres som akkrediteret teknisk prøvning, og målerapporterne skal udfærdiges som akkrediterede prøvningsrapporter. Målelaboratoriet skal være akkrediteret til bestemmelse af de aktuelle stoffer i røggassen af Den Danske Akkrediterings- og Metrologifond eller et tilsvarende akkrediteringsorgan, som er medunderskriver af EA's multilaterale aftale om gensidig anerkendelse.

Kapitel 6

Kontrol og rapportering

§ 8. Nye motorer og turbiner, der hver især har en indfyret termisk effekt større end 10 MW, skal være forsynet med AMS til måling af NO_x.

Stk. 2. Bestående motorer og turbiner, der hver især har en indfyret termisk effekt større end 10 MW, skal være forsynet med AMS til måling af NO_x senest fra den 7. januar 2021.

Stk. 3. Stk. 1 og 2 finder ikke anvendelse på enkeltanlæg med en indfyret termisk effekt større end 10 MW, hvis det årlige antal driftstimer er under 500 som et rullende gennemsnit over 5 år.

Stk. 4. Til kontrol af grænseværdierne i bilag 1 skal der føres egenkontrol i overensstemmelse med bilag 2.

§ 9. Virksomheden skal hvert år inden den 1. februar indsende resultaterne af AMS-kontrollen for NO_x-emissionen på enkeltanlæg, der er omfattet af § 8 til tilsynsmyndigheden. Opgørelsen skal ske for hver måned i det forudgående kalenderår. For anlæg, der er omfattet af bekendtgørelse om godkendelse af listevirksomhed, kan godkendelsesmyndigheden fastsætte vilkår om en anden fremsendelsesfrist for måleresultater, hvis dette er mere hensigtsmæssigt for rapporteringen af den samlede egenkontrol på virksomheden.

Stk. 2. For enkeltanlæg med en indfyret termisk effekt større end 10 MW og med et årligt antal driftstimer under 500 som et rullende gennemsnit over 5 år skal virksomheden årligt rapportere driftstimeantallet til tilsynsmyndigheden.

§ 10. På motorer og turbiner med en samlet indfyret motor- eller turbineeffekt over 5 MW, dog 1 MW hvis motor- eller turbineanlægget anvender biogas eller forgasningsgas fra forgasningsanlæg, der anvender biomasseaffald, som defineret i bekendtgørelse om biomasseaffald, skal der foretages præstationsprøvninger, jf. bilag 2, med henblik på at dokumentere, at grænseværdierne for CO og NO_x i bilag 1 er overholdt.

Stk. 2. For anlæg under 100 driftstimer skal der kun måles en gang efter anlægget er sat i drift. Målingen skal foreligge senest 6 måneder efter et nyt anlæg er taget i brug eller i forbindelse med revurdering af godkendelsen for et eksisterende anlæg. Alternativt kan dokumentationen foreligge i form af en måling, der er udført på en teknisk identisk motor eller turbine, samt eventuel rensningsteknologi. For anlæg fra 100 til og med 1500 driftstimer måles hvert tredje år. For anlæg mellem 1500 og til og med 3000 driftstimer måles hvert andet år. For anlæg med over 3000 driftstimer måles hvert år. Driftstimerne opgøres som et rullende gennemsnit over 5 år.

Stk. 3. De i stk. 1 nævnte målinger skal udføres som akkrediteret teknisk prøvning, jf. bilag 2, og målerapporterne skal udfærdiges som akkrediterede prøvningsrapporter. Målelaboratoriet skal være akkrediteret til bestemmelse af de aktuelle stoffer i røggassen af Den Danske Akkrediterings- og Metrologifond eller et tilsvarende akkrediteringsorgan, som er medunderskriver af EA's multilaterale aftale om gensidig anerkendelse.

Stk. 4. Rapport over målingerne skal indsendes til tilsynsmyndigheden senest to måneder efter, at de er foretaget.

Kapitel 7

Tilsyn

§ 11. Kommunalbestyrelsen påser, at reglerne i denne bekendtgørelse overholdes.

Stk. 2. Miljøstyrelsen er dog tilsynsmyndighed for fyringsanlæg på virksomheder, hvor Miljøstyrelsen er godkendelsesmyndighed.

Kapitel 8

Straffebestemmelser

§ 12. Medmindre højere straf er forskyldt efter den øvrige lovgivning, straffes med bøde den, der

1) overtræder § 4, stk. 1 og 2,
2) undlader at foretage anmeldelse efter § 6,
3) undlader at installere AMS-udstyr for NO_x efter § 8, stk. 1 og 2,
4) undlader at gennemføre egenkontrol efter § 8, stk. 4,
5) undlader at indsende resultater af AMS-kontrollen efter § 9 eller § 10, stk. 4, eller
6) undlader at foretage målinger efter § 10, stk. 1.

Stk. 2. Straffen kan stige til fængsel i indtil 2 år, hvis overtrædelsen er begået forsætligt eller ved grov uagtsomhed, og hvis der ved overtrædelsen er

1) voldt skade på miljøet eller fremkaldt fare derfor eller
2) opnået eller tilsigtet en økonomisk fordel for den pågældende selv eller andre, herunder ved besparelse.

Stk. 3. Der kan pålægges selskaber m.v. (juridiske personer) strafansvar efter reglerne i straffelovens kapitel 5.

Kapitel 9

Ikrafttrædelses- og overgangsbestemmelser

§ 13. Bekendtgørelsen træder i kraft den 7. januar 2013.

Stk. 2. Bekendtgørelse nr. 621 af 23. juni 2005 om begrænsning af emission af nitrogenoxider, uforbrændte carbonhybrider og carbonmonooxid m.v. fra motorer og turbiner ophæves.

§ 14. Vilkår om overholdelse af emissionsgrænseværdier og måling af UHC og lugt fastsat i godkendelser udstedt før den 7. januar 2013 bortfalder.

Miljøministeriet, den 20. december 2012

IDA AUKEN

/ Michel Schilling

Bilag 1

Emissionsgrænseværdier

Alle emissionsgrænseværdierne i tabel 1 - 4 er angivet ved referencetilstanden (mg/normal m^3), som er tør røggas omregnet til 15% O_2, 0 ºC og 101,3 kPa. NO_x er summen af NO og NO_2 i røggassen. NO regnes vægtmæssigt som NO_2.

Grænseværdier for motorer

Tabel 1. Emissionsgrænseværdier for nye motorer med en samlet indfyret termisk effekt fra 120 kW til 50 MW

Brændsel	**Over 120 kW og til og med 5 MW termisk indfyret effekt**		**Over 5 og under 50 MW termisk indfyret effekt**	
	NO_x	CO	NO_x	CO
Naturgas, LPG, biogas og forgasningsgas	190	190*	115	190*
Dieselolie, gasolie, fuelolie og vegetabilsk olie	190	190	115	190

* Dog er grænseværdien for biogas 450 og for forgasningsgas 1125.

Tabel 2. Emissionsgrænseværdier for bestående motorer med en samlet indfyret termisk effekt fra 120 kW til 50 MW

Brændsel	**Over 120 kW og til og med 5 MW termisk indfyret effekt**		**Over 5 og under 50 MW termisk indfyret effekt**			
	Fra den 7. januar 2013		Indtil den 7. januar 2021		Fra den 7. januar 2021	
	NO_x	CO	NO_x	CO	NO_x	CO
Naturgas og LPG	205	190	205	190	115	190
Biogas	375	450	375	450	190	450
Forgasningsgas	205	1125	205	1125	190	1125
Dieselolie, gasolie, fuelolie og vegetabilsk olie	205	190	205	190	115	190

Grænseværdier for turbiner

Tabel 3. Emissionsgrænseværdier for nye turbiner med en samlet indfyret termisk effekt fra 120 kW til 50 MW

Brændsel	**Over 120 kW og til og med 50 MW termisk indfyret effekt**

	NO_x	CO
Naturgas, LPG, biogas og forgasningsgas	75	100
Dieselolie, gasolie, fuelolie og vegetabilsk olie	75	100

Tabel 4. Emissionsgrænseværdier for bestående turbiner med en samlet indfyret termisk effekt fra 120 kW til 50 MW

Brændsel	**Indtil den 7. januar 2021**		**Fra den 7. januar 2021**	
	NO_x	CO	NO_x	CO
Naturgas og LPG	75	56	75	100
Biogas	110	80	75*	100
Forgasningsgas	110	80	75*	100
Dieselolie, gasolie, fuelolie og vegetabilsk olie	75	56	75	100

* For turbiner, der er anmeldt til tilsynsmyndigheden før den 6. juli 2005, eller som er godkendt efter miljøbeskyttelseslovens § 33 før samme dato, og hvor turbinen ikke er i drift mere end 1500 timer som rullende gennemsnit over en femårs periode, er emissionsgrænseværdien 110 mg/normal m^3.

Grænseværdier for motorer og turbiner, der benytter to eller flere brændselstyper samtidigt

For motorer og turbiner, der benytter to eller flere brændselstyper samtidig, beregnes emissionsgrænseværdien af virksomheden ved at benytte emissionsgrænseværdierne for hver brændselstype og hvert forurenende stof, som angivet i bilag 1, tabel 1 – 4, at de brændselstypevægtede emissionsgrænseværdier bestemmes ved at gange hver af de relevante emissionsgrænseværdier med den indfyrede termiske effekt fra hver brændselstype og dividere resultatet af hver multiplikation med summen af den indfyrede termiske effekt fra samtlige brændselstyper, og at de brændselstypevægtede grænseværdier lægges sammen til en grænseværdi.

Bilag 2

Krav til egenkontrol

A. Måling af emissioner

1. Kontinuerte målinger

AMS-målere, der opfylder præstationskrav i DS/EN 15267-3 eller tilsvarende standarder, vil kunne anvendes. Andre målere kan anvendes, hvis de med hensyn til kvalitet og nøjagtighed svarer til ovennævnte målere.

AMS-måling af NO_x, der gennemføres i medfør af § 8, skal desuden omfatte måling af røggassens iltindhold, temperatur, vanddampindhold og tryk. Måling af vanddampindholdet i røggassen er ikke nødvendig, forudsat at gasprøven tørres, inden emissionerne analyseres. Måling af vanddampindhold og tryk kan erstattes af beregnede eller konstante værdier, når det er dokumenteret, at de er repræsentative.

AMS skal overholde følgende kvalitetskrav udtrykt som den maksimale usikkerhed (95 % konfidensinterval): 20 % af grænseværdien for NO_x.

Kvalitetssikring af AMS skal gennemføres i overensstemmelse med principperne i EN14181. AMS skal ved ibrugtagning kalibreres (QAL2 omfattende 5 parallelle målinger udført over én dag).

Derefter underkastes AMS kontrol med parallelle målinger efter referencemetoder (AST omfattende 3 parallelle målinger) hvert 3. år. AMS skal gennemgå en årlig kontrol og et årligt serviceeftersyn (funktionstest uden linearisering). AMS efterses og justeres med kalibreringsgasser efter leverandørens anvisninger (som erstatning for QAL3).

Andre metoder (f.eks. PEMS) til kontinuert måling af NO_x kan anvendes, hvis der er en tilsvarende sikkerhed for, at målingen af den udledte mængde af NO_x, regnet som NO_2, er som ved AMS-målingen. Den alternative metode skal kvalitetssikres og kontrolleres efter principperne i EN14181, som beskrevet for AMS, i det omfang det er teknisk muligt.

2. Præstationskontrol for NO_x og CO

Ved præstationskontrol foretages to enkeltmålinger hver af en varighed på 45 minutter med henblik på at dokumentere, at emissionsgrænseværdierne for NO_x og CO er overholdt. Dette gælder dog ikke for NO_x, hvis der er udført AMS-kontrol. Målingerne skal foretages under repræsentative driftsforhold. Præstationsmålingerne skal ikke udføres under opstart og nedlukning.

B. Prøvetagningsmetoder samt kvalitetssikring af AMS

Prøvetagning og analyse skal ske efter de metoder, der er nævnt i nedenstående tabel eller efter internationale standarder af mindst samme analysepræcision og usikkerhedsniveau.

Tabel 5. Metoder for prøvetagning og analyse samt kvalitetssikring af AMS

Navn	**Parameter**	**Metodeblad nr. ***
Bestemmelse af koncentrationer af kvælstofoxider (NO_x) i strømmende gas	NO_x	MEL-03
Bestemmelse af carbonmonooxid (CO) i strømmende gas	CO	MEL-06
Kvalitetssikring af Automatisk Målende Systemer (AMS)	QA af AMS	MEL-16

* Se hjemmesiden for Miljøstyrelsens Referencelaboratorium for måling af emissioner til luften: www.reflab.dk

C. Overholdelse af grænseværdier

1. Kontinuerte målinger

NO_x emissionsgrænseværdier, der måles for ved AMS-kontrol, anses for overholdt, når det aritmetiske gennemsnit af samtlige 1-timesmålinger i løbet af kontrolperioden er mindre end eller lig med grænseværdien. Kontrolperioden er en kalendermåned. Dog regnes perioder uden emission af det pågældende stof ikke med til kontrolperioden.

Usikkerheden på et enkelt måleresultat udtrykt som værdien af 95 %-konfidensintervallet må ikke overskride 20% af emissionsgrænseværdien for NO_x. Timegennemsnitsværdier bestemmes som de målte timegennemsnitsværdier efter fratrækning af værdien af konfidensintervallet.

2. Præstationskontrol for NO_x og CO

Emissionsgrænseværdierne under normal drift, dvs. drift uden for opstart og nedlukning, anses for overholdt, når det aritmetiske gennemsnit af alle enkeltmålinger udført ved præstationskontrollen er mindre end eller lig med emissionsgrænseværdien.

D Emission analysis equipment calibration report

19-20 november 2014
Anders Tiedje

Kalibrering af ABB URAS CO/O2

CO: Range CO: 0-10000ppm

Kalibreret nul-punkt efter flow af ren nitrogen i ca. 10 minutter.

Span gas for CO: 3800ppm CO i nitrogen (indeholder også 3800ppm CO2 og 4,50% O2).

Før kalibreringen viste analysatoreren ca. 2570ppm med span gassen (3800ppm CO) og der kom en advarsel om ”Half drift limit exceeded error”, billede 1 og 2 (det er besked nr. 304 på side 70 i manualen).

For at komme videre foretog jeg en ”Basic calibration”. For at få lov til det, skal bruges en adgangs kode som er 081500 for disse analysatorer.

Herefter foretaget almindelig kalibering af nul og span.

O_2: Range O2: 0-25%

Kalibreret nul-punkt efter flow af ren nitrogen i ca. 10 minutter.

Span gas for O_2: 9,50% O_2 i nitrogen (indeholder også 4,50% CO og 18,9% CO_2).

Før kalibreringen viste analysatoreren en advarsel om ”Half drift limit exceeded error” (det er besked nr. 304 på side 70 i manualen).

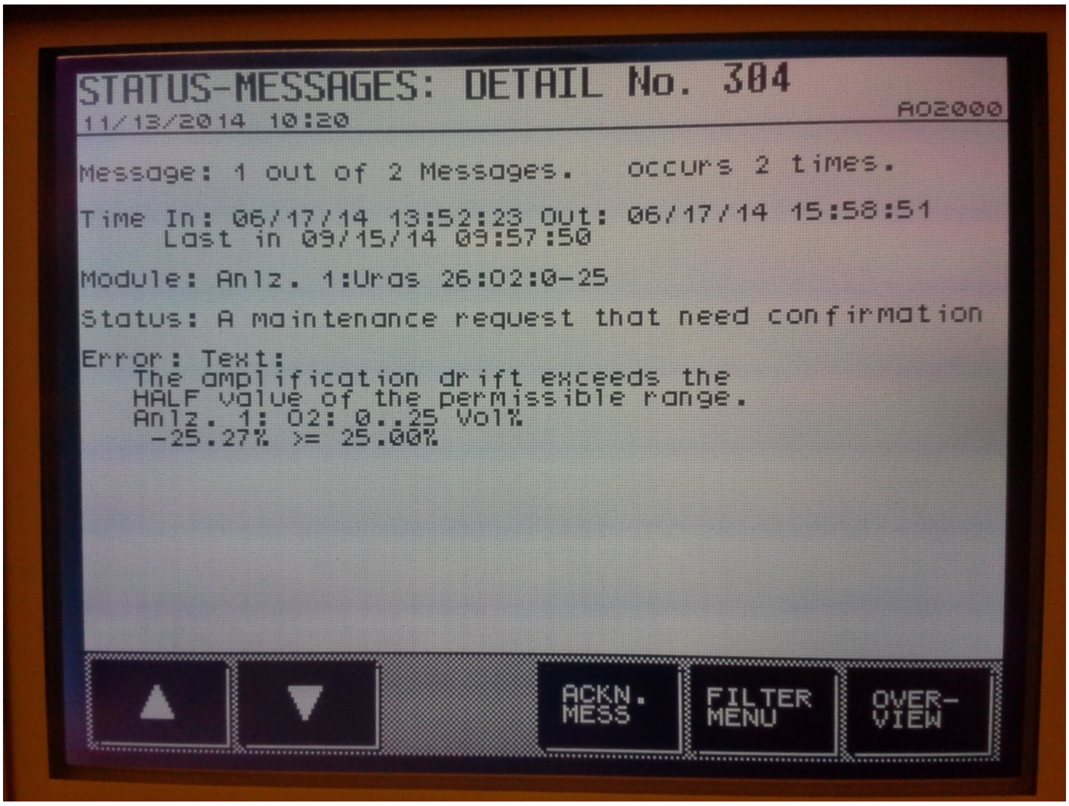
For at komme videre foretog jeg en ”Basic calibration”. For at få lov til det, skal bruges en adgangs kode som er 081500 for disse analysatorer.

Herefter foretaget almindelig kalibering af nul og span.

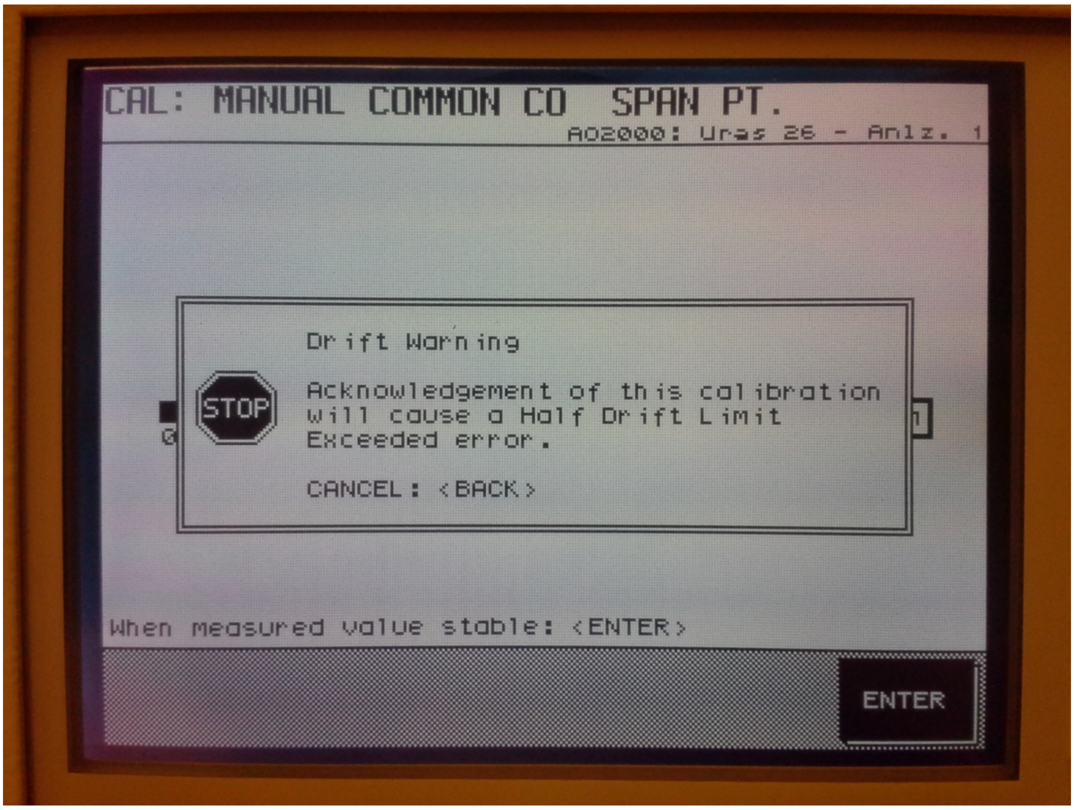
Efter denne kalibrering blev igen skyllet med N_2, men efter et par minutter stod CO-kanalen stadig og skiftede mellem hhv. -50000ppm og -99999ppm. Muligvis som følge af en ”overdosis CO fra span-gassen?

Lavede herefter en ”Calibration reset” (billede 3), ny ”basic calibration” og almindelig kalibrering af denne kanal.

Begge kanaler er nu kalibrerede.



Billede 1: Message nr. 304.



Billede 2: Drift warning.

Billede 3: Calibration reset.

Kalibrering af ABB LIMAS NO_2/NO

NO: Range NO: 0-2000ppm

Kalibreret nul-punkt efter flow af ren nitrogen i ca. 10 minutter.

Span gas for NO: 906ppm NO i nitrogen (indeholder også 449ppm SO_2). Kalibrering OK.

Havde et meget lille off-set for kalibreringen (ca. 10ppm).

NO_2: Range NO_2: 0-2000ppm

Kalibreret nul-punkt efter flow af ren nitrogen i ca. 10 minutter.

Span gas for NO_2: 2000ppm i nitrogen. Kalibrering OK.

Begge kanaler er nu kalibrerede.