The Performance Of a Dual Fuel HCCI Engine

Master thesis / Proyecto final de carrera

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A mi familia y amigos

Abstract

The HCCI engine has shown great qualities such as high efficiency combined with low emissions. However, many issues have not totally been solved yet. One way to control the combustion timing, one of the commented issues, is dual fuel utilizing the difference in ignition temperature between different fuels.

So, from this point on it is not difficult to think about developing a big mapping of one HCCI engine, controlling this variable to obtain the best performance of this kind of engine. After doing that, the most interesting ranges of operation will be known as well as the possibility of studying them more in depth.

The test engine is a modified six cylinder Scania DSC12 turbo charged diesel engine. All experiments are conducted with natural gas and n-heptane; the natural gas with nearly 120 in the octane scale (difficult ignition) and the n-heptane, zero in the octane scale (really easy ignition). This is one of the keys to control the combustion.

In order to study the performance of the dual fuel engine, the operating parameters as engine speed, inlet pressure, inlet temperature, load and, of course, the combustion timing, are varied and analyzed reading emissions, efficiencies, fuel consumption, variability, losses … in all their ranges. Afterwards in the points which exceeded the limits regarding the in cylinder pressure rise rate and NO_x emissions the engine has been run with the help of the turbo charger varying the inlet air pressure or, in other words, the inlet area of its turbine. In this way the NO_x emissions do not reach an undesirable rate and the engine is always working under 15 bars of pressure rise per crank angle degree.

The test results show a higher efficiency close to 50 % and low emissions in the entire mapping using the turbo charger in the suitable condition. On the other hand, more CO and HC are produced. They can however be reduced with an oxidizing catalyst.

Acknowledgements

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Rafael Vargas Muñoz

Nomenclature

ABDC	After Bottom Dead Center
ATDC	After Top Dead Center
BBDC	Before Bottom Dead Center
BSFC	Brake Specific Fuel Consumption
BTE	Brake Thermal Efficiency
BTDC	Before Top Dead Center
CA50	Crank Angle of 50 % heat released
CAD	Crank Angle Degree
CI	Compression ignition
CO	Carbon monoxide
CombEff	Combustion efficiency
CO_2	Carbon Dioxide
CR	Compression Ratio
DI	Direct Injection
EGR	Exhaust Gas Recirculation
EVO	Exhaust Valve Opening
FMEP	Friction Mean Effective Pressure
FuelMEP	Fuel Mean Effective Pressure
HC	Hydrocarbons (unburned)
HCCI	Homogeneous Charge Compression Ignition
IMEP	Indicated Mean Effective Pressure
IVC	Inlet Valve Closing
MBT	Maximum Brake Torque
ms	Milliseconds
Nm	Newton metre
NO_x	Nitrogen oxides
PFI	Port Fuel Injection
PM	Particulate Matter
PPC	Partially Premixed Combustion
Pr	Prandtl number
RoHR	Rate of Heat Release
Rpm	Revolutions per minute
SI	Spark Ignition
SOI	Start of injection
THC	Total Hydrocarbons (unburned)

Greek symbols:

ν	Kinematic viscosity
λ	Air/fuel equivalence ratio
ϕ	Fuel/air equivalence ratio

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1 INTRODUCTION

In this initial part, a short general background regarding the subject of my project is explained, before introducing this master thesis. Then, the main objective of the project and the carried out method will be presented. Finally, this work's limitations will be mentioned.

For the last two or three decades, no more, the environment has been a concern for the world we live in, and our concern is getting bigger each day. For this reason, all the exhaust emissions (engines, machines, factories...) are starting to be controlled by the legislation. The automobile research is trying to develop engines with almost zero pollution because the car engines are one of the principal sources of the emissions to the atmosphere. But these engines are not only to reduce the emissions but also have to get enough power to work throughout the obtainable load/speed range.

Homogeneous charge compression ignition, HCCI, is a hybrid of the well-known spark ignition and compression ignition engines. This engine started to be developed around the 80´s and it might be a step closer to the engine of the future. The primary purpose of using HCCI combustion in 2 stroke engines was to reduce the HC emissions at part load operation. Later studies on 4 stroke engines have shown that it is possible to achieve high efficiency and low NO_x emissions avoiding throttling of intake air and using high compression ratio and a highly diluted mixture that keeps the temperature low during combustion. The drawback of this is that more HC is produced.

The engine developed seems very attractive due to the low emissions produced and the good qualities presented. But it has some hard challenges too as there is no direct means to control the combustion timing. Using dual fuel (liquid and gas with different ignition points) is one possible way of gaining indirect control of the ignition timing.

To investigate the operation range, in terms of air/fuel ratio, engine load and emissions characteristics of HCCI operation, several extensive tests have been performed within this project.

1.1 Objective

The objective of this master thesis work is to evaluate the performance of the Scania DSC12 based HCCI engine. Engine performance will be evaluated throughout the obtainable load/speed range of the engine to try to find the best operative conditions.

Operating characteristics such as exhaust gas emissions, fuel-consumption, efficiencies, losses, torque, combustion duration, maximum net indicated efficiency, timing and several heat-release analysis related parameters will be examined and mapped for the operational regime of the engine.

1.2 Method

According to the objective, an operational strategy based on physical limitations will be developed in order to find a preferable engine operation strategy. Operating parameters are varied in order to study every possible configuration in the operational range. Within each point some optimization of the operational parameters will be needed in order to optimize the engine performance.

All the data is obtained with different logger programs, most of them designed in the department, and analyzed with the help of Matlab. These programmes are also used in order to put together and compare the performance maps.

1.3 Limitations

First of all, the master thesis readers have to know that this work has been developed in the research area. The engine is experimental and this means that it is not always guaranteed that it will work.

To achieve the objective of this master thesis, the engine has worked throughout the operational load/speed range but without actually knowing where the exact limits were. This means that the engine has sometimes been forced to work in precarious and inadequate conditions; suffering and causing sometimes damages to it.

Moreover there are three different types of limits. The most important is the load (IMEPn), which without the turbo charger help has a range between 2 (it is not able to work with lower load because the engine would have a really bad combustion) and 5 bars (because the chamber pressure is very high). However, by applying supercharging via turbo charger, 10 bars more or less have been obtained for the fastest speeds.

The other limits are due to the speed and the time at which the combustion starts, but they are not so significant as the load.

2 THE HCCI ENGINE

One easy way to define the HCCI engine is to consider it a hybrid between the Spark Ignition engine (SI) and the Diesel engine. It runs with a premixed air-fuel charge just like the SI engine and it is ignited by the compression heat as in a Diesel engine. This work explains the definition mentioned above.

The first studies known of HCCI combustion were made by Onishi in 1979 on two stroke engines in an effort to reduce the hydrocarbon (HC) emissions at part load operation and to decrease fuel consumption by stabilizing the combustion of lean mixtures. Since the end of the 80's and 90's HCCI research has been vastly intensified.

In this chapter, the SI engine and Diesel engine concepts are reviewed to be able to understand how they work so that the reader unfamiliar with these concepts will get a background to compare with the HCCI engine. Afterwards, the HCCI concept is explained, concluded by its advantages and disadvantages.

2.1 The principle of the SI engine

The SI engine works according to the Otto cycle where we can divide it into four-stroke principle as is shown in Figure 2.1.

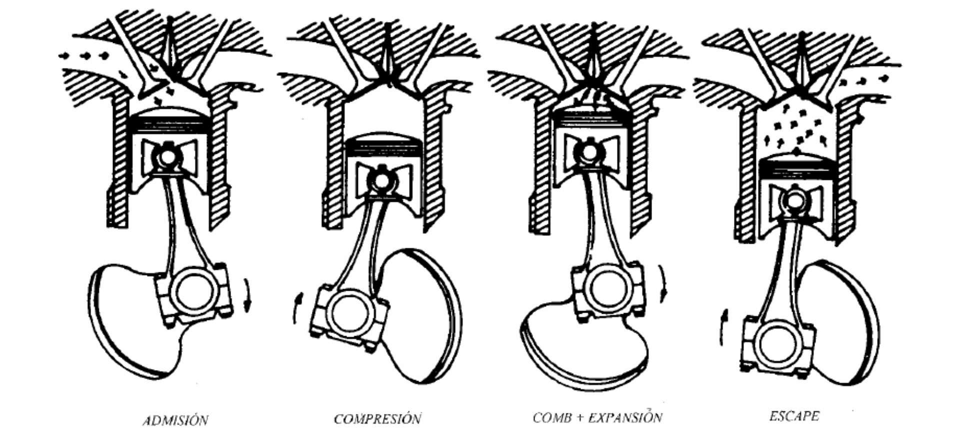


Figure 2.1 Four stroke cycle

1. Intake: The fuel mixture is atomized, vaporized and mixed with the air that is drawn into the cylinder through the intake valve. The mixing process continues and a close to homogeneous mixture is created.
2. Compression: The mixture is compressed by the piston and ignited by a spark plug at the end of this stroke. The spark causes a flame that starts to propagate through the combustion chamber and the pressure rises.
3. Expansion: The combustion continues until all fuel is burned while the piston moves away.
4. Exhaust: The exhaust valve opens and the piston evacuates the exhausts.

The load of a SI engine is controlled by the amount of charge drawn into the combustion chamber. This control is achieved by the use of a throttle in the intake system. This throttling causes quite high pumping losses at part load and idling operation of the engine. As nowadays an engine is mostly operated at part load, the efficiency of a SI engine is poor compared to a Diesel engine, which works without a throttle. Due to the risk of knocking, a SI engine has lower compression ratio than a Diesel engine; that causes also lower engine efficiency.

The power density of this engine is quite good and moreover it is very clean if the exhaust gas passes through a three-way catalyst. It does not produce any soot neither.

2.2 The principle of the Diesel engine

The typical Diesel engine, also called compression ignition (CI) engine, works mainly according to the four-stroke principle and is direct injected.

In the intake stroke, pure air is drawn into the combustion chamber and then compressed during the compression stroke. At the end of this stroke the fuel is injected at high pressure into the hot compressed air. The fuel atomizes and starts vaporising, creating a partially premixed zone. When in-cylinder pressure and temperature reach the fuel’s ignition condition, the auto ignition starts the combustion. First the premixed part of the charge burns rapidly. A steep rise in pressure is the consequence; it causes the typical knocking diesel sound. Later on, the mixture formation is still proceeding and the diffusion combustion occurs. The main amount of mixture is burnt here. At the end of the combustion, pressure and temperature in the flame drop and the chemistry gets slow compared to the mixing processes taking place at the same time.

The combustion rate is said to be mixing controlled, because it depends on the mixing rate of the air and fuel. This rate can be improved by higher pressure, multi-hole nozzles and higher turbulence in the cylinder.

The load of the Diesel engine is controlled by changing the amount of injected fuel. For turbocharged diesel engines, the airflow increases with higher engine load and thus boosts pressure.

The Diesel engine has higher part load efficiency than the SI engine because of the low pumping losses due to the lack of throttle. The high compression ratio of the Diesel engine also contributes to the high efficiency. On the other hand, the biggest disadvantages of the Diesel engine are the soot production combined with high NO_x production, which is caused by a high combustion temperature. Usage of a $DeNO_x$ catalyst however can reduce NO_x.

2.3 The principle of the HCCI engine

As we presented before, the HCCI engine has characteristics from each of the most popular forms of combustion used, the SI and the Diesel engine. HCCI can be achieved both with two and four stroke engines. In this present report the four-stroke engine is examined.

Like the SI engine, the HCCI engine injects the fuel in the intake manifold. A homogeneous or close to homogeneous mixture is created and it is drawn into the combustion chamber. This mixture is highly compressed by the piston leading to such high temperature and pressure that auto-ignition occurs like in the Diesel engine. Combustion initiation occurs, where the conditions for auto-ignition are most favourable, usually at several locations simultaneously.

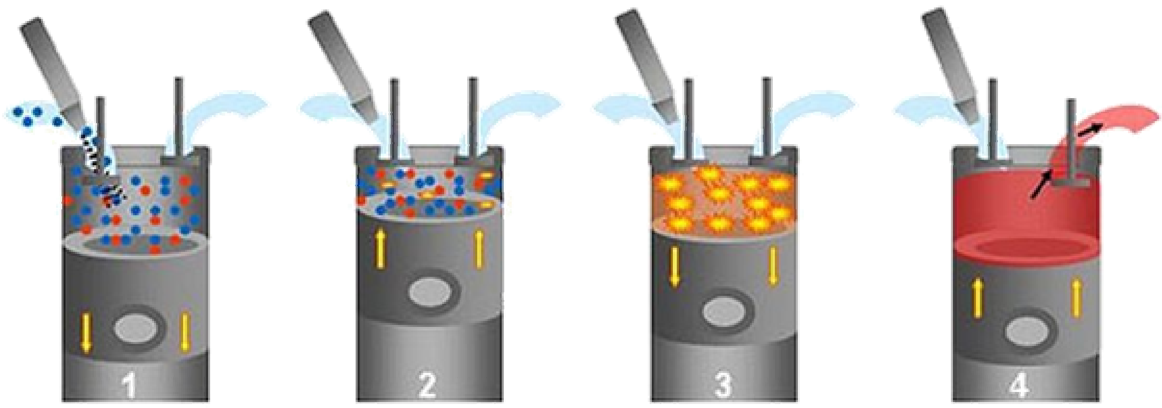


Figure 2.3 HCCI process

The HCCI combustion shows no flame propagation like the SI and Diesel engine. Instead the whole charge burns almost homogeneously and quite rapidly. To avoid a too fast combustion highly diluted mixtures must be used. This can be achieved by excess air and/or with EGR (Exhaust Gas Recirculation).

The conditions to achieve auto-ignition require a high-compression ratio and/or preheating of the intake air. In order to set the auto-ignition timing correctly at a desired crank angle position, a certain combination of the parameters compression ratio, fuel quality, inlet temperature and inlet pressure has to be used.

As HCCI has no flame propagation, the turbulence is not as important as for the Diesel and SI engines. The homogeneous combustion is controlled mainly by chemical kinetics, so that the combustion rate is strongly affected by the air/fuel ratio and the temperature instead of the turbulence.

The engine load of the HCCI engine is controlled by changing the amount of fuel injected and thus changing the air/fuel ratio, λ, similar to the Diesel engine. Therefore no throttle is needed and the engine always operates with full airflow. At the moment HCCI works best for λ between 2 and 5, which means that the power density is only about half as high as for SI engines. But using very high EGR rates (> 50%), HCCI operation becomes possible up to stoichiometric conditions ($\lambda \approx 1$).

2.3.1 - HCCI – The advantages

The HCCI combustion mode offers two important advantages compared to SI and Diesel, which makes its use and development interesting for both engine types.

Compared to the SI engine the HCCI engine has a higher efficiency due to unthrottled operation, higher C_p/C_v when it is running with lean mixture, faster combustion (it could be a drawback at high load), low heat losses due to lower combustion temperature and often higher compression ratio.

Compared to the Diesel, engine the HCCI engine has lower emissions of NO_x and PM due to the homogeneous, diluted combustion.

To make HCCI operation possible, a complete re-design of the engine is not necessary. In principle, the HCCI concept defined earlier can be run on a standard SI engine with modifications mainly related to the valve train and a switch from port injection to direct injection. However, it is possible that the HCCI engines may require new features compared with conventional engines to extend the HCCI operating range and to control the engine under transient and mode switching operation.

2.3.2 - HCCI – The disadvantages

Engine control is a hard challenge for the HCCI engine, since there is no possibility of direct ignition timing control. For SI and Diesel engines the need for combustion monitoring is rather small, because a means of direct ignition control is used in both engine types. The ignition timing of the SI engine is controlled by the spark timing whereas in the Diesel engine it is controlled by the injection timing.

In the HCCI engine, the control of a number of different parameters such as inlet air temperature, compression ratio, fuel reactivity and amount of EGR requires a complex control method for keeping a stable ignition timing.

Another disadvantage is due to the low combustion temperature, crevice effects and the wall quenching that leaves some HC unburned as well as incomplete oxidation, which produces CO emissions.

One more disadvantage is related to noise, vibrations and harshness; as the upper load boundary for HCCI operation is often defined by the maximum rate of pressure rise that can be allowed during combustion, which if exceeded causes excessive noise.

3 EMISSIONS

In the next paragraphs the most important emissions from combustion engines, which are harmful for the nature, will be explained.

3.1 Nitrogen oxides

Nitrogen oxides, NO and NO_2, are often given the common designation NO_x. The impact of NO_x on the nature is that it is a factor in forming acid rain and photochemical smog. There are three major sources of NO_x emissions from combustion: thermal, prompt and fuel NO_x. Prompt NO_x is a term for nitrogen oxides formed directly in the flame front, fuel NO_x is formed from the nitrogen contained in the fuel reacting with the oxygen in the air. The formation of thermal NO_x is very temperature dependent, 1800 K is generally regarded as the combustion temperature over which NO_x production becomes significant and increasing the temperature gives a high production rate increase due to the exponential temperature dependence. Obviously, the temperature is not the only factor in the NO_x formation; oxygen must be available as well.

Finally, it is important to add that for an HCCI engine, NO_2 is often the dominating NO_x emissions and NO is only the small part.

3.2 Carbon monoxide

Carbon monoxide, CO is formed, as an intermediate step towards CO_2 in combustion, however if the combustion is incomplete CO will be present in the exhaust gases. Oxidizing CO to CO_2 releases energy; so naturally, if CO is present in the exhaust it indicates that not all available energy has been released during combustion. The rate of the CO to CO_2 reactions is very sensitive to the peak combustion temperature and as HCCI operates with low temperature combustion, CO emissions can be a problem. The minimum temperature required for complete combustion is around 1500 K.

Carbon monoxide causes suffocation in humans by blocking the haemoglobin from adsorbing oxygen.

3.3 Carbon dioxide

Carbon dioxide forms as a product from complete combustion; it is widely regarded as the most important greenhouse gas, a factor in global warming. The amount of CO_2 in the exhaust of an engine is related to the amount of fuel that was combusted; so reducing fuel consumption is a way to reduce carbon dioxide emissions.

3.4 Unburned hydrocarbons

Unburned hydrocarbons are obviously unwanted in the exhaust as this is due to incomplete combustion and basically is a waste of fuel. The obvious incomplete combustion case is when the fuel/air mixture does not ignite, but there are other ways in which fuel can escape combustion. One of these ways is when operating very lean; fuel can also enter the crevice between the cylinder and the piston and as the combustion is quenched near walls, this fuel does not participate in combustion but can become a part of the emissions, however this is not a problem for diesel engines no fuel is present near the crevices.

Unburned hydrocarbons are poisonous and sometimes carcinogenic.

3.5 Soot

Soot consists mainly of carbon particles. Unburned or partially oxidized hydrocarbons can stick to the surface of these particles. Soot is carcinogenic and harmful to the respiratory organs. Engine out emissions of soot depends on two basic processes, one is the total amount of soot that is formed during combustion and the other is the soot oxidation process that always occurs when excess air is available.

4 THEORETICAL CONCEPTS

Here the equations and definitions of the most important variables and parameters used are shown. All variables are given in SI units, except for the engine speed, n, which is given in rpm. The equations are valid for four stroke engines.

4.1 Engine fundamentals

<u>λ - Relative air/fuel ratio</u>, which is defined as ratio between the actual air/fuel ratio and the stoichiometric air/fuel ratio.

$$\lambda = \frac{\left(\dfrac{\dot{m}_{Air}}{\dot{m}_{Fuel}}\right)_{actual}}{\left(\dfrac{\dot{m}_{Air}}{\dot{m}_{Fuel}}\right)_{Stoich}} \tag{4.1.1}$$

$\dot{m}_{Air}$ (Kg/s) = mass flow of air

$\dot{m}_{Fuel}$ (Kg/s) = mass flow of fuel

λ larger than 1 means fuel lean mixture, in fact excess of air.

<u>MEP – Mean Effective Pressure</u>. Engine load can be expressed as torque or MEP. Engine torque is measured in the unit Newton metre but MEP is measured in pressure units, mega Pascal or bar. The MEP is defined as the work per cycle, W_c, per unit displaced volume, V_d.

$$MEP = \frac{W_c}{V_d} \tag{4.1.2}$$

The MEP concept can be generalized to include all kinds of energy variables for a combustion engine.

<u>FuelMEP – Normalized fuel consumption</u> (Pa), represents the fuel heat of the injected amount of fuel.

$$FuelMEP = \sum_{All\_fuel} \frac{\dot{m}_{Fuel} \cdot Q_{LHV,\,Fuel}}{n/120 \cdot V_d} \tag{4.1.3}$$

$Q_{LHV,\,Fuel}$ (J/kg) = refers to the lower heating value of the fuel

<u>QMEP</u>, is a measure of the amount of energy in the fuel that is converted to heat during combustion.

$$QMEP = \frac{Q_{HR}}{V_d} \tag{4.1.4}$$

Q_{HR} (J) = The accumulated heat release per cycle

<u>IMEP – Indicated Mean Effective Pressure</u>. Two different IMEP can be defined, gross and net IMEP. The gross IMEP is based on the work on the piston during the compression and the expansion stroke only and the net IMEP is based on the work on the piston during the entire cycle.

$$IMEPg = \frac{1}{V_d} \int_{Closed\_Cycle} p \cdot dV \tag{4.1.5}$$

$$IMEPn = \frac{1}{V_d} \int_{Complete\_Cycle} p \cdot dV \tag{4.1.6}$$

p (Pa) = cylinder pressure

dV (m^3) = change of the cylinder volume

The difference between net and gross IMEP represents normalized pumping losses (PMEP).

<u>BMEP – Break Mean Effective Pressure</u>, is the engine work per cycle although it is also defined as the normalized torque.

$$BMEP = \frac{4\pi T}{V_d} \tag{4.1.7}$$

T (Nm) = Brake torque

τ – Combustion duration, represents the crank angle between when 10% and 90% of the combustion has been completed.

$$\tau = CA90 - CA10 \tag{4.1.8}$$

η_{net} – Net indicated efficiency. This efficiency is obtained from the entire cycle and it includes the pumping work.

$$\eta_{i,n} = \frac{IMEP_n}{FuelMEP} \tag{4.1.9}$$

η_{gc} – Gas exchange efficiency, is the ratio between the indicated work during the complete cycle and the closed part of the cycle.

$$\eta_{ge} = \frac{IMEP_n}{IMEP_g} \tag{4.1.10}$$

4.2 Analysis of cylinder pressure data

Information about the combustion event can be obtained from the measured in-cylinder pressure by performing a heat release calculation. The calculations used for this master thesis assume that temperature and gas composition is equal in the whole combustion chamber.

The energy, or heat, released from combustion, Q, can be expressed as a function of the internal energy, U, the work performed, W, the heat transferred to the cylinder walls, Q_{HT}, and the mass loss to crevices, Q_C.

$$\partial Q = \partial U + \partial W + \partial Q_{HT} + \partial Q_C \tag{4.2.1}$$

Differentiating the expression for the internal energy, $U = m \cdot C_v \cdot T$ leads to the following equation

$$\partial U = m \cdot C_v \cdot \partial T + C_v \cdot T \cdot \partial m \tag{4.2.2}$$

Neglecting crevice losses, $\partial m = 0$ and expressing the temperature using the equation of state $p \cdot V = m \cdot R \cdot T$ in its differentiated form, considering m and R to be constant, leads to the following expression

$$\partial U = \frac{C_v}{R} \cdot \left(p \cdot \partial V + V \cdot \partial p \right) \tag{4.2.3}$$

The work performed on the piston can be written as $\partial W = p \cdot \partial V$, thus the expression for the released energy becomes:

$$\partial Q = \frac{C_v}{R} \cdot \left(V \cdot \partial p + p \cdot \partial V \right) + p \cdot \partial V + \partial Q_{HT} \tag{4.2.4}$$

If the gas is considered to be an ideal gas the following equations can be used

$$R = C_p - C_v \tag{4.2.5}$$

$$\frac{C_p}{C_v} = \gamma \tag{4.2.6}$$

Rewriting this to suitable forms, give

$$\frac{C_v}{R} = \frac{\gamma}{\gamma - 1} - 1 = \frac{1}{\gamma - 1} \tag{4.2.7}$$

The end result for the change per crank angle degree, θ, is

$$\frac{\partial Q}{\partial \theta} = \frac{\gamma}{\gamma - 1} \cdot p \cdot \frac{\partial V}{\partial \theta} + \frac{1}{\gamma - 1} \cdot V \cdot \frac{\partial p}{\partial \theta} + \frac{\partial Q_{HT}}{\partial \theta} \tag{4.2.8}$$

From the last equation it is seen that the pressure and volume as well as their derivatives are needed. Volume and its derivative are easily calculated providing the engine geometry is known. The pressure is measured with a pressure transducer and should always be logged whenever doing engine measurements of any kind. The ratio of specific heats, γ, can be found in tables and for the calculations done for the papers leading to this thesis it was assumed that γ has a linear dependence on temperature.

The heat transfer to the cylinder walls has to be estimated and one way of doing this is to assume a homogeneous temperature in the combustion chamber and a constant wall temperature, this results in the following equation

$$\frac{\partial Q_{HT}}{\partial t} = h \cdot A_{wall} \cdot \left(T_{gas} - T_{wall}\right) \tag{4.2.9}$$

5 EXPERIMENTAL SETUP

This chapter contains all the information about the engine that we have used for the experiments, including its setup and its outlining parts, the equipment and the measurement technique.

5.1 The test engine

The test engine is a modified six cylinder Scania DSC12 (12-liter) turbo charged diesel engine. All engine specifications are shown in the Table 5.1. It is mainly used in trucks applications.

Table 5.1: Specifications of the Scania DSC12 test engine

Engine type	Four-valve diesel engine converted to HCCI
Displacement volume	11705 cm^3
Valves per cylinder	4
Number of cylinders	6 in line
Bore	127 mm
Stroke	154 mm
Connecting Rod	255 mm
Exhaust valve open	82 ° BBDC
Exhaust valve close	38 ° ATDC
Inlet valve open	39 ° BTDC
Inlet valve close	63 ° ABDC
Compression ratio	18:1
Fuel	Natural gas / n-heptane

The engine works with a low-pressure sequential port fuel injection system instead of the original diesel injection system that works with electronic unit injectors. The engine has four valve cylinder heads with two inlet ports per cylinder. The injection system can thereby supply two fuels to each cylinder, one in each port.
In this way we can control the combustion timing with the amount of each fuel supplied to each cylinder from a controlling computer.

Since the two fuels are injected in different ports, stratification between the two fuels could be expected. In addition, the two fuels are in different phases, one of them is gaseous, natural gas, and the other is liquid, n-heptane.

The injectors used for liquid fuels operate by the same principle as a commercial port fuel injection system in a passenger car. And the basic working principle of the system for natural gas is quite the same as for the liquid fuels.

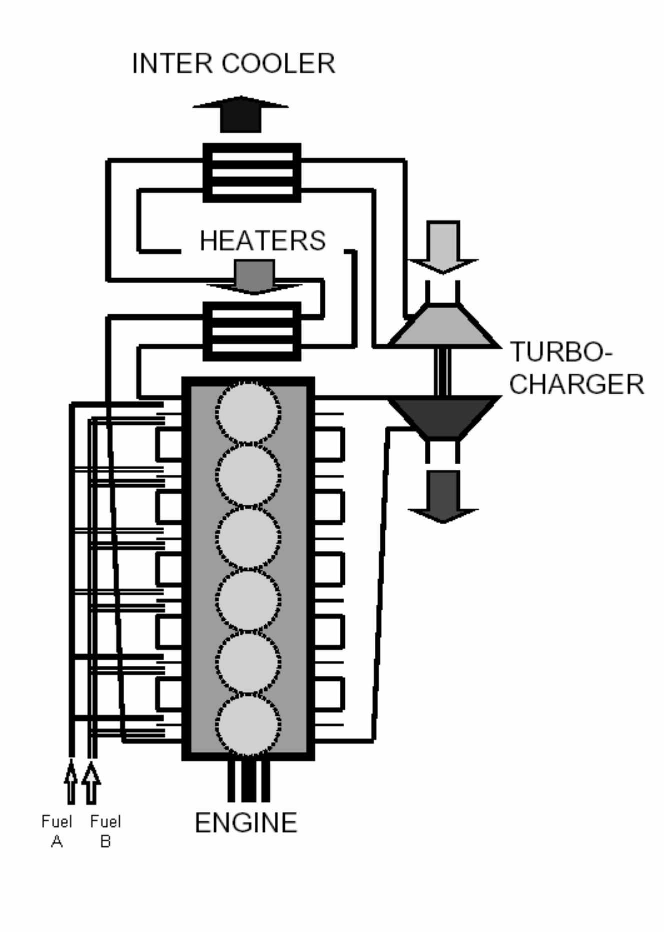


Figure 5.1 Engine system

Each cylinder is equipped with a cylinder pressure sensor to allow monitoring of the combustion. The pressure data are also used online for combustion control and computation of indicated parameters.

The inlet manifold has been extended to supply space for the injectors. In this way the injectors are placed just outside the original inlet ports of the cylinder heads.

The original Scania DSC12 engine worked with a turbo-charger dimensioned for the diesel cycle in a truck application. The turbine in this turbo charger was unsuitable for an HCCI application and it has been replaced because of the low exhaust gas temperature in order to generate boost pressure. The new turbo charger has a turbine with a variable inlet area so that the HCCI engine is able to run in a more suitable way.

The intake system is formed for two different subsystems, a city-water cooled CAC (charge air cooler) and electrical heaters. The CAC is turned off at lower loads to allow higher inlet temperature and improved combustion efficiency. Heating is only used at low loads to keep emissions of HC and CO down, i.e. to keep combustion efficiency up. High temperature and pressure are required in order to initiate HCCI combustion. Heating power is continuosly adjustable between 0 and maximum power.

Apart from these changes the engine was in its original configuration, with both pistons and cylinder heads unchanged. Figure 5.1 shows a general engine system outline.

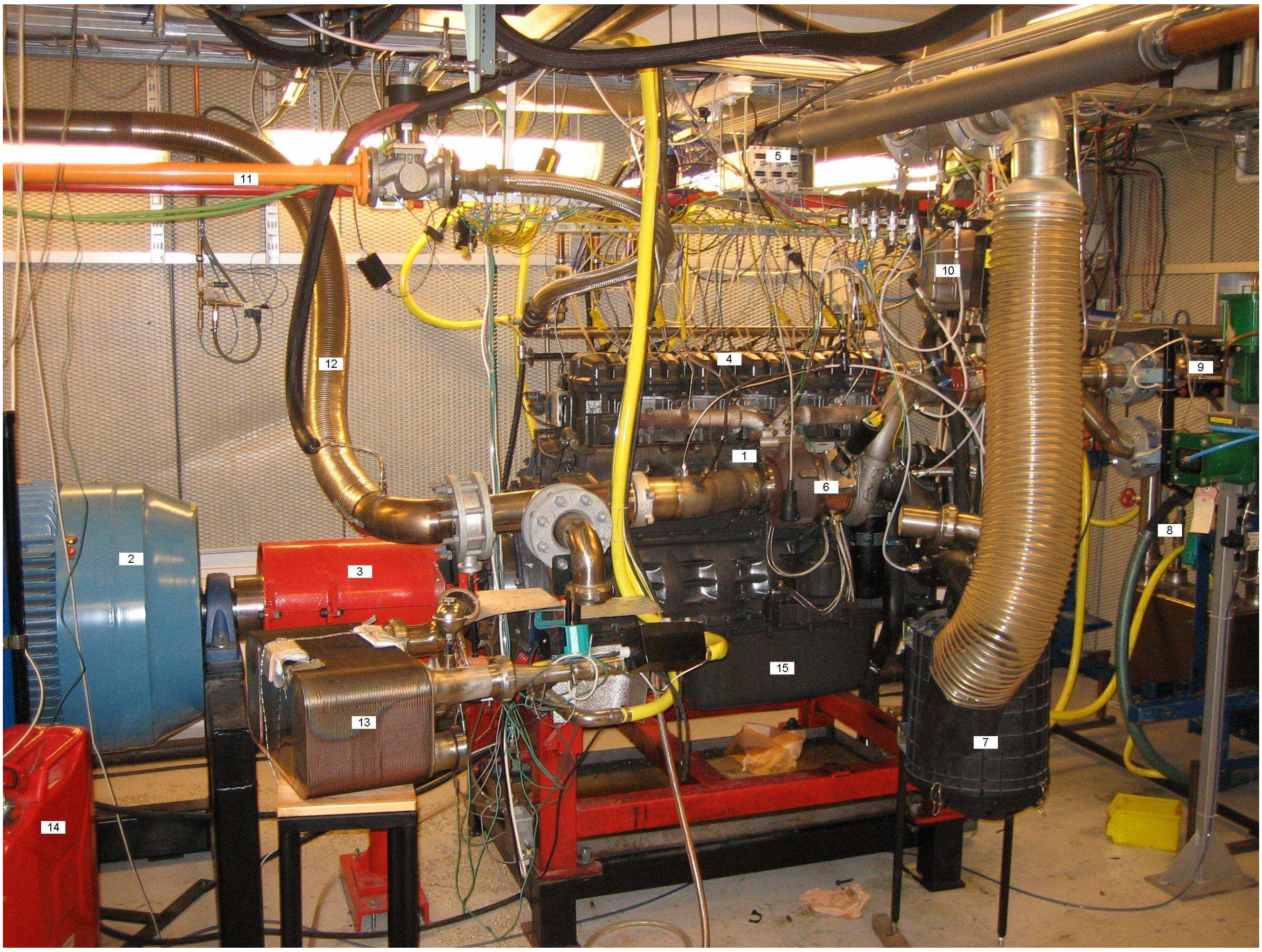


Figure 5.2 The engine test rig

1. Engine
2. Dynamometer
3. Drive shaft
4. Intake manifold (behind)
5. Charge amplifiers
6. Turbo charger
7. Air filter
8. Intercooler
9. Heaters
10. Cooling
11. Natural gas supply pipe
12. Exhaust pipe
13. EGR cooler
14. n – heptane tank
15. Sump

5.2 The test rig

5.2.1 – Dynamometer

The dynamometer is a main part of the installation in the test rig. It consists of an *ABB* AC motor with an accompanying control interface, which manages the motor speed and monitors motor currents and other parameters. When speed is adjusted, the dynamometer changes its torque dependent upon the engine's output torque. It either motors or brakes the engine keeping the desired speed.

5.2.2 - Data logging

The most important piece of instrumentation is the water-cooled piezoelectric cylinder pressure sensor, Kistler 7061B. One sensor per cylinder is used. The piezoelectric sensor is linear over a wide range and provides very rapid response. The sensor is connected to a charge amplifier that converts the charge signal from the sensor to a voltage output, which can be read by an A/D converter.

Each exhaust port has a thermocouple, measuring the exhaust stagnation temperature of each cylinder individually. Thermocouples are also used for monitoring inlet air temperature, before and after the turbo compressor, after the inter-cooler and after the heaters. Strain gauge absolute pressure sensors are used for measuring inlet air pressure and exhaust pressure before and after the turbine.

Most of the temperatures and pressures outside of the cylinder are sampled by a low-speed data acquisition system, whereas the in-cylinder pressure, the intake pressure and the intake temperatures are sampled by a high-speed data acquisition system, used for closed- loop control.

Cylinder pressures, inlet air temperature before and after the heaters and the inlet pressure are all monitored by the controlling computer. Cylinder pressures are sampled at a rate of 1800 samples per cycle for each cylinder. Inlet conditions are sampled once per cycle. These variables are sampled by a 16 bit multiplexed A/D converter on a multifunction card connected to the PCI bus of the controlling computer. The same card is used to control the heaters by two digital channels and a 12 bit analog output.

5.2.3 - Exhaust gas measurements

An emission measurement system sampling exhaust after the turbine is used to measure dry concentrations of O_2, CO and CO_2, and wet concentrations of HC, NO_x and NO. This is done with a system from BOO Instruments Inc.

The wet concentration of HC is measured by a flame ionization Detector (FID), and is calibrated using CH_4. The HC analyser has several different measuring ranges from 0 to 100000 PPM.

The NO_x analyser measures the wet concentration of NO, NO_2 and NOx. This is achieved with a chemiluminescence detector (CLD). The analyser has four ranges from 0 to 100000 PPM.

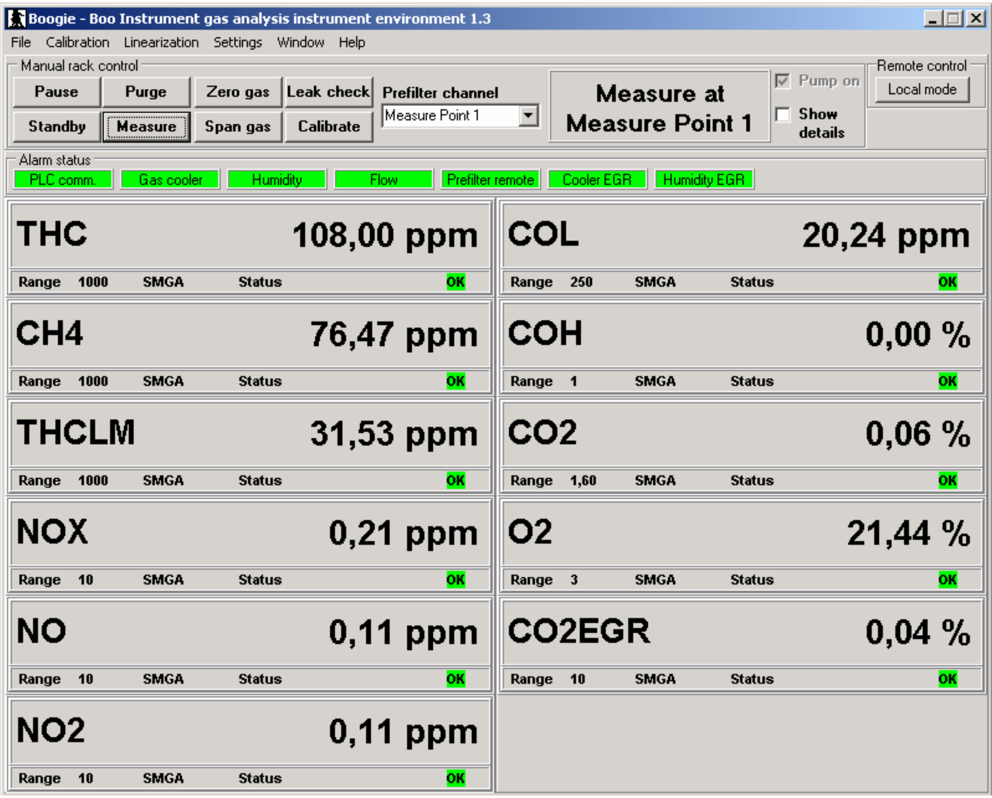


Figure 5.3 The emission analysis equipments graphic interface

The CO_2, O_2 analyser measures the dry concentration of CO_2 with a non-dispersive infra red (NDIR) detector in two ranges from 0 to 1,6% or from 0 to 16%

and the dry concentration of O_2 with the paramagnetic detector in ranges from 0 to 3% or from 0 to 25%.

The CO analyser measures the dry concentration of CO in two different levels, CO_{high} ranging from 0 to 0,1% and CO_{low} from 0 to 250 PPM or from 0 to 2500 PPM.

All these instruments are connected to a computer program called Boogie that shows, in a graphic interface, the emissions concentration. This interface is shown in Figure 5.3.

The air/fuel ratio is also measured directly in the exhaust manifold pipe with an LA3 Etas lambda meter. The LA3 uses a Bosch broadband lambda probe for measuring.

5.3 Control system

The engine is controlled from a program based on Delphi(object oriented Pascal) code that has been developed within the division. Figure 5.4 shows its interface.

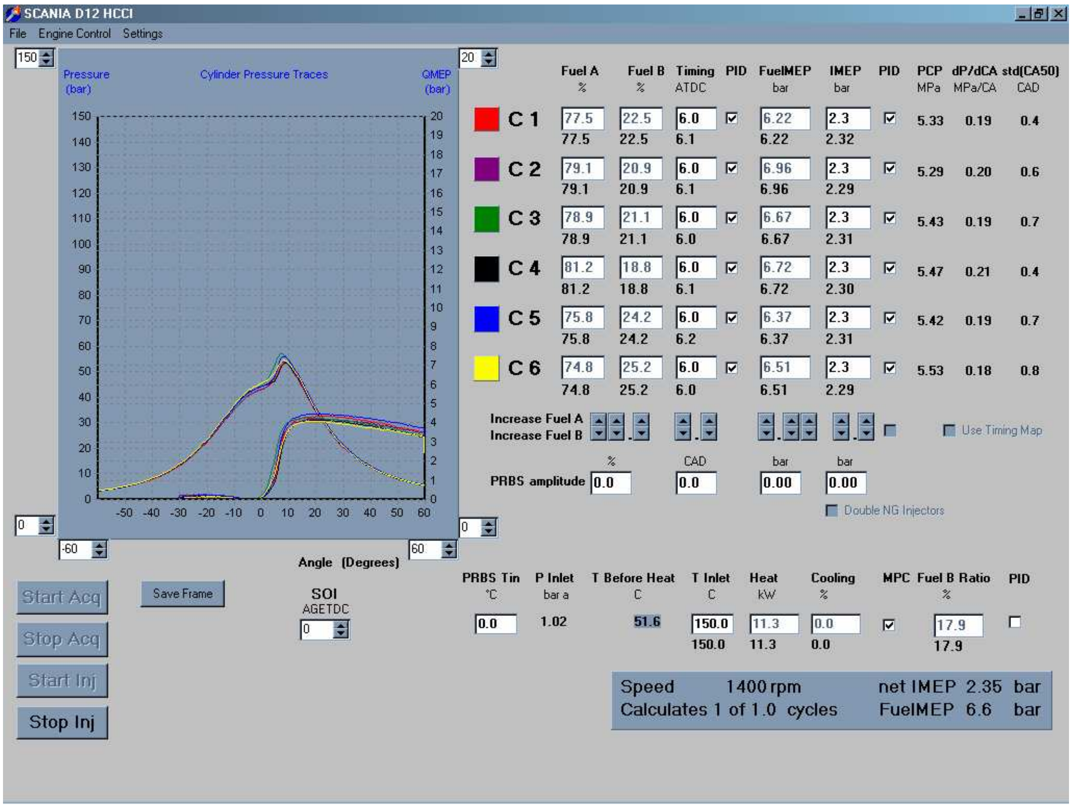


Figure 5.4 The control programs graphic interface

The program is able to control many different variables but for this thesis we only use the control of the amount of the different fuels to get the homogeneous suitable mixture in each moment for proper ignition timing, IMEP and inlet temperature.

Another useful data that this program also displays is the peak cylinder pressure (PCP), pressure derivative (dP/dCAD) and the standard deviation for crank angle of 50% burned fuel (std (CA50)) which is also a measurement of the combustion stability. This is done individually for each cylinder. The program also monitors the inlet air pressure.

This program also visualises the in-cylinder pressure trace for the cylinders, which is a feedback for the user about when the autoignition occurs and how the fuel burns as well as the acumulated heat released.

The control signals for injection are sent to a trig box manufactured at the division. It sends pulses to the injectors, synchronized with the engine through the crankshaft encoder.

The injection timing is entered individually for each cylinder in CAD ATDC. The injection duration controls the amount of fuel for each cylinder and it is calculated from the FuelMEP value.

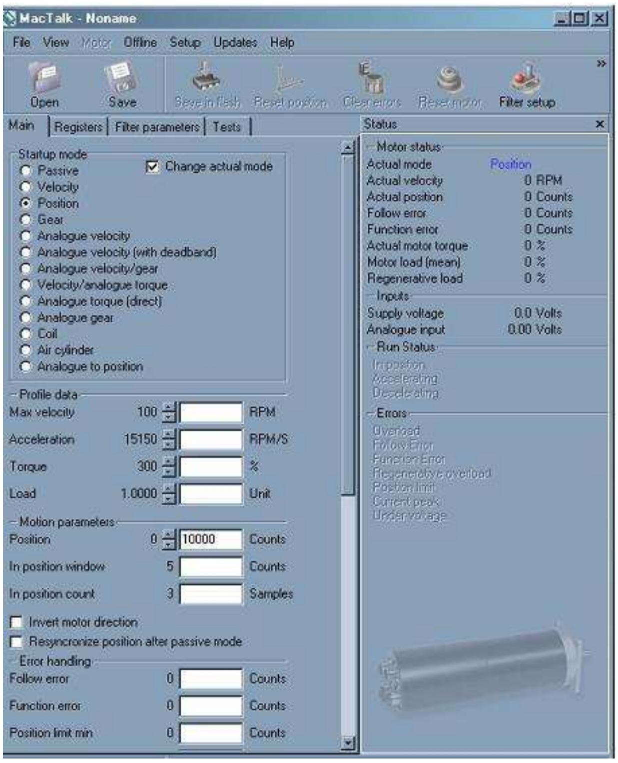


Figure 5.5 The turbo control programs graphic interface

In some parts of the load/speed range it is necessary to boost the inlet pressure using the turbo charger. The turbo is controlled with a JVL program, which mainly manages the servo motor position that changes the turbines blade angles. The programs interface is shown in Figure 5.5. The servo motor position represents the turbine area with which the turbo charger is working in these conditions.

5.3.1 - Measurement procedure

In this thesis the following main engine parameters have been varied during the measurement campaign to try to get the mapping.

- Speed
- Load
- Timing

The measurement procedure was the same for all sweeps.

First of all, the engine was heated up motoring until almost constant engine parameters were achieved. Afterwards the engine was started up and the ratio between the two fuels was adjusted to be able to switch on the PID controllers for timing and IMEP. Later on, the engine was run for each speed at all different values of the timing (-4 to 16 CAD) and of the IMEP (2 to 5 bars). The load (IMEP) is limited due to the pressure rise. The pressure rise rate permitted in this work is 15 bars/CAD. Figure 5.6 shows the derivative pressure (dP/dCAD) and the peak cylinder pressure (PCP). Earlier points were not always able to be run for all the speeds due to the pressure rise.

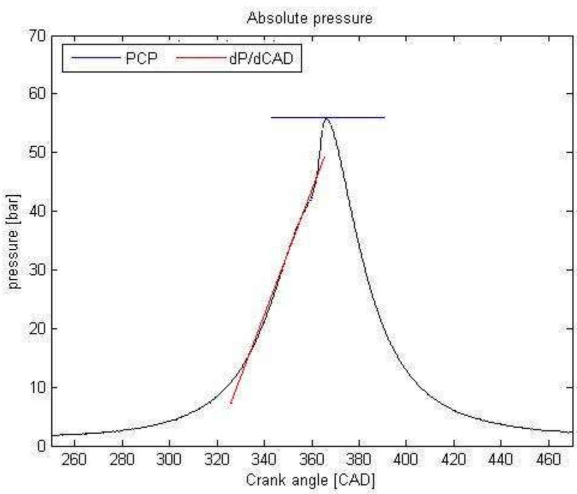


Figure 5.6 Pressure trace in a cycle

At each step, the suitable ratio between the different fuels is adjusted controlling the inlet temperature. To get an adequate controller range the amount of n-heptane could not be less than 10% of the mixture because otherwise the PID controllers were not able to control the combustion. To reach the desired inlet temperature, the MPC controller had to be connected to switch on/off the heaters or the cooling.

At each step engine data was logged.

After this first data campaign, the mappings were analyzed. The points with NO_x rates over 0.2 g/Kwh (NO_x limit considered undesirable in this work) have been retaked charging the engine with the turbo. The turbo charging is set with the servomotor position; 0 counts (100 % opened) and 110000 counts (0% opened).

With the help of the turbo, the engine power density increases. Then the engine is able to run to higher loads.

In the second campaign, the mappings was made bigger in the load range until 7 bars.

5.4 Data acquisition

Inlet pressure and inlet temperature can be saved from the control program for each measuring point pressure traces. To get adequate statistics for the pressure traces every save results in 100 consecutive pressure traces for each monitored cylinder. The result is one binary file for each cylinder and two separate files for inlet pressure and inlet temperature.

The engine is continuously monitored by a Logger program. Figure 5.6 shows its interface. This program was developed by the division and is based on Visual Basic and acquires data via Hewlett Packard logger.

Figure 5.6 The logger programs graphic interface

The data contains a wide range of different variables but in this thesis we used principally the emission values, combustion efficiency, BSFC, torque, BTE, the different temperatures, fuel flow, lambda, etc. For each measuring point approximately 40 readings are obtained during a two-minute period to get the adequate results.

5.5 Data analysis

All the data acquired are analyzed with different Matlab files programmed by myself. In these different files are plotted:

- All the emissions, the lambda, the maximum chamber temperature, the combustion efficiency, the brake specific fuel consumption, the brake thermal efficiency, the ratio between natural gas and n-heptane, the standard deviation of CA50, the coefficient of variation of the load, the pressure derivative and gas exchange efficiency regarding timing and load in each speed.

- The different loads regarding the amount of natural gas and timing and regarding the combustion duration, the brake specific fuel consumption, the net indicated efficiency and timing.
- All the emissions, the combustion efficiency, the brake specific fuel consumption, the timing, the ratio between natural gas and n-heptane, the standard deviation of CA50, the coefficient of variation of the load, the combustion duration and gas exchange efficiency the brake thermal efficiency regarding timing and load in each speed in the state of maximum net indicated efficiency.
- The absolute pressure, heat losses, heat (dQ/dCAD) and accumulative heat release regarding timing for each cylinder and for each measure. These plots are also displayed with mean values.

6 RESULTS

6.1 Emissions

In this thesis work, the emissions will be presented as specific emissions in grams per kilowatt-hour to be able to compare the different emissions with the same units.

All comparisons are made with emissions measured before the catalyst.

6.1.1 - NO_x

Almost zero nitrogen oxides (NO_x) are one of the best advantages of the HCCI engine. These results are reached working with a homogeneous process of a highly diluted premixed mixture, which means that the combustion temperature becomes low in the combustion chamber. And due to combustion process it occurs very fast.

The NO_x formation rate has a kind of exponentially dependence on the temperature. The amount of NO_x and the maximum chamber temperature versus the load and the combustion timing are plotted in Figure 6.1 (a) and Figure 6.1 (b) running the engine at 1200 rpm. Both variables have the same trend and the almost exponential behaviour of the NO_x regarding the temperature is easily noticed comparing the two Figures.

Figure 6.1 (a) also shows the load dependence, how the NO_x increases as the loads and the timing dependence and how the NO_x increases as combustion timings decrease. For instance, when the engine is running at 1200 rpm and the combustion starts at 4 CAD, the NO_x amount and the maximum chamber temperature for 3, 4 and 5 bars are 0.033, 0.36 and 2.35 g/Kwh and 1552, 1682 and 1745 °K, respectively.

From these values, the conclusion are that the NO_x are primarily formed at combustion temperature exceeding 1600 ºK and at 1700 ºK approximately, the formation of NO_x increases rapidly with the increased temperature.

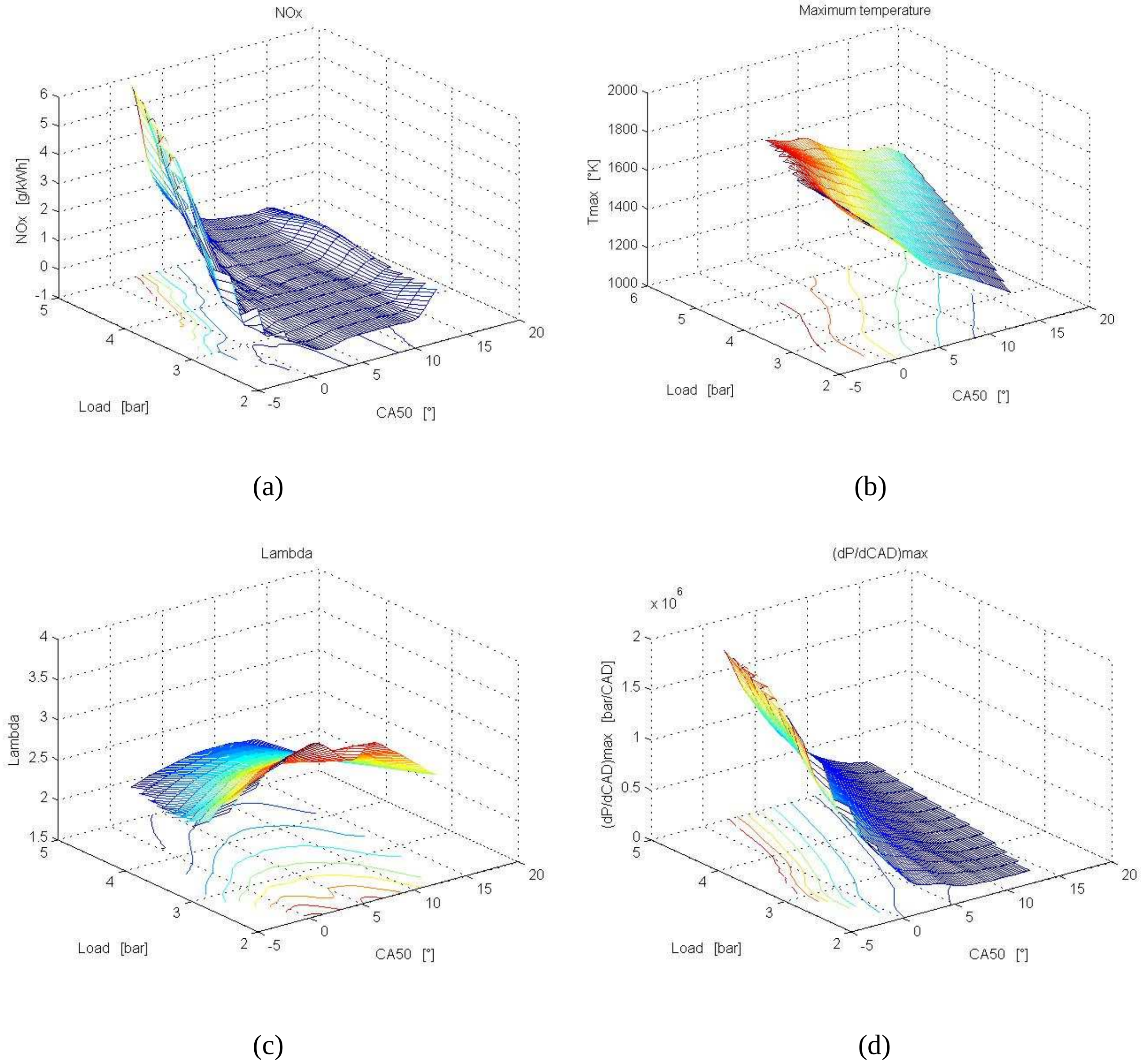


Figure 6.1 (a) NO_x emissions. (b) Maximum chamber temperature during combustion. (c) Relative air/fuel ratio (λ). (d) Maximum derivative pressure in chamber during combustion. The engine speed for these measures is 1200 rpm and it is working with minimum boost pressure.

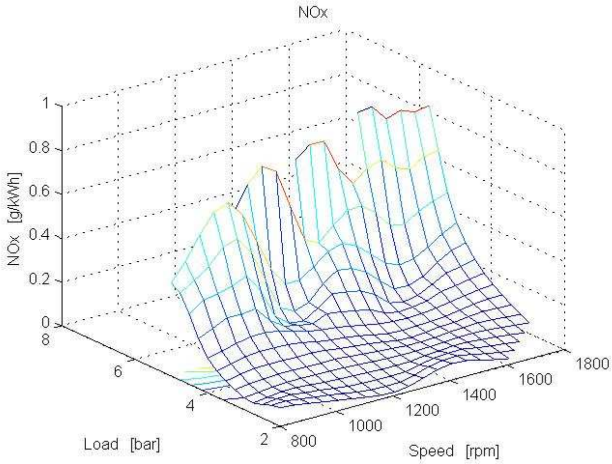


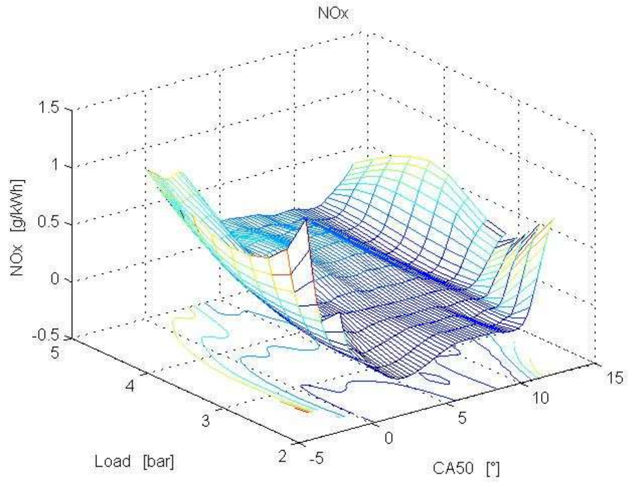
Figure 6.2 NO_x emissions in maximum net indicated efficiency condition.

Figure 6.2 shows again the load dependence but in this case as a function of the different speeds and loads at maximum net indicated efficiency condition instead of with combustion timings.

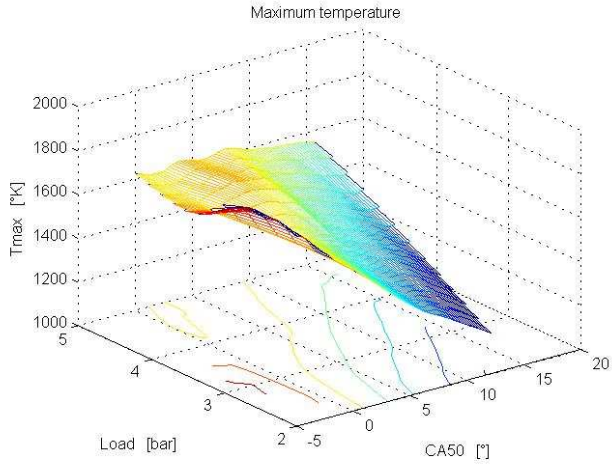
The relative air/fuel ratio, λ, also strongly influences the NO_x rate because of the combustion temperature, which also depends on λ. Figure 6.1(c) shows that this relation is about inversely to the temperature. With lower λ, the NO_x rates are higher. For the above instance, the different λ for the different loads are 2.98, 2.37 and 1.99 respectively. It can be supposed that with leaner mixtures than $\lambda \sim 3$ almost zero NO_x is produced and with below than 2.5, NO_x formation increases steeply as λ decreases.

When the engine is running with high combustion rate, which are caused by low λ, the engine starts to support an unacceptable steep pressure rise, which leads to knocking with severe pressure oscillations in the cylinder. This gives a very noisy engine operation and in the worst possible case it can lead to engine failure. For these reasons, the pressure rise is limited to 15 bars/CAD. Figure 6.1 (d) shows the almost direct proportional relation between the derivative pressure and the maximum temperature during the combustion. This means that with higher temperatures , as it was explained above, and higher pressures, the NO_x amount increases.

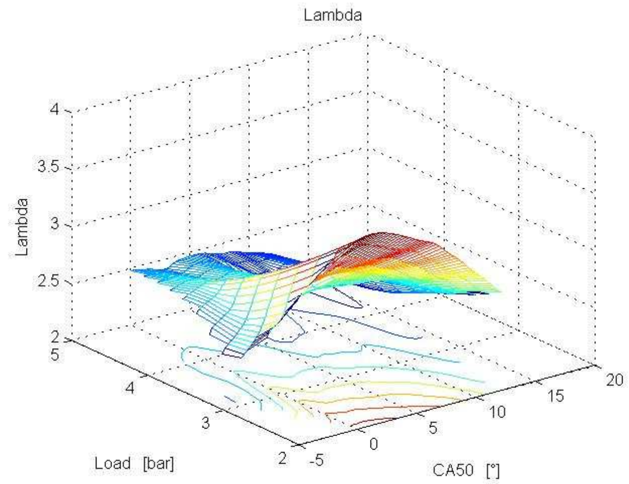
One-way to reduce the NO_x rate, is increasing the inlet air pressure or in other words, charging more air in the combustion chamber with the help of the turbo.This means that the relation air/fuel, λ, increases its value raising the pressure level and



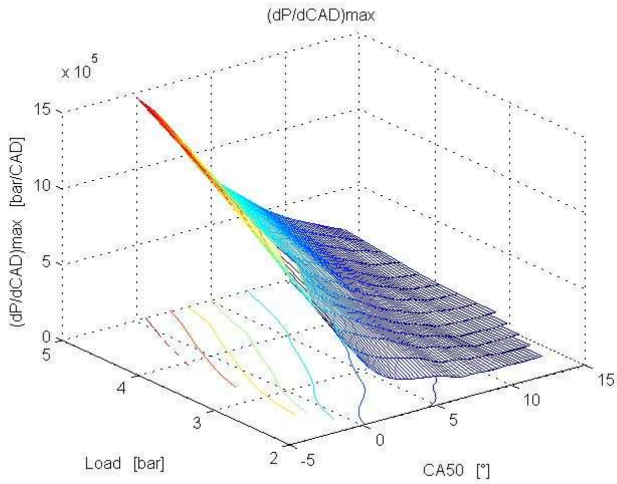
(a)



(b)



(c)



(d)

Figure 6.3 (a) NO_x emissions. (b) Maximum chamber temperature during combustion. (c) Relative air/fuel ratio (λ). (d) Maximum derivative pressure in chamber during combustion. The engine speed for these measures is 1200 rpm and it is working with the suitable proportion of turbo charging for each point.

then the temperature required to reach the autoignition conditions increases. However, at highly boosted conditions λ can be limited as in naturally aspirated operation by too high peak cylinder pressure.

Figure 6.3 contains the corresponding graphics than Figure 6.1 but with the engine helped by the turbo charger when it is necessary to decrease the NO_x emissions.

The lean limit is when the combustion starts to falter and too much HC and CO are generated. This point will be analyzed in paragraphs below.

The turbo charger has only been used in the conditions when it was really needed, in other words, when the NO_x amount is really high and when with turbo charging the NO_x rates can be improved and decreased. This controlled use of the turbo is due to the pumping losses introduced by the turbo. The turbo is used to try to decrease the NO_x until one suitable rate for our goals as 0.2 g/Kwh but no more, because if the turbo is working always in its maximum working condition the losses will be higher than if it only works obtaining the needed inlet pressure to get the desirable rate. Sometimes even using the maximum turbo charging it is not possible to reach this suitable rate of NO_x because the chamber pressure and the temperature during the combustion are very high in these work conditions.

To continue the comparisons with the above instance, the NO_x amount and the inlet air pressure in the example conditions, working the engine with minimum boost pressure, are 0.033, 0.36 and 2.35 g/Kwh and 1.03, 1.04 and 1.07 bars, respectively. At 3 bars of load, the NO_x rate is really tiny (smaller than 0.2 g/Kwh) and the turbo charging is not necessary. For the others two loads, the rate is higher and the turbo is needed. At 4 bars, the turbo's servomotor is set for around 67000 counts (60 % opened) increasing the inlet pressure from 1.04 to 1.10 bars to get down the NO_x until 0.19 g/Kwh, under the marked rate. At 5 bars, the turbo in this case works nearly its maximum condition (almost closed) and it is set for around 77000 counts increasing the inlet pressure from 1.07 to 1.32 bars and the NO_x decreases again under the marked rate to 0.18 g/Kwh. Without the turbo help, the engine was not able to run in higher loads than 5 bars because of the exceeded pressure rise rate, but with it, at 6 load the NO_x amount starts to be elevated even using the turbo. In this load, the turbo is working in its maximum condition and provides 1.32 bars of inlet pressure. The NO_x rate in this case at 4 CAD is 0.7 g/Kwh. For higher loads the NO_x increases steeply because of the chamber conditions are very hard.

Figure 6.5 and Figure 6.6 show all NO_x emissions for all the speed range when the engine is working without and with the turbo charging, respectively.

The instance used until now can be generalized more or less for all the speed range because of the NO_x formation rate is almost independent of the engine speed if the turbo is left out. This can be explained because the inlet temperature goes up when the engine speed increases in order to maintain fix combustion timing. This behaviour can be looked at Figure 6.4, running the engine in all speed range and with the combustion fixed at 8 CAD and 4 bars of load. But increasing engine speed also means shorter residence time. So, both effects, shorter time and higher temperature balance each other, resulting in almost the same NO_x. This independence is shown in the different graphics of Figure 6.5. To assess this Figure the scale has to be taken into consideration because at 800 and 1000 rpm the engine has been run until -4 CAD and the maximum NO_x in some point is nearly 10 g/Kwh. However, because of the pressure rise at 1200, 1400 and 1600 rpm the engine has only been run until -2 CAD, this is the reason that in these graphics the maximum NO_x is only around 5 g/Kwh. At 1800 rpm the engine has only been run until 0 CAD.

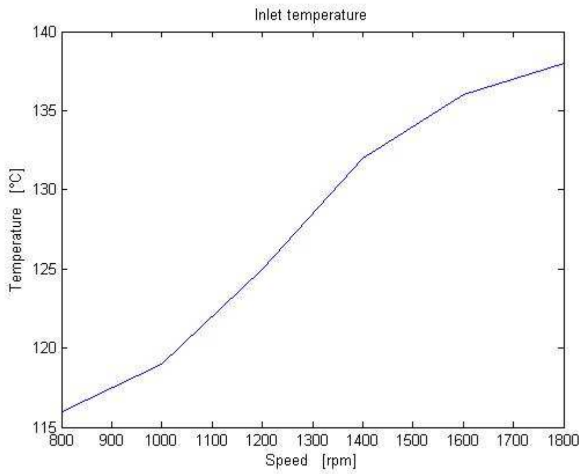


Figure 6.4 Inlet temperature for all speed range fixing the combustion timing at 8 CAD and the load at 4 bars.

The graphics shape is almost the same for all the speeds because of the commented independence. NO_x is reduced as the combustion timing is delaying, which decreases the combustion temperature. When the combustion starts to falter in the last timings is due to low chamber temperature during the process causing bad combustion and NO_x amount increases again as the graphics show. This growth is due to work with specific emissions, the NO_x grams decreases but the Kwh generated

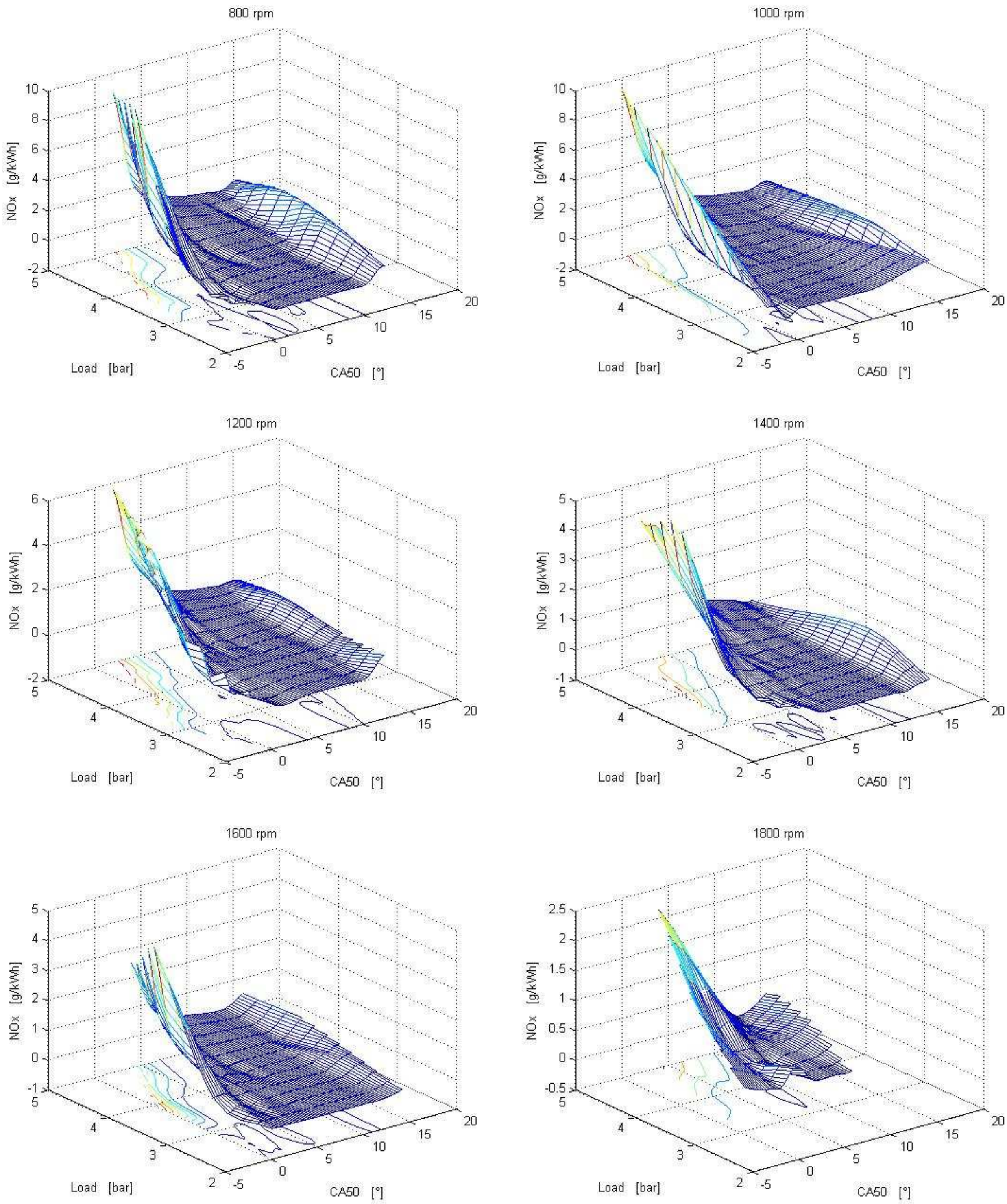


Figure 6.5 NO_x emissions graphics for all the speed range, working the engine with minimum boost pressure.

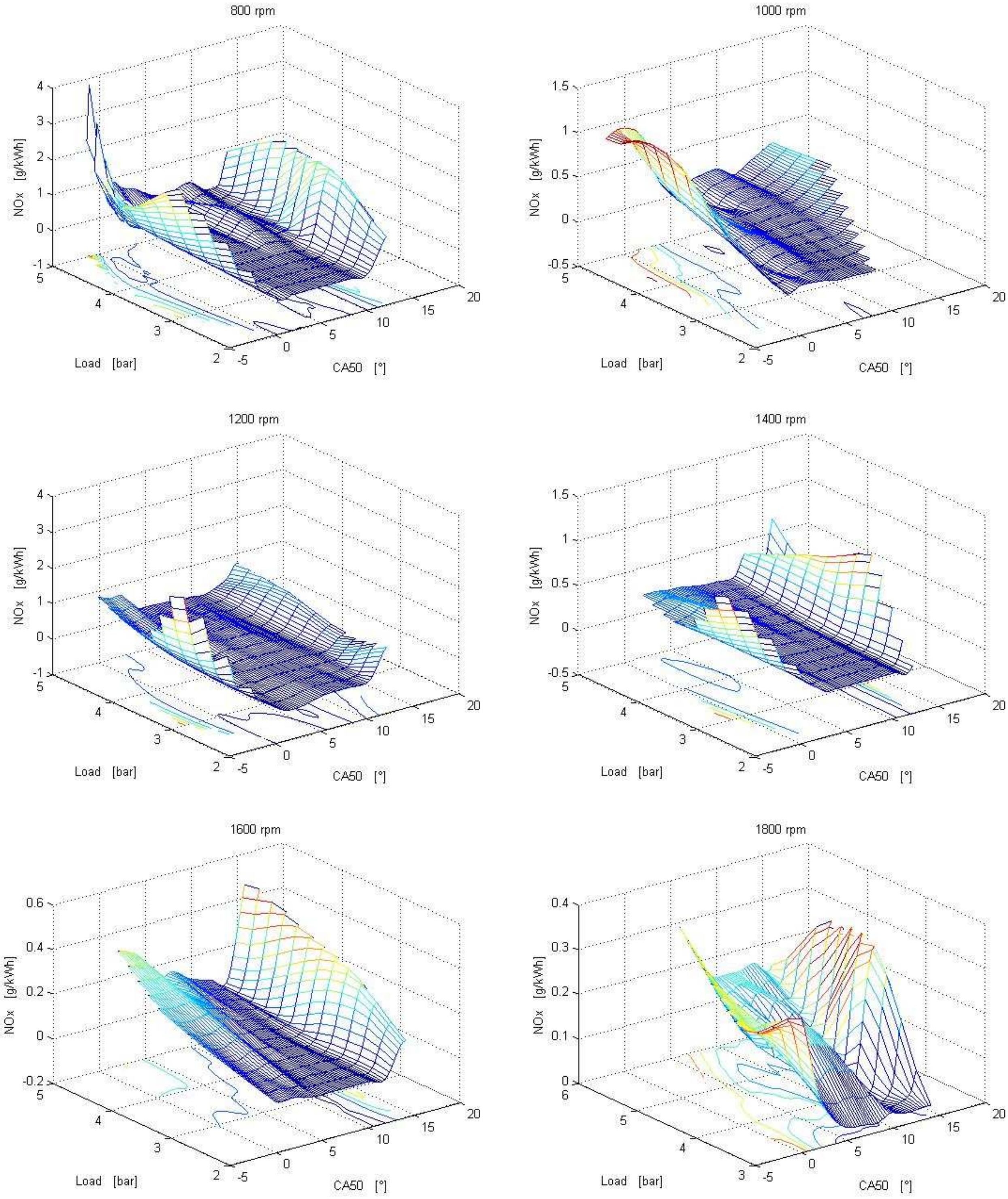


Figure 6.6 NO_x emissions graphics for all the speed range, working the engine with the suitable proportion of turbo charging for each point.

decreases even more increasing the total value. NO_x is also reduced, as the load is smaller because the temperature into the cylinders is lower.

One thing that is interesting to remark is when the engine is running at slow speeds, it is more difficult for the turbo charger to increase the inlet pressure because of the air flow that is circulating across the turbo is smaller than at faster speeds. Then the higher levels of NO_x appear in lower loads. This behaviour can be noticed in Figure 6.6. For instance, when the engine is running at 800 rpm there are really high rates at earlier points in 5 load while at 1800 rpm the NO_x amount starts to be high at 7 load.

The turbo charger comes up with the same problem than before but now it is difficult for it increases the inlet pressure in the earlier timings with low loads. The explanation for that is because the amount of work to make the turbo achieve the requested boost has to be extracted from the exhaust and as the combustion occurs earlier, then the gases have more time to transfer heat and therefore the exhaust gas temperature is lower. This can be noticed in 1200 and 1400 rpm graphics in Figure 6.6.

In this present work, the engine has been run with two fuels. The normal mixture is around 85% of natural gas and 15% n-heptane. The natural gas with high octane number requires higher compression ratio and/or higher inlet temperature to ignite. This leads to higher combustion temperature and the most important, more NO_x produced.

6.1.2 - HC

The HCCI engine is usually related as a really clean engine. But to reach this goal, the unburned hydrocarbons (HC) and CO have to be oxidized in an after treatment system because the rates of these emissions are really high and dangerous for the environment. The HC emissions can be between 2 or 3 times higher than in a SI engine.

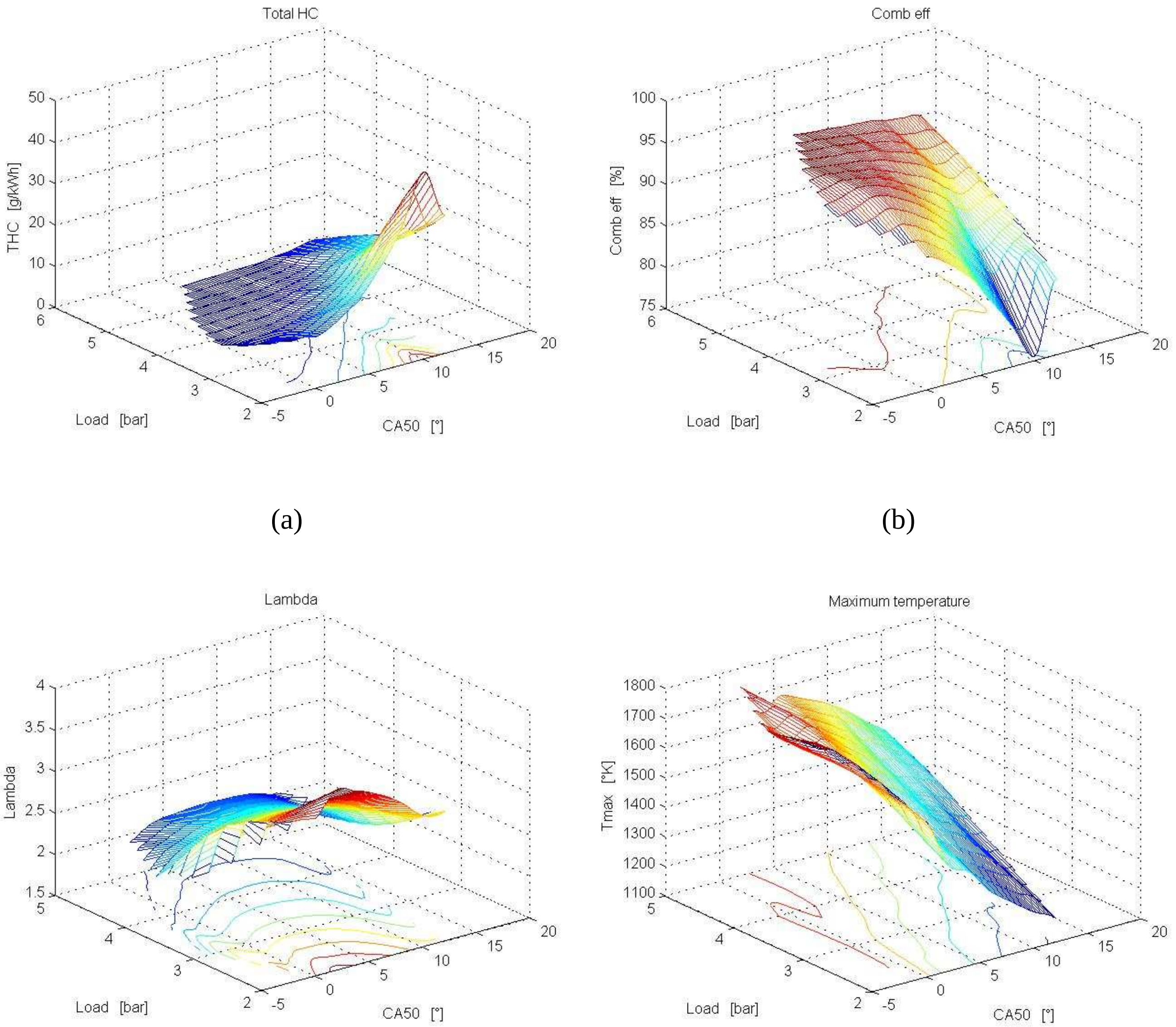
Due to the homogeneous combustion of a lean mixture, the combustion temperatures are low, and then the bulk temperature drops early in the expansion stroke resulting in a shorter time for oxidation. The exhaust emission of unburned hydrocarbons are from fuel that escapes combustion due to the low temperature during the cycle that fails to oxidize completely and not for the lack of oxygen, because the engine is working with premixed lean charge.

The main source for HC is crevices, primary the piston topland crevices (volume between the piston and the cylinder wall above the upper compression piston ring). Wall effects and bulk quenching proved to be less important.

Figure 6.7 (a) shows HC emissions when the engine is running at 800 rpm with minimum boost pressure. Comparing with the combustion temperature in Figure 6.7 (d), a kind of logarithm relation between them can be assessed. The lower the temperature is, the HC rates are higher. For instance, when the engine is running at that speed and the combustion starts at 12 CAD working with loads 2, 3, 4 and 5 bars, the HC rates and the chamber temperatures are 33.23, 21.38, 13.96 and 7.5 g/Kwh and 1180,1348,1451 and 1541 ºK, respectively.

As it was mentioned above, in the NO_x paragraphs, the relative air/fuel ratio, λ, is inversely related with the combustion temperature. This relation is shown in Figure 6.7 (c) and Figure 6.7 (d). So, as the mixture is leaner, the HC amount is higher. For the above instance, the λ for the different loads are 3.06, 2.51, 2.34 and 2.09 respectively, that verify the last premise.

Figure 6.7 (b) shows the combustion efficiency when the engine is running at 800 rpm. If the levels of HC are so high, this will affect the combustion efficiency, which in turn reduces the overall engine efficiency because of fuel escapes combustion or in other words, it is not producing heat, as it should. This means losses.



(a) (b)

(c) (d)

Figure 6.7 (a) HC emissions. (b) Combustion efficiency. (c) Relative air/fuel ratio (λ). (d) Maximum temperature in chamber during combustion. The engine speed for these measures is 800 rpm and it is working with minimum boost pressure.

In the used instance, the different combustion efficiencies are 84.1, 87.3, 90.4 and 94.8 %, respectively. These values show clearly, that it highly affected the combustion efficiency (more than 15%) in low loads because the fuel is not burn properly.

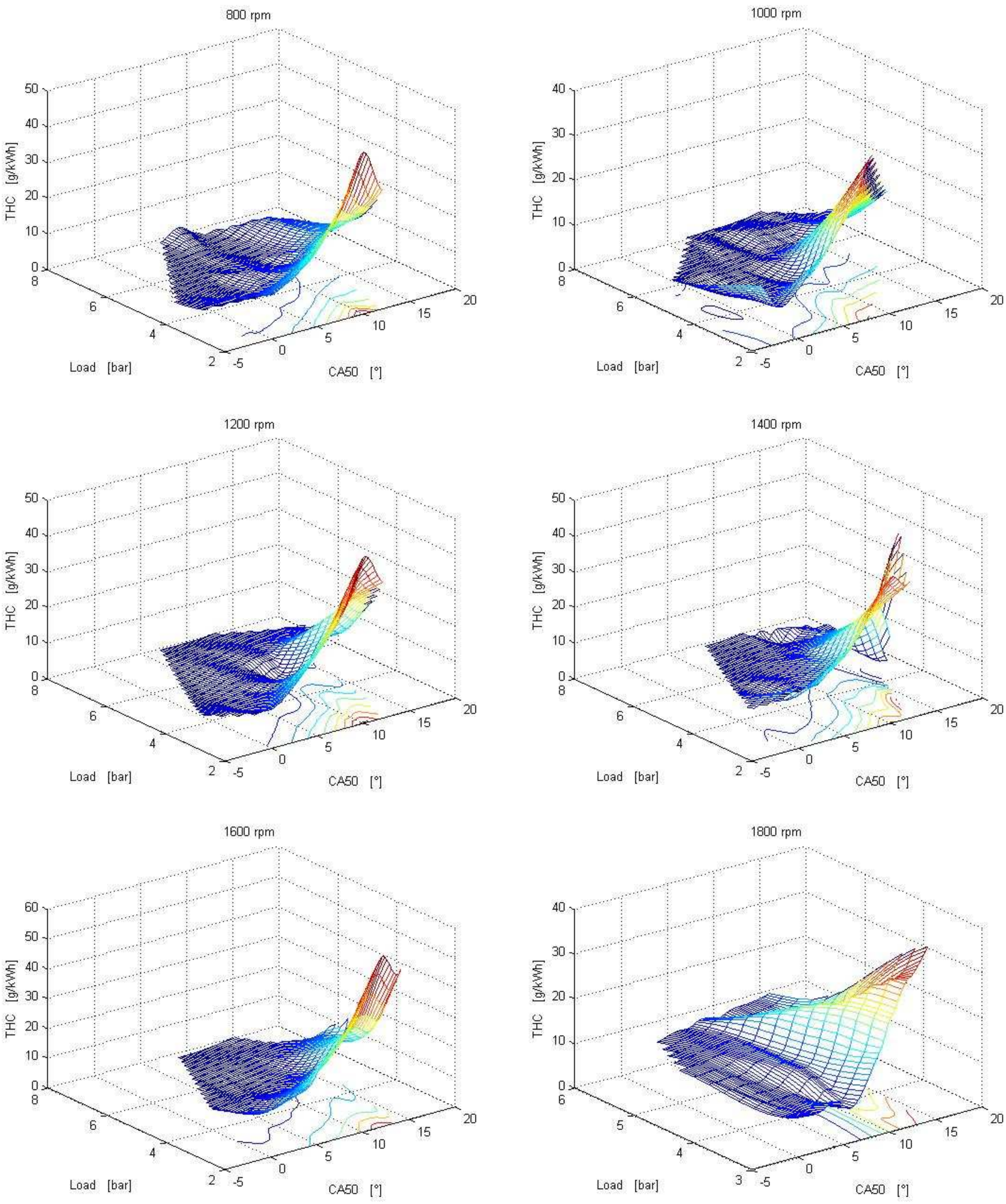


Figure 6.8 HC emissions graphics for all the speed range, working the engine with the suitable proportion of turbo charging for each point.

Coming back to the analysis of Figure 6.7 (a). At very low load, when the charge is strongly diluted, the chemical reactions are slow and they are not able to finish and even in the bulk HC can escape oxidation. At medium and high load, the oxidation is not complete but at least, this only happens near the wall (crevices). Some representatives values to explain this are for instance, when the engine is running at 1200 rpm and the combustion is occuring at 10 CAD, the HC increases from 7.2 to 46.37 g/Kwh changing the load from 5 to 2 bars.

Figure 6.7 (a) also shows that HC emissions increase when combustion is retarded. The explanation for this is the lower combustion efficciency due to the lower maximum temperature obtained in the combustion chamber in these points. For instance, the engine is running at 1200 rpm and 3 bars of load, the HC rate in -4 CAD is 7.75 g/Kwh and it increases until 29.5 g/Kwh in 12 CAD.

Figure 6.8 shows that the HC emissions graphics are quite similar for all the speeds. This almost independence can be explained as NO_x because both emissions are influenced by the same parameters. When the engine speed decreases, the inlet temperature goes down to maintain fix combustion timing and residence time reaction is longer. Both effects balance each other resulting in almost the same HC.

The use of the turbo charger in high loads does not really affect the HC emissions since even decreasing the chamber temperature when it is used; it is not enough to harm the complete fuel oxidation in these conditions.

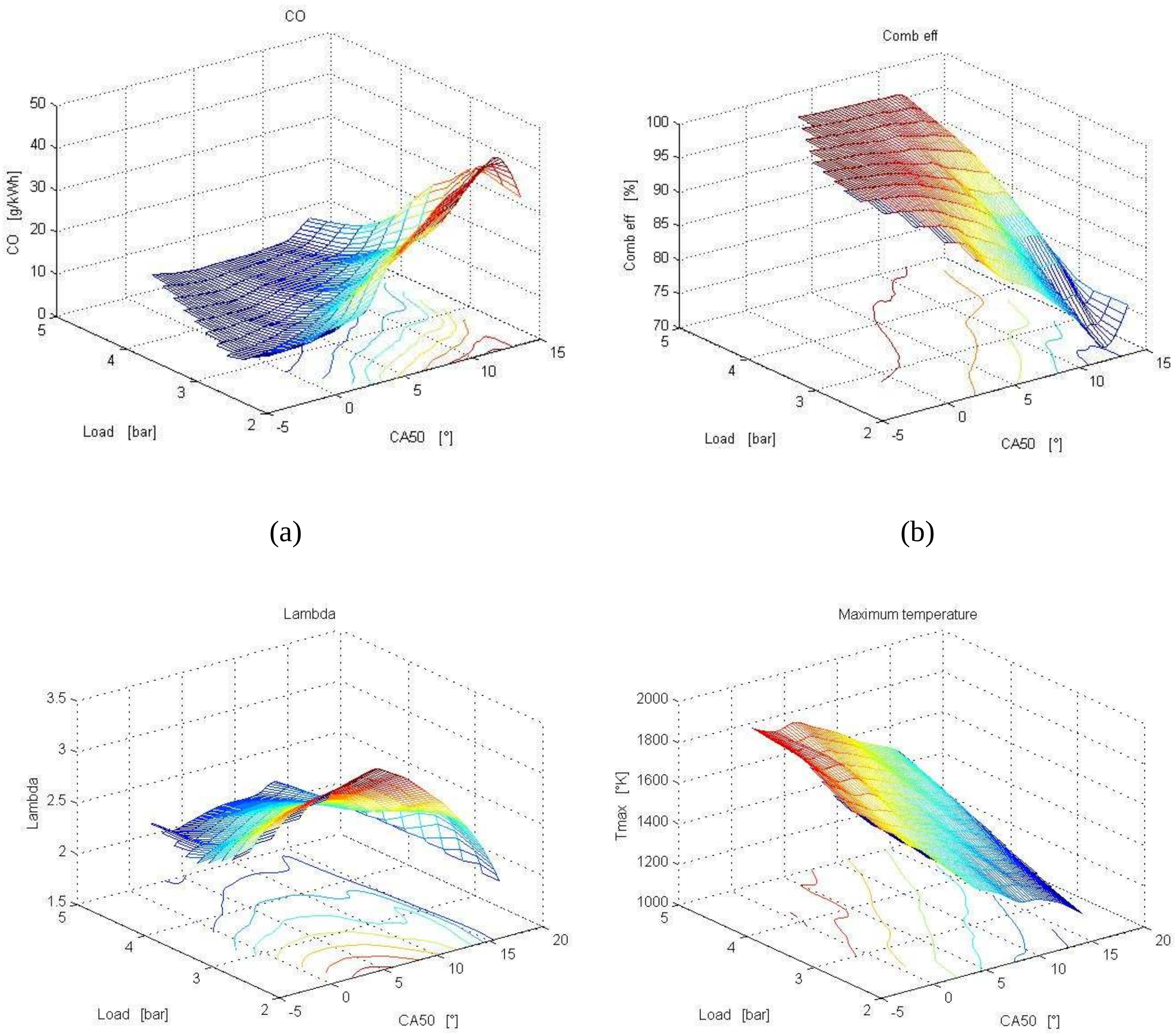
One point it is curious to remark that turns up in most of the graphics of Figure 6.8, it is that the highest emission rate is not produced in the latest timing in the lowest load. The highest rate is always at 10 or 12 CAD in 2 bars and after that the rate decreases a little bit. Every maps have been plotted using cubic interpolation and this behaviour is probably due to a Matlab operative error during the interpolation.

Finally, it is important to mention when the HC emissions are high, it is not a really great problem because using an oxidizing suitable catalyst the HC is reduced to an insignificant amount.

6.1.2 - CO

As it was commented above, CO emissions have to be treated to become a really clean engine.

Due to HCCI nature, the engine works with homogeneous combustion of a lean mixture and therefore low combustion temperature as it was explained before.



(a) (b)

(c) (d)

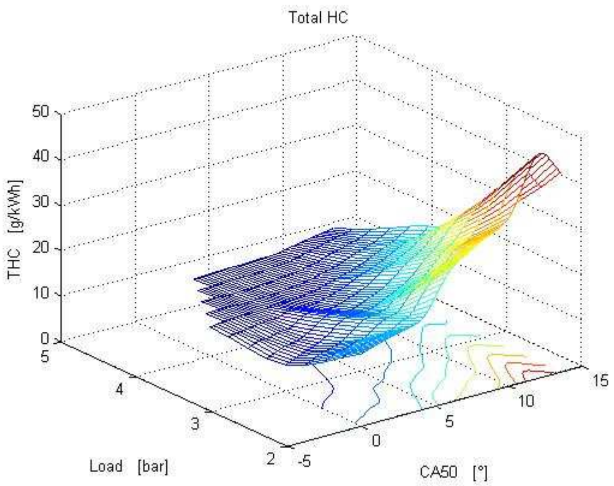
Figure 6.9 (a) CO emissions. (b) Combustion efficiency. (c) Relative air/fuel ratio (λ). (d) Maximum temperature in chamber during combustion. The engine speed for these measures is 1400 rpm and it is working with minimum boost pressure.

This results in an incomplete oxidation of the fuel, producing CO.

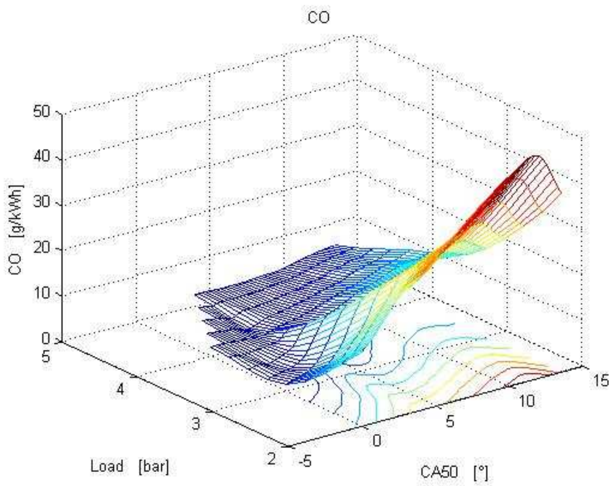
CO is very dependent on the combustion temperature. These variables have a kind of logarithm relation between them. This can be assessed in Figure 6.9 (a) and Figure 6.9 (d). For instance, when the engine is running at 1400 rpm and the combustion starts at 12 CAD working with loads 2, 3, 4 and 5 bars, the CO rates and the chamber temperatures are 45.39, 20.33, 8.05 and 3.4 g/Kwh and 1229, 1326, 1439 and 1586ºK, respectively.

It has also been explained in before paragraphs the relation between the relative air/fuel ratio, λ, and the temperature. CO is very dependent on λ and much CO is related with very lean mixtures (high λ). Figure 6.9 (a), Figure 6.9 (c) and Figure 6.9 (d) show the maps of these variables. Close to the rich limit and with early combustion timing, the CO generated is very low. In the opposite way, close to the lean limit so much CO is produced. For the above instance, the λ for the different loads are 2.91, 2.55, 2.32 and 2.1, respectively.

From the same point of view than HC, Figure 6.9 (b) shows how is affected the combustion efficiency due to HC and CO emissions. This also affected indirectly on brake torque efficiency.



(a)



(d)

Figure 6.10 (a) HC emissions. (b) CO emissions. The engine speed for these measures is 1600 rpm and it is working with minimum boost pressure.

All the comparisons that it was analyzed to HC, it has also been repeated for the CO. Both emissions depends on the same variables and comparing Figure 6.10 (a) and Figure 6.10 (b) it is easy to notice that they behave really similar.

Analyzing more in depth Figure 6.9 (a), it can be continue with the combustion timing and load dependence. Advancing the combustion timing and increasing the load will reduce the amount of CO produced as combustion temperature increases and there is more time available for in cylinder oxidation. For the used instance, the CO emission when the engine is running in 2 bars of load and the combustion is produced at 12 CAD is 45.39 g/Kwh. It decreases until 2.15 g/Kwh when the engine runs in 7 bars and 2 CAD.

Figure 6.11 shows the nearly speed independence for the CO emissions. The explanation for this is the same than for NO_x and HC. If the engine speed increases, higher inlet temperature is required in order to maintain fix the combustion phasing, but it also means less time for oxidation of CO. These effects are balanced each other resulting in almost the same CO.

It is important to know that all CO emissions can be reduced using an oxidizing catalyst until 99%, much more than HC because the temperature required for the complete oxidation are lower and closer to the exhaust gases temperatures.

Finally, only remark that this present work has been using natural gas as one of their fuels and less CO is generated with this fuel due to the demand of a higher inlet air temperature.

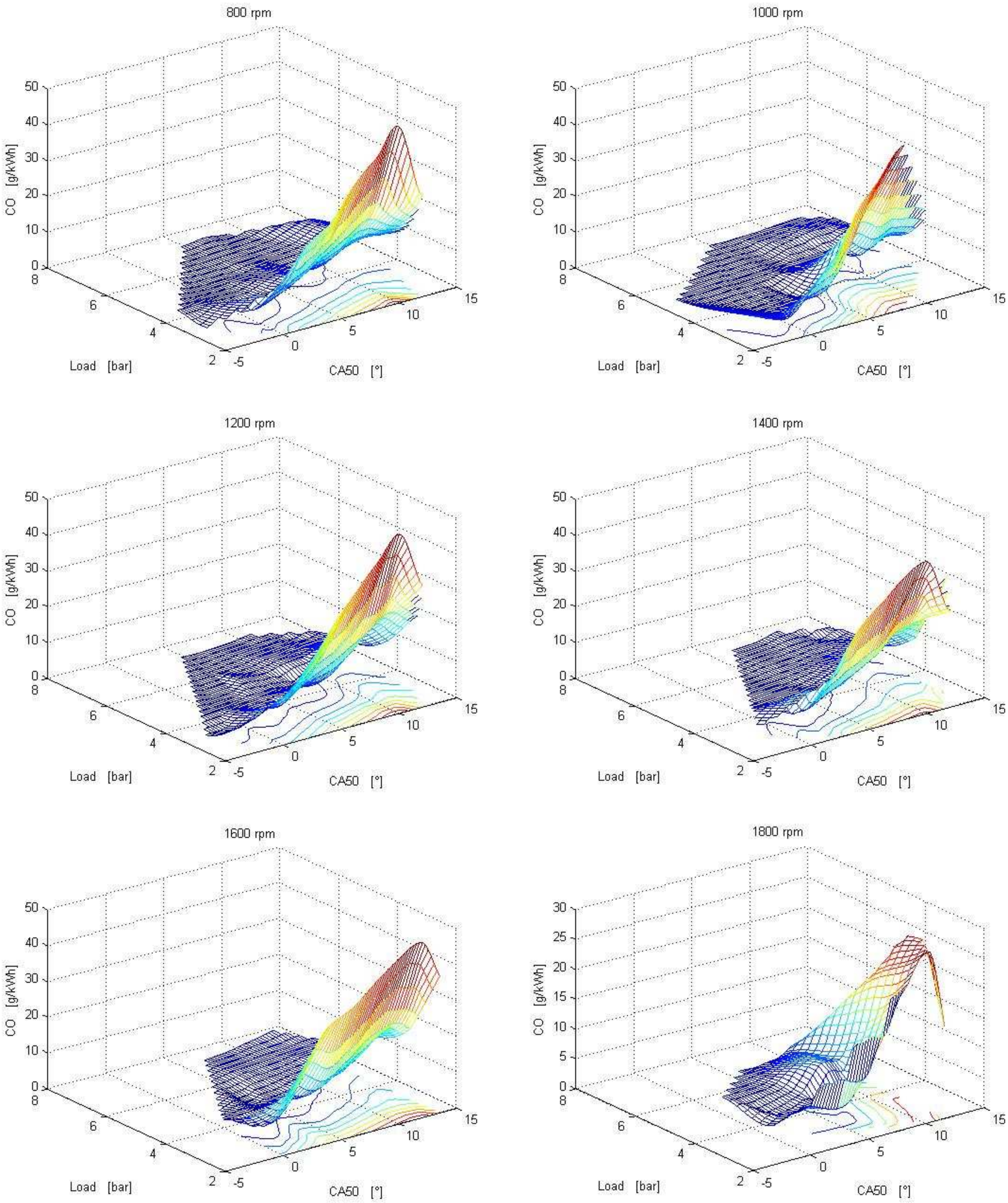


Figure 6.11 CO emissions graphics for all the speed range, working the engine with the suitable proportion of turbo charging for each point.

6.2 Engine efficiency

Higher efficiencies are expected with HCCI engine. This is due to the way of working with low combustion temperature, short combustion period and unthrottled operation.

The HCCI engine used in this master thesis works with high compression ratio (18:1) that means high efficiency from a thermodynamic point of view although it increases the heat losses since it also increases pressure and temperature.

6.2.1 – Combustion efficiency

Combustion efficiency is a calculation of how well the engine is burning the fuel. To reach 100% combustion efficiency is not realistically achievable in HCCI, but very good combustion efficiency should be around 95%.

HCCI operation is very sensitive to combustion timing, as it influences the combustion temperature. Late combustion timing means decreased temperature so

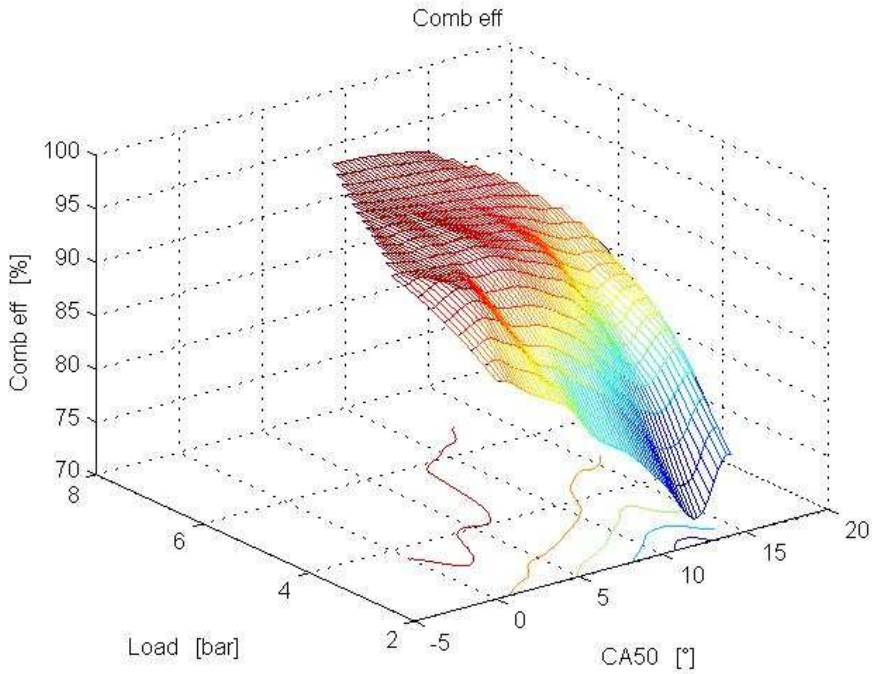


Figure 6.12 Combustion efficiency. The engine speed for these measures is 1600 rpm and it is working with the suitable proportion of turbo charging for each point.

that the combustion is worse and the combustion efficiency is reduced. This comparison can be assessed in Figure 6.12 and Figure 6.13 (a).

This also can be analyzed by another point of view, the inlet temperature. Although in the heart of the matter it arrives to the same point. The combustion efficiency improves as the inlet temperature is raised since the autoignition points are closer and it is easier to reach the total combustion. For instance, running the engine at 1600 rpm, at 4 bars of load and combustion is produced at 0 CAD, then the

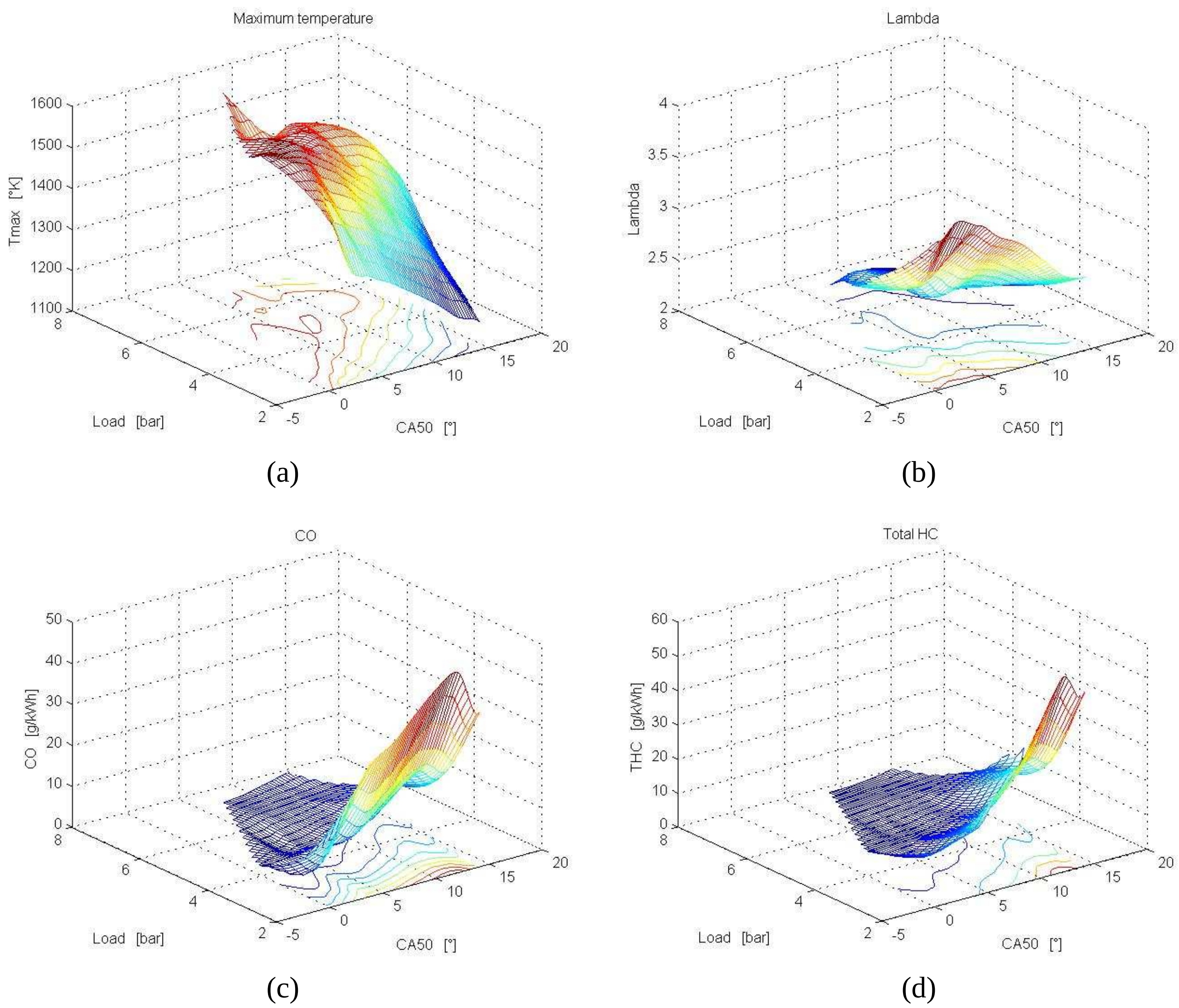


Figure 6.13 (a) Maximum temperature in chamber during combustion. (b) Relative air/fuel ratio (λ). (c) HC emissions. (d) CO emissions. The engine speed for these measures is 1600 rpm and it is working with the suitable proportion of turbo charging for each point.

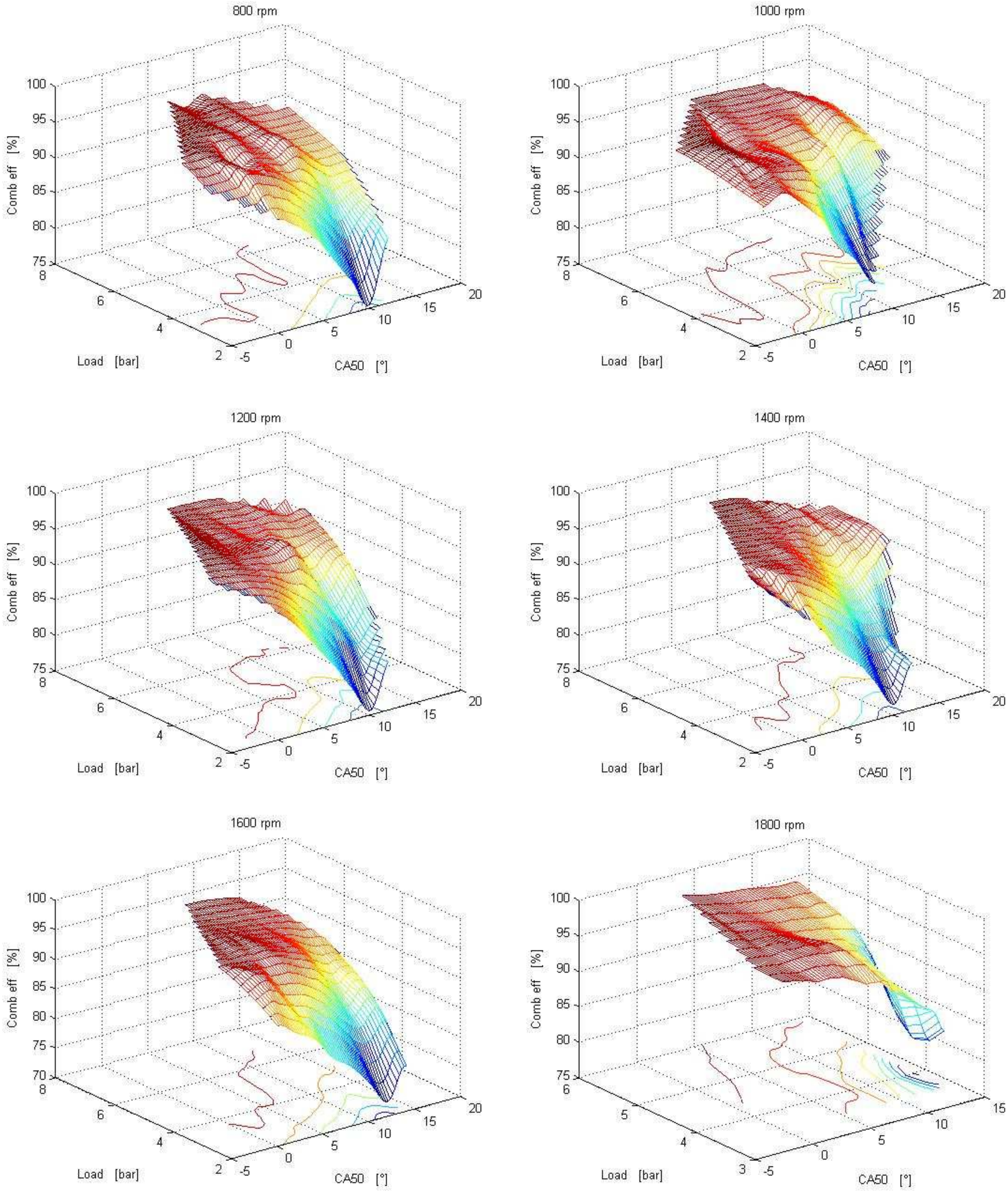


Figure 6.14 Combustion efficiency graphics for all the speed range, working the engine with the suitable proportion of turbo charging for each point.

combustion efficiency and the combustion temperature are 95.26 % and 1593 ºK. If it is produced at 14 CAD it will decrease until 88.19 % and the temperature will be 1341 ºK. The inlet temperature as the combustion temperature, also decreases, although in less proportion, between 140 ºC to 115ºC.

Another thing to analyze in Figure 6.12 is the load dependence. When the load increases its value, λ decreases so the mixture is richer and the combustion efficiency improves. The highest combustion efficiency got for this speed has been 96.7 % in 7 bars (highest load run in this master thesis) and when the combustion was produced at 6 CAD.

One important aspect to remark is when the combustion efficiency is really bad, in the lowest load and in one of the later phasing. Then the combustion is incomplete and some fuel escapes combustion producing high levels of HC and CO. This can be examined comparing Figure 6.12 with Figure 6.13 (c) and Figure 6.13 (d).

The combustion is almost independent of the engine speed as it can be appreciated in Figure 6.14. The tiny differences between the graphics can be due to inaccuracy when they are plotted given that not all the points have been able to be run for all the speeds.

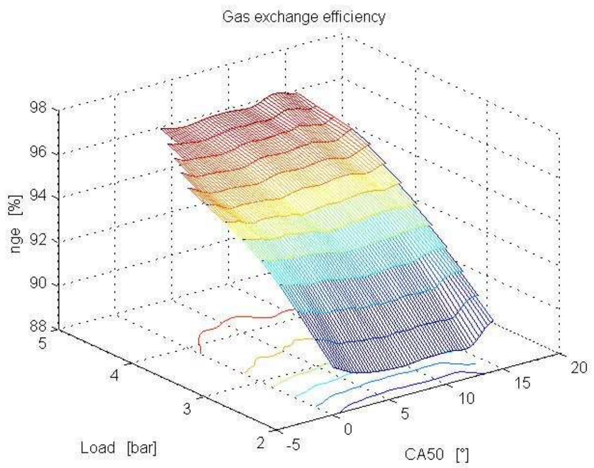
6.2.2 – Gas exchange efficiency

The HCCI engine operates unthrottled, which reduces the pumping losses at the part load compared to the conventional spark ignition engine.

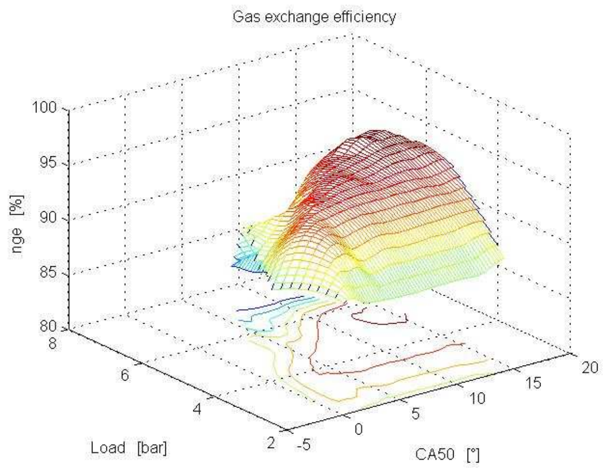
Gas exchange efficiency shows the engine losses due to the pumping work.

When the engine is running at some speed with minimum boost pressure, the pumping losses are almost constant for the same load and it is totally independence of the combustion timing. This premise can be checked in Figure 6.15 (a).

For this speed, the gas exchange efficiencies are 89, 93, 95 and 96.5 % for the loads 2, 3, 4 and 5 bars, respectively. As the speed decreases, the pumping losses are a bit smaller and the gas exchange efficiency improves. In fact, the gas exchange efficiencies at 800 rpm for the load range are 96, 97, 98 and 99 %, respectively. The graphics shape for all the speeds is quite the same than the shown one, only changing the rise rate that is a little bit higher as the engine runs faster.



(a)



(b)

Figure 6.15 Gas exchange efficiency. The engine speed for these measures is 1600 rpm and it is working with (a) minimum boost pressure. (b) the suitable proportion of turbo charging for each point

But the gas exchange efficiency is more important and shows more useful information when the engine runs with the turbo charging.

HCCI engine works with a highly diluted charge and also with a rapid combustion. Then the temperature in the cylinder is very low at the exhaust valve opening causing that the exhaust gases have a low temperature. The amount of work to make the turbo achieve the level of boost requested has to be extracted from the exhaust. With a lower temperature, this requires a higher pressure drop, which causes high pumping losses and therefore, decreases the gas exchange efficiency.

Comparing Figure 6.15 (a) and Figure 6.15 (b) it is easily appreciated when the turbo is used. First of all, the load scale has to be take into consideration because it is not the same for both Figures. The reason for that is because without the turbo charging the engine is only able to run until 5 bars and with the turbo charging the power density increases. In this master thesis, the engine has only been run until 7 bars of load. Figure 6.15 (b) shows how the gas exchange efficiency starts to be constant in 6 bars or even decreases in 7 bars instead of continue improving as the trend in Figure 6.15 (a). This is due to the use of the turbo. The other part of the

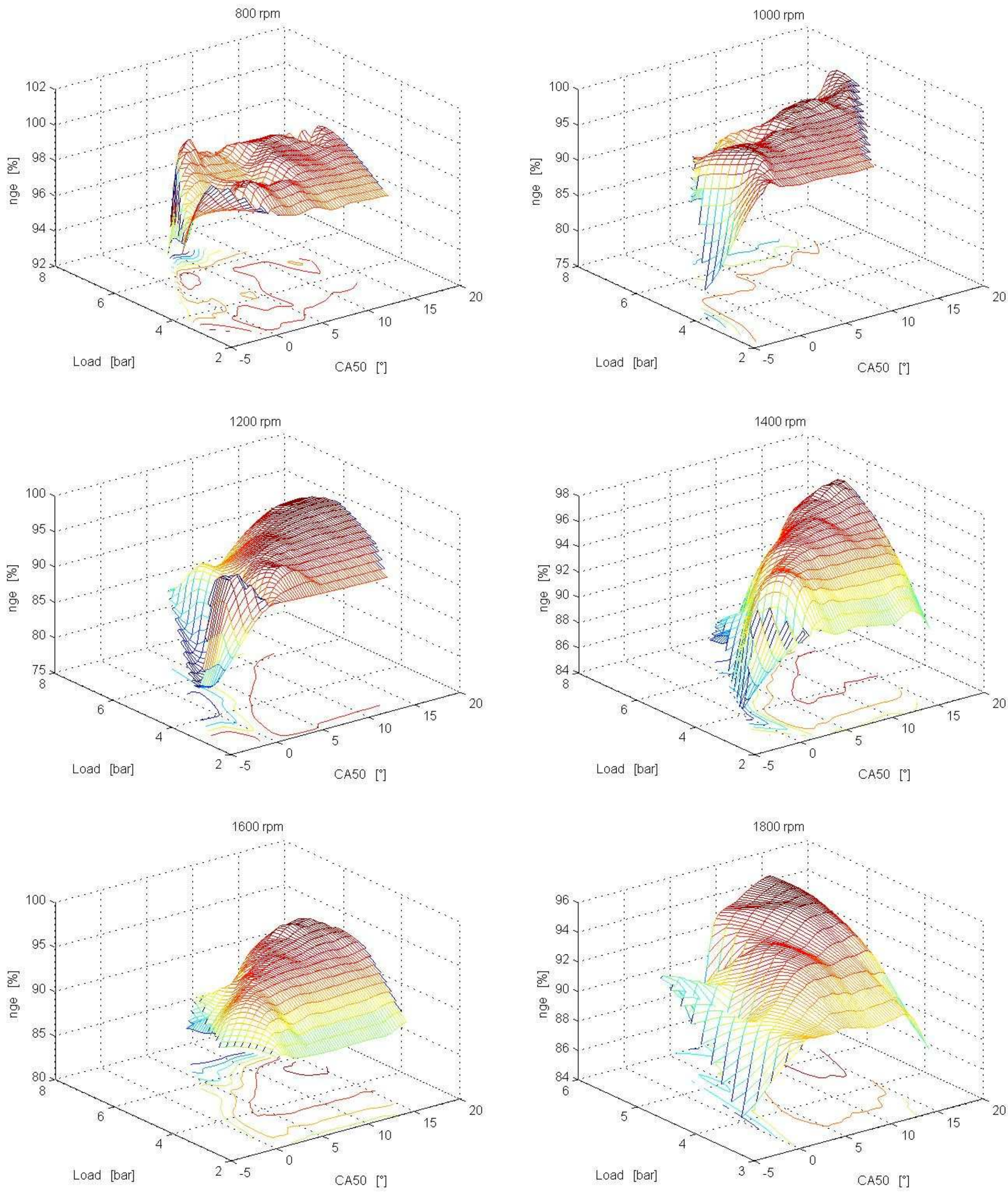


Figure 6.16 Gas exchange efficiency graphics for all the speed range, working the engine with the suitable proportion of turbo charging for each point.

map where the turbo charging is used, is in the earlier combustion timing to decrease the NO_x rate until a suitable amount (0.2 g/Kwh).

Figure 6.16 shows in the different graphics the gas exchange efficiency for all the engine speeds using the turbo charging when it is necessary. It can easily be appreciated when the turbo is used. One thing more or less interesting to remark is that the pumping losses at 800 rpm are not really high because it is more difficult for the turbo produce the suitable charge at lower speeds due to the decreased exhaust gas flow. Then its use produces less pumping losses.

Finally, this part can finish with the conclusion that the turbo charger is not really good from an efficiency stand point as it can be for the others kind o engines. But it is however needed in HCCI to increase the load range with reasonable pressure oscillations and to decrease the NO_x emissions.

6.2.3 – Brake torque efficiency

Adding up pumping losses, mechanical losses, thermal losses and losses during the combustion, it is possible to know the real engine efficiency, the brake torque efficiency.

The pumping losses and the combustion losses have already been explained in last paragraphs.

The mechanical losses are generally due to the friction work produced by the movements of the different parts of the engine. Increasing engine speed also increases the friction torque since the tensions in the oil films increase more or less linearly with speed. So the mechanical efficiency decreases when the speed is faster. This effect is difficult to appreciate in the different graphics of Figure 6.17 because the difference in this scale is really tiny. The maximum brake torque efficiency decreases from 48.47 % at 800 rpm until 43.65 % at 1800 rpm.

The heat losses are expected to be low for the HCCI engine concept. This engine always operates with a lean mixture, resulting in gas temperatures significantly lower than in any other engine type. This, together with a rapid combustion, should result in an engine with very low heat losses. Furthermore, the homogeneous mixture does not generate soot during the combustion because of that radiation is not an issue for an HCCI engine. On the other hand, the homogeneous mixture burns close to the wall, transferring heat and producing heat losses across the wall.

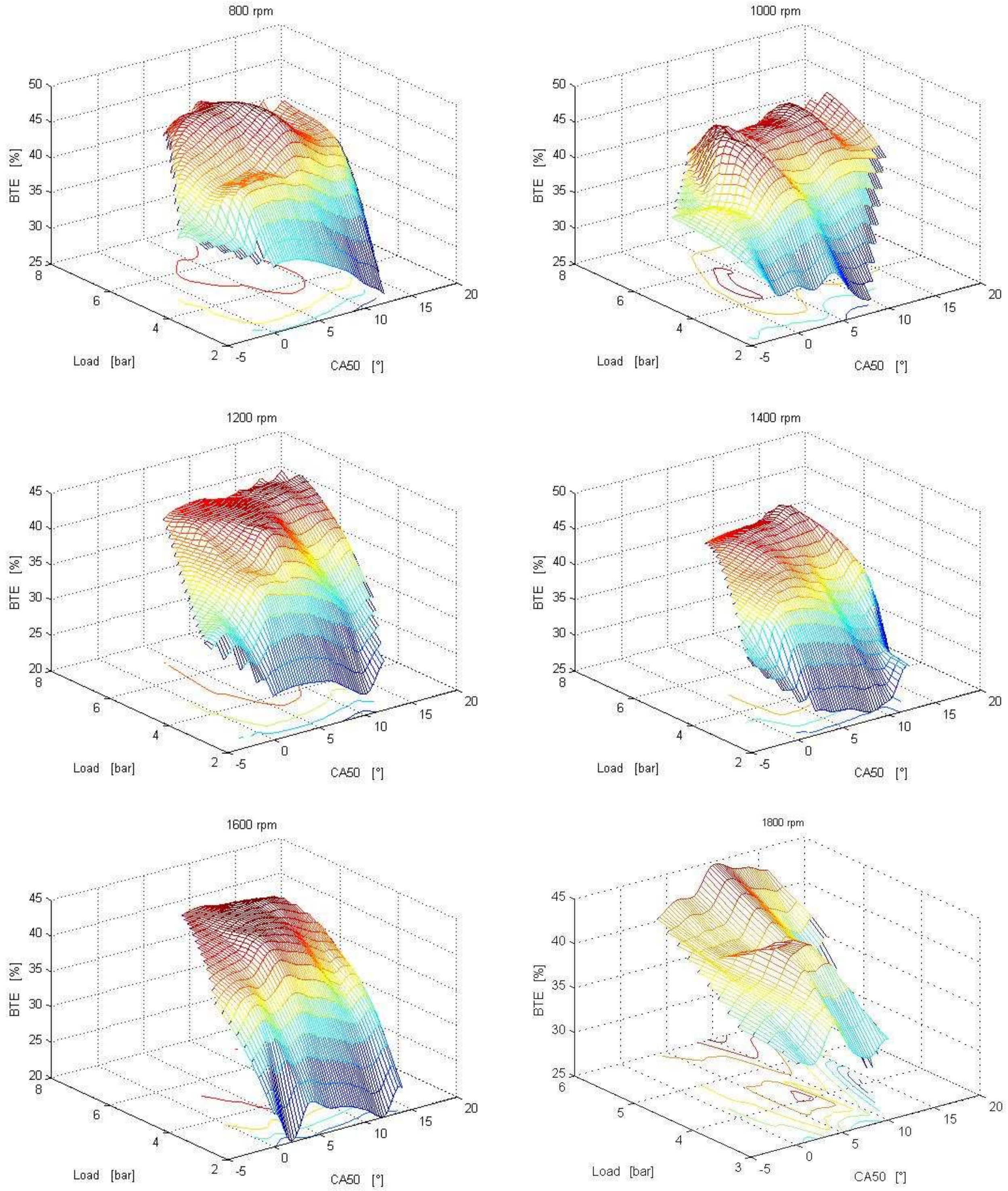


Figure 6.17 Brake torque efficiency graphics for all the speed range, working the engine with the suitable proportion of turbo charging for each point.

After that it will be studied the load and the combustion timing dependences in the brake torque efficiency in Figure 6.18. Advancing the combustion timing leads to faster heat release, steeper pressure rise and to higher combustion temperature. This results in increased heat losses and less work on the piston, so that the efficiency is reduced. But with too late timing the combustion is worse, which also reduces the efficiency. So the optimum timing varies with the load. Generally when load is increased the optimum timing is forced later into the cycle. At low loads, the main point of heat release should be close to 0 CAD while at high loads should be between 4 and 12 CAD.

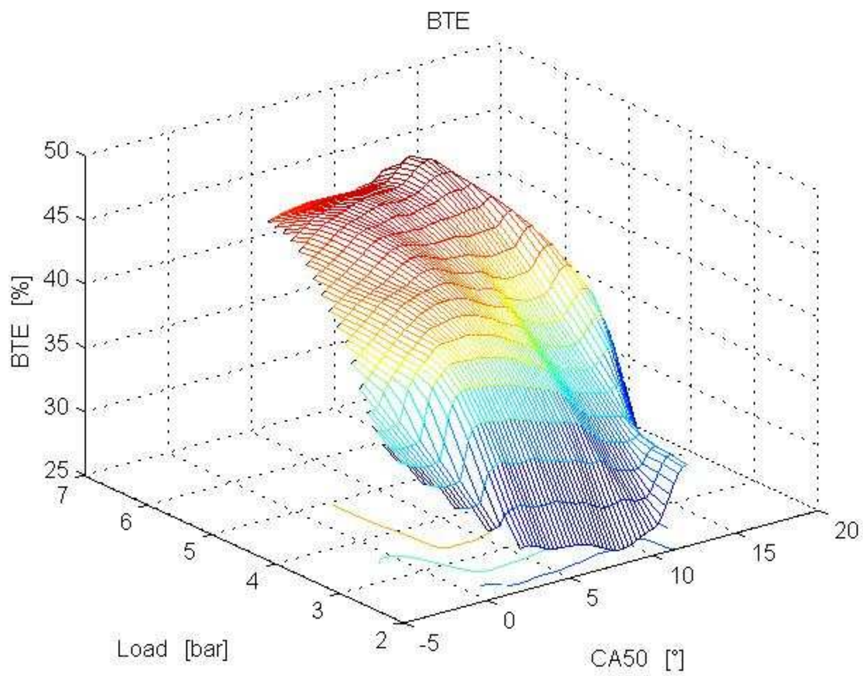


Figure 6.18 Brake torque efficiency. The engine speed for these measures is 1400 rpm and it is working with the suitable proportion of turbo charging for each point.

In the case of the Figure 6.18, the optimum timing in 2, 3, 4, 5, 6 and 7 bars of load are produced in 2, 4, 8, 10, 12 and 8 CAD respectively. And the brake torque efficiencies in these points are 26.33, 38.07, 42.1, 44.31, 46.03 and 42.2 being the maximum in 6 bars and 12 CAD.

6.2.4 – Net indicated efficiency

Net indicated efficiency is the brake torque efficiency but without including the mechanical losses.

$$\eta_{net} = \frac{BTE}{\eta_m} \qquad (6.2.4.1)$$

In this master, this parameter has been used to try to find an approximation to maximum torque. This has been done because the measure system cannot measure so much accurate torques.

Maximum net indicated efficiency and maximum torque could be approximated as the equation (6.2.4.1) shows. The only disadvantage using this parameter is that the maximum torque value is not known.

The graphics of Figure 6.20 show how the maximum net indicated efficiency is moving to later CAD as the load increases. Comparing the different graphics for all the speed range in Figure 6.20, it can be appreciated how the maximum net indicated efficiency increases with the load until 5 or 6 bars and after that decrease for the highest load.

It is good to complement this explanation the Figure 6.19.

The maximum net indicated efficiency obtained in the measures has been close to 51 % at 800 rpm and 6 bars of load. But the maximum values for all the

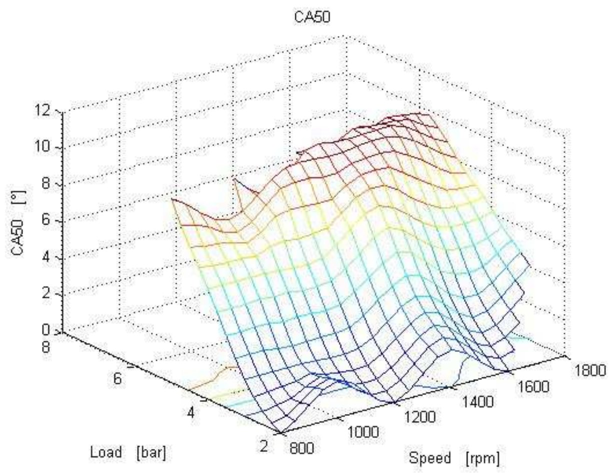


Figure 6.19 It represents in which combustion timing the maximum net indicated efficiency is obtained.

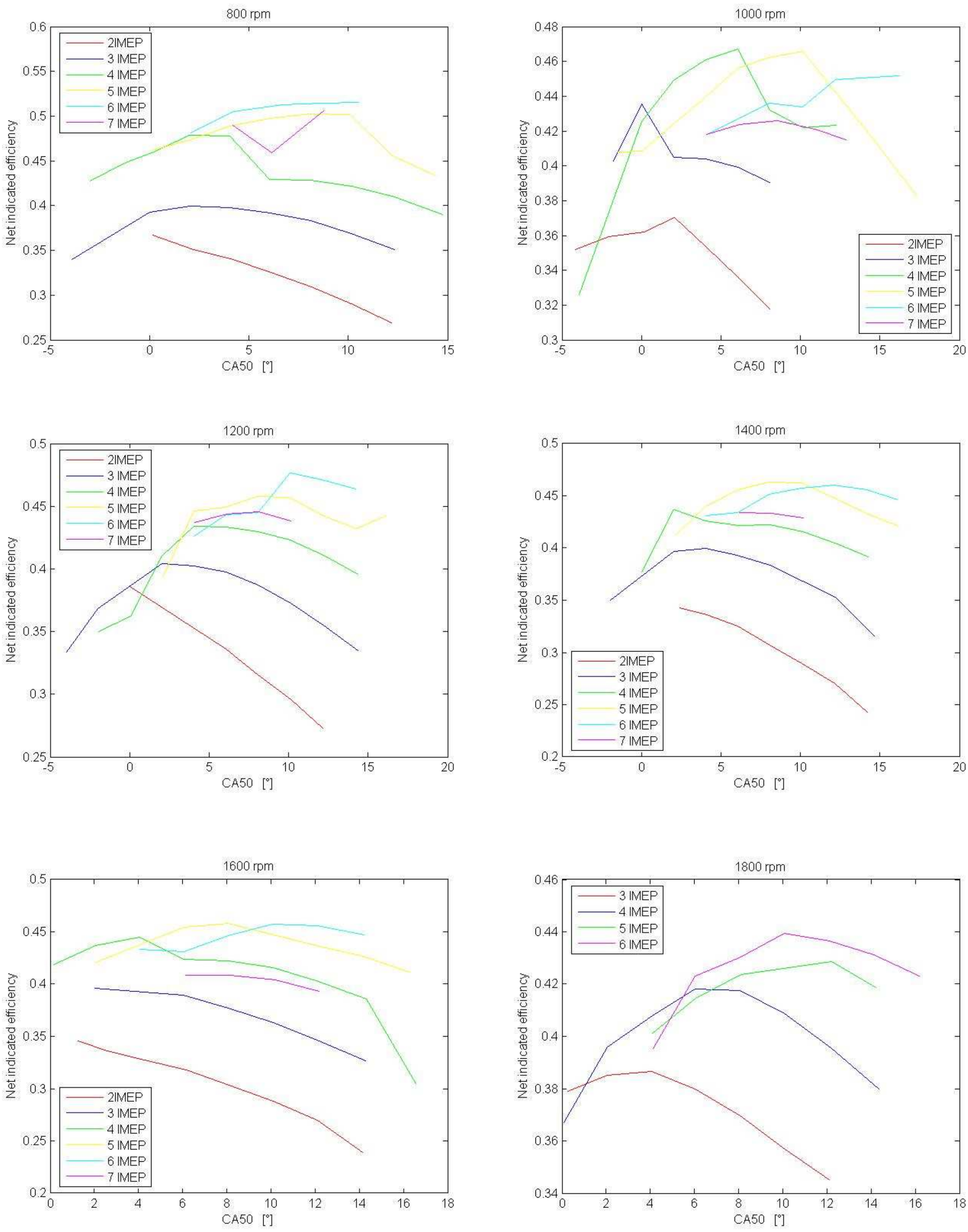


Figure 6.20 Net indicated efficiency graphics for all the speed range, working the engine with the suitable proportion of turbo charging for each point.

speeds are always between 45 to 50 %.

In the last part of the appendix are attached some graphics with the variables which are explained during this chapter in the conditions of maximum net indicated efficiency. Most of these graphics show the expected results explained in the different parts of this chapter. Only the combustion duration has an instable response.

6.3 Brake specific fuel consumption

Brake specific fuel consumption (BSFC) is a measure of an engine's efficiency. It is a rate of fuel consumption divided by the rate of power production.

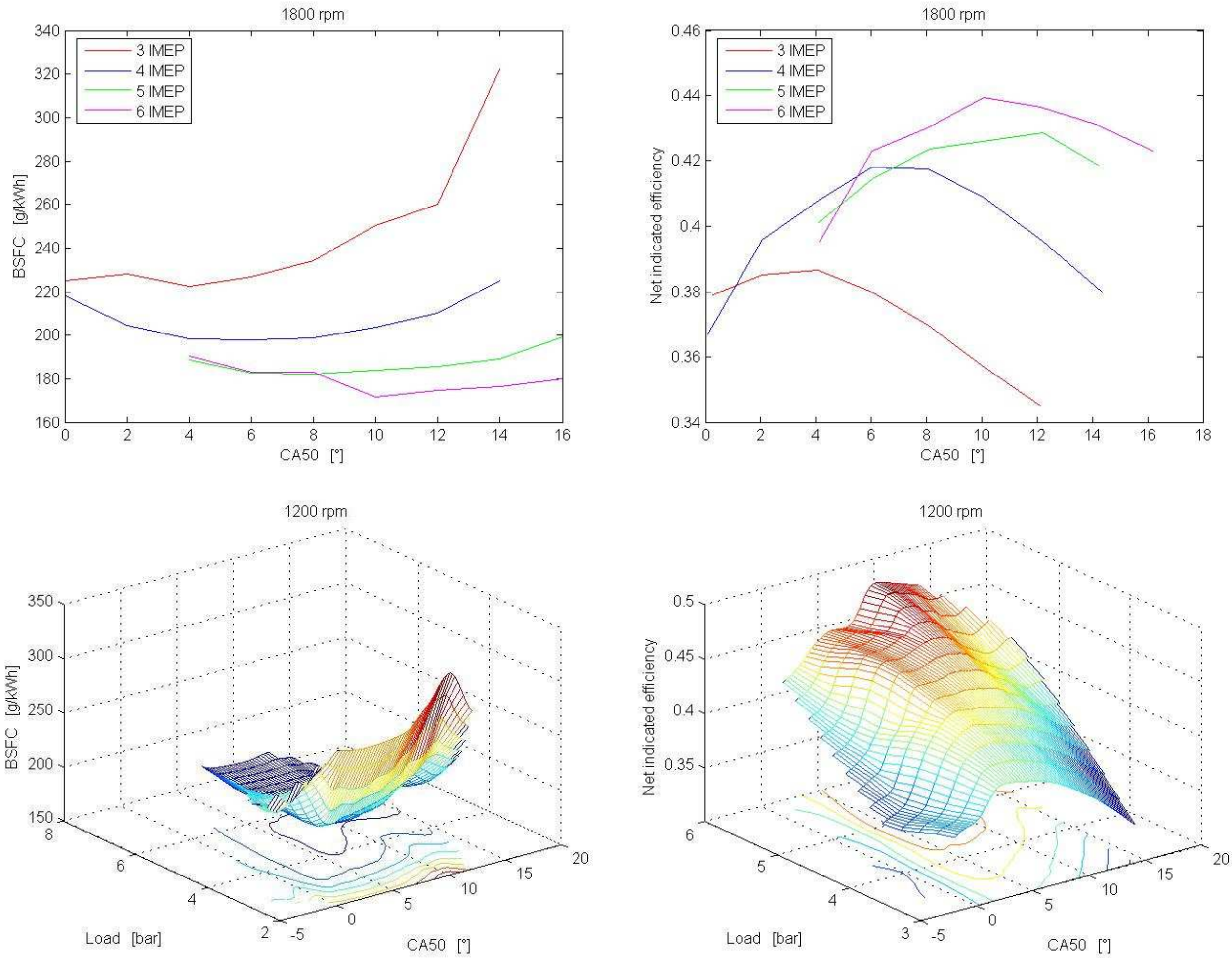


Figure 6.21 Brake specific fuel consumption and net indicated efficiency graphics for some speeds, working the engine with the suitable proportion of turbo charging for each point.

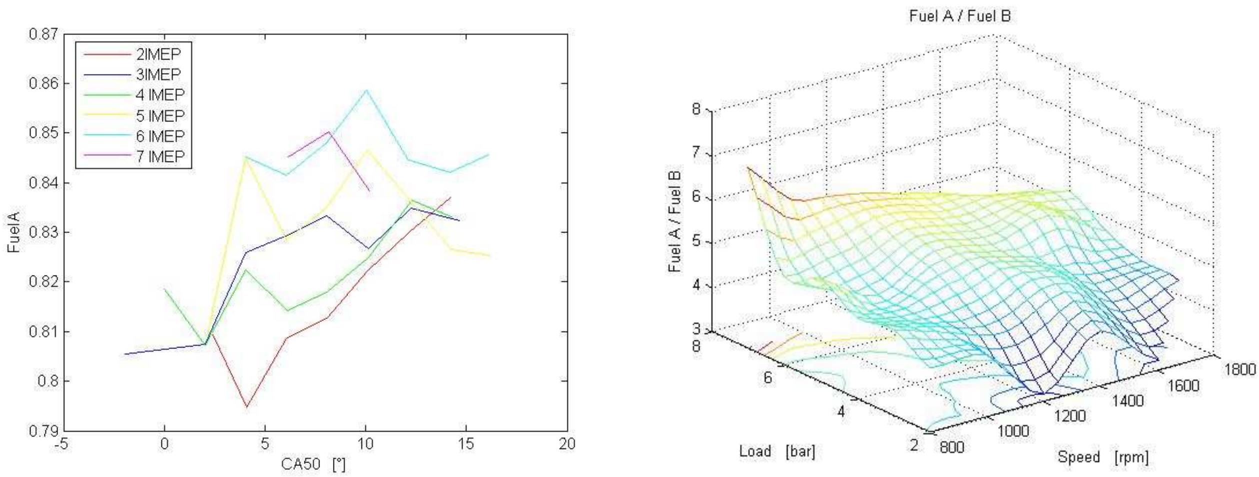
The BSFC has to be the inverse from the net indicated efficiency. Then the maximum values for net indicated efficiency correspond to the minimum values for BSFC. The graphics of Figure 6.21 show this concept.

One comment for the high consumption points is if the combustion efficiency is not really good, that means fuel escapes combustion, then more fuel consumption is necessary to reach the load required.

6.3.1 – Different fuel use

As it was explained before in the theoretical part of this report, the fuels used are natural gas and n-heptane. Before this present work started, it was decided working with the less possible amount of n-heptane. But because of the controllers, this amount has to be one minimum. It was decided to try to work with 15-20 % of n-heptane to run the engine properly without taking the risk of controller's failures.

The ratio natural gas/n-heptane in some measures can be looked at Figure 6.22. The ratio should be between 80/20=4 and 85/15=5.67, but some points differ a little bit in the maps limit where the conditions are more irregular.



(a) (b)

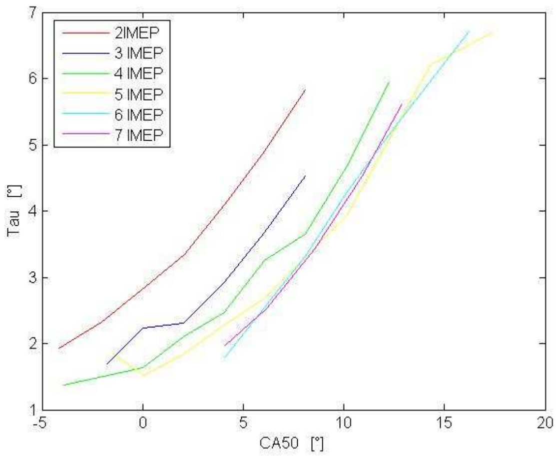
Figure 6.22 Natural gas / n-heptane relation working the engine (a) at 1400 rpm (b) in maximum net indicated efficiency condition.

6.4 Combustion duration

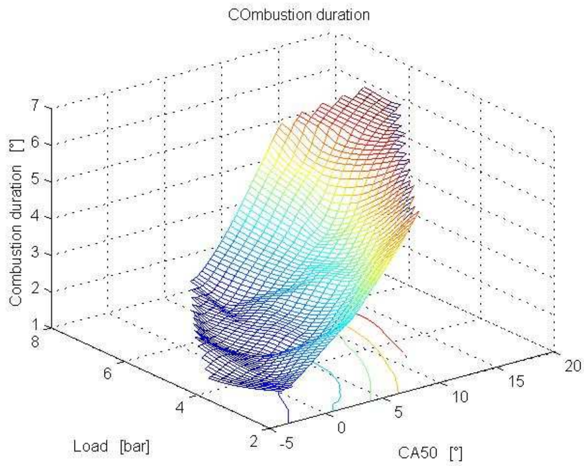
The combustion duration measures how fast the combustion process is. This can be divided into three different parts:

1) Early combustion phase lasts from the autoignition occurs until 10 percent of the total heat released.
2) Rapid burning phase lasts between 10 to 90 percent of the total heat released.
3) Late phase lasts from 90 to 100 percent heat released.

As these phases are not really accurate, the combustion duration has been estimated to the position of 10 and 90 burned and the parameter unit used in this thesis is crank angle degree.



(a)



(b)

Figure 6.23 Combustion duration. The engine speed for these measures is 1000 rpm and it is working with the suitable proportion of turbo charging for each point.

The main factor, which affects combustion duration, is the combustion timing. As the combustion timing delays, the combustion duration is longer since the maximum cycle temperature decreases. This premise can be analyzed in Figure 6.23 (a).

As regards load dependence, Figure 6.23 (b) shows that the combustion lasts more or less the same for all the load range.

6.5 Measures deviations

In this present work, three main variables are varied throughout a large ranges; speed (rpm), combustion timing (CA50) and load ($IMEP_n$).

The engine speed is controlled by the dynamometer and its value is almost perfect. But the load and the combustion timing are managed by PID controllers that they do not give an exact response. This is the reason to analyze their little deviations.

6.5.1 – Combustion timing

The combustion timing will be analyzed with the standard deviation, which measures how spread out the values in a data are.

Figure 6.24 shows one graphic shape with a behaviour quite similar for all the speeds. The accuracy for the part load is really good because the standard deviation is close to 0 (dark blue in the graphic). The zones where the measures are less precise

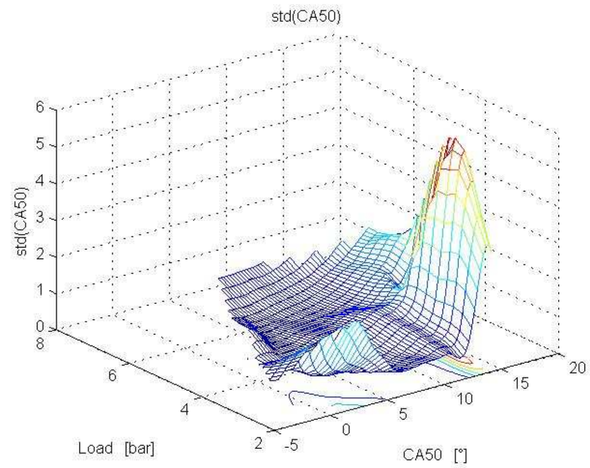


Figure 6.24 Standard deviation for the combustion timing

are generally in the earlier points with the lowest load and the ones with later combustion timing in all their load range. The reason for this it could be that in the latest points the combustion is not really good and the general engine behaviour is so instable.

6.5.2 – Load

The coefficient of variation is going to be the method used to analyze the load. It is defined by:

$$COV(IMEP_n) = \frac{std(IMEP_n)}{mean(IMEP_n)} \qquad (6.5.2.1)$$

It is a form to measure the dispersion of a probability distribution. It is used because the standard deviation for the load is significantly less than its mean value.

As it has been made with the combustion timing, Figure 6.25 shows one graphic shape with a common repeat behaviour in the most of the speeds. The part load is again the more accurate zone, as the latest combustion timing is the less precise. There is only one change against the timing. Instead of the bad part in the low loads, here it is in the highest load where is less accurate than the main part but anyway it is not really bad.

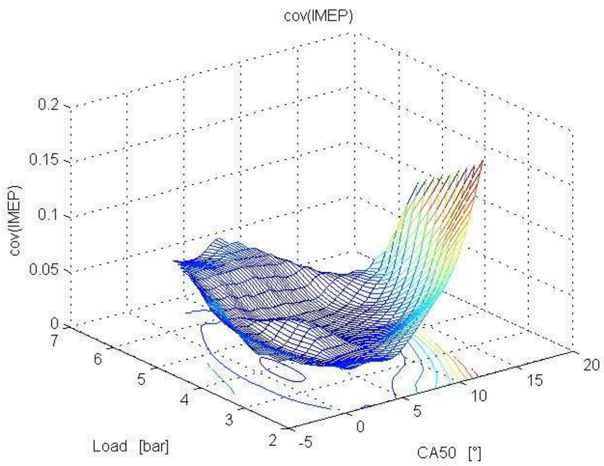


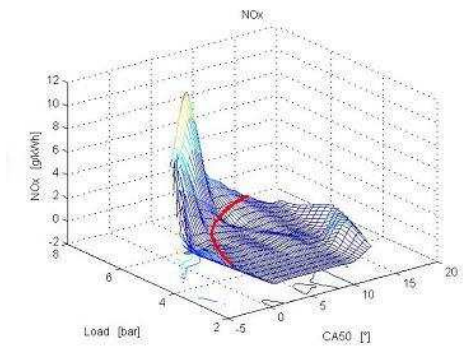
Figure 6.25 Coefficient of variation for the load

7 CONCLUSIONS

The emissions and the efficiencies are the most important results for the conclusions of this master thesis.

7.1 Emissions

The combustion process in HCCI engines is very fast and it works with a highly diluted mixture causing low combustion temperature. The engine usually works in these conditions in its part load producing almost zero NO_x. But when the conditions are harder like in early combustion timings or high loads the NO_x rate increases.



The NO_x limit considered in this work is 0.2 g/Kwh. This limit is reached dependent on speed in specific combustion timing for each load. Approximately this limit can be found in the graphics used in this report tracing one imaginary ¼ circle (traced in the figure in red colour) with centre in the latest timing and in the lowest load and with the distance to the first NO_x rate higher than 0.2 g/Kwh as radius.

The limit temperature in which the NO_x amount starts to increase steeply is around 1700 ºK and the λ limit is around 2.5.

Charging the engine with the help of turbo, it is possible decrease the NO_x rates under the suitable rate 0.2 g/Kwh. But anyway if this mark is not reach, the NO_x rate will decrease a significant amount that in fact it is one of the goals.

Another aspect to remark is when the combustion timing is in their later points, then the chamber temperature is really low and the combustion starts to falter causing bad combustion and higher $NO_{x.}$ This unexpected growth is due to work with specific emissions.

But these HCCI combustion conditions are not good for all the emissions, specifically much more HC and CO are produced due to the low temperatures.

The HC and CO emissions increases as the loads are lower and the combustion timing are later or in other words, as the combustion temperature decreases and the mixture is leaner. Their behaviour throughout the entire mapping is quite similar for both emissions.

When these emissions are really high, the combustion efficiency will be affected because of fuel escapes combustion producing combustion related losses.

Finally, tell that these emissions are not really important either because they can be reduced to an insignificant amounts using oxidizing suitable catalyst.

7.2 Efficiencies

With HCCI engine, higher efficiencies in the part load are expected because it works with low combustion temperature, short combustion period and unthrottled operation.

The best obtained brake torque efficiency in all the measurements is a little bit more than 48 % that is a really good efficiency comparing with the other engine types. It is a bit curious because normally the lowest loads are related with bad combustion and the efficiency expected was lower.

The normal brake torque efficiency obtained at part load conditions is always around 45%.

The gas exchange efficiency is a parameter really interesting in this thesis work because it shows the pumping losses.

When the turbo charger is used either to decrease the NO_x rates or to increase the power density, it causes high pumping losses because of the exhaust gases have a low temperature.

This indicates that an intermediate point relating the pumping losses, the NO_x emissions and the power density has to be found to reach the best work conditions. Because if the turbo charger is used more than it should, its use starts to be less interesting from an efficiency stand point.

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A Appendix

A.1 Matlab scripts

Program 1

```
%__CONTOURS MAPS FOR DIFFERENT VARIABLES REGARDING LOAD AND CA50__%

 clear all,close all
 cd('D:\Rafa\Slow computer\turbo800 rpm');   %Do not forget change the directory in the line 165 as well
 files=dir('*.HP');
 DispVol=0.011705;   %m3
 rpm=[];   load=[];   timing=[];
 long = length(files);
for i=1:long
    rpm=[rpm;str2num(files(i).name(8:11))];
    load=[load;str2num(files(i).name(13:14))];
    timing=[timing;str2num(files(i).name(16:17))];
end
 dat=[rpm load timing];
[arrangedat,index] = sortrows(dat,[1:3]);
for i=1:long
     var=dlmread(files(index(i)).name,'\t',1,0);
     med(i,:)=mean(var);      % med = mean value
e(i,:)=emissions(DispVol,arrangedat(i,1),arrangedat(i,2),med(i,37),med(i,43),med(i,45)/1000000,med(i,50)/1000000,med(i,54)/1000000,med(i,52)/100,med(i,51)/100,'NG','
NHeptane'); % NG_flow   Gasoline_flow  x_HC  x_NOx  x_CO  x_CO2   x_O2
end
% Variable definition to plot them
z=med(:,[45 50 53 54 66 69 70]);       %[THC  NOx  COhigh  COlow  Combeff  BSFC  BTE]
ca50=arrangedat(:,3);
fuelAB=med(:,37)./med(:,43);
lambda=med(:,59);

% ______PLOTS_______  %

%%__THC__%%
 figure
[XI, YI]=meshgrid(-4:.25:16, 2:.25:6);
ZI = griddata(ca50,load,e(:,8),XI,YI, 'cubic');
meshc(XI,YI,ZI), hold
title('Total HC')
xlabel('CA50   [°]')
ylabel('Load   [bar]')
zlabel('THC   [g/kWh]')
figure
[C,h]=contour(XI, YI,round(ZI),20);
clabel(C,h);
title('Total HC')
xlabel('CA50   [°]')
ylabel('Load   [bar]')

IS THE SAME  PROCEDURE FOR ALL THE  PLOTS

 %___________________________________________________________________________%

 clear all
 cd('D:\Rafa\Fast computer\Press_mat\turbo800 rpm');                         %Choose the speed you want to plot
 files=dir('*.mat');
 rpm=[]; timing=[]; carga=[];  % carga = load
long = length(files);
for i=1:long
    rpm=[rpm;str2num(files(i).name(8:11))];
    carga=[carga;str2num(files(i).name(13:14))];
    timing=[timing;str2num(files(i).name(16:17))];
end
 dat=[rpm carga timing];
[arrangedat,index] = sortrows(dat,[1:3]);
 for i=1:long
     load(files(index(i)).name);
     stdCA50(i)=std(CA50);
     stdIMEP(i)=std(IMEPn);
     medIMEP(i)=mean(IMEPn);
     cov(i)=stdIMEP(i)/medIMEP(i);
     medPCPD(i)=mean(PCPD);             %PCPD=(dP/dCAD)max
     medFuelA(i)=mean(FuelA);
end

 % ______PLOTS_______  %

IS THE SAME PRCEDURE THAN BEFORE  FOR THIS VARIABLES
```

Program 2

```
%_____GRAPHICS FOR DIFFERENT PARAMETERS IN MAXIMUM NET INDICATED EFFICIENCY CONDITION REGARDIND SPEED AND LOAD______%

clear all,close all
cd('D:\Rafa\Slow computer\All dates together');
  files=dir('*.HP');
 DispVol=0.011705;   %m3
 rpm=[]; timing=[]; carga=[];   % carga= load
long = length(files);
for i=1:long
    rpm=[rpm;str2num(files(i).name(8:11))];
    carga=[carga;str2num(files(i).name(13:14))];
    timing=[timing;str2num(files(i).name(16:17))];
end
dat=[rpm carga timing];
[arrangedat,index] = sortrows(dat,[1:3]);
for i=1:long
     var=dlmread(files(index(i)).name,'\t',1,0);
     med(i,:)=mean(var);      % med = mean value
e(i,:)=emissions(DispVol,arrangedat(i,1),arrangedat(i,2),med(i,37),med(i,43),med(i,45)/1000000,med(i,50)/1000000,med(i,54)/1000000,med(i,52)/100,med(i,51)/100,'NG','
NHeptane'); % NG_flow   Gasoline_flow  x_HC  x_NOx  x_CO  x_CO2   x_O2
end
% Find the maximum torque, saving in  ca50 to know what file we are using % and to know its mean values
cd('D:\Rafa\Fast computer\Press_mat\All dates together');
files=dir('*.mat');
for i=1:long
     load(files(index(i)).name);
     medNetindiceff(i)=mean(IMEPn./FuelMEP);
end
maxtorq=medNetindiceff(1);
maxtorqfile=files(index(1));
maxtorqmed(1,:)=med(1,:);
maxtorqe(1,:)=e(1,:);
j=1;
for i=1:(long-1)
    if arrangedat(i,1)==arrangedat(i+1,1) & arrangedat(i,2)==arrangedat(i+1,2)
        if medNetindiceff(i+1)>=maxtorq(j)
            maxtorq(j)=medNetindiceff(i+1);
            maxtorqfile(j)=files(index(i+1));
            maxtorqmed(j,:)=med(i+1,:);
            maxtorqe(j,:)=e(i+1,:);
        end
    else j=j+1;
        maxtorq(j)=medNetindiceff(i+1);
        maxtorqfile(j)=files(index(i+1));
        maxtorqmed(j,:)=med(i+1,:);
        maxtorqe(j,:)=e(i+1,:);
    end
end
% Variable definition to plot
for i=1:j
     load(maxtorqfile(i).name);
     stdCA50(i)=std(CA50);
     stdIMEP(i)=std(IMEPn);
     medIMEP(i)=mean(IMEPn);
     cov(i)=stdIMEP(i)/medIMEP(i);
     medTau(i)=mean(CA80-CA20);
     medFuelA(i)=mean(FuelA);
end
z=maxtorqmed(:,[45 50 53 54 66 69 70]);       %[THC  NOx  COhigh  COlow  Combeff  BSFC  BTE]
ca50=[];
speed=[];
load=[];
for i=1:j
ca50=[ca50;str2num(maxtorqfile(i).name(16:17))];
speed=[speed;str2num(maxtorqfile(i).name(8:11))];
load=[load;str2num(maxtorqfile(i).name(13:14))];
end
datas=[speed load ca50 maxtorqe(:,8) maxtorqe(:,10) maxtorqe(:,9) z(:,5) z(:,6) z(:,7) stdCA50' cov' medTau' (medFuelA./(1-medFuelA))'];

% ______PLOTS_______  %

%%__CA50 @ MBT__%%
figure
[XI, YI]=meshgrid(800:50:1800, 2:.25:6);
ZI = griddata(speed,load,ca50,XI,YI, 'cubic');
meshc(XI,YI,ZI), hold
title('CA50')
xlabel('Speed   [rpm]')
ylabel('Load   [bar]')
zlabel('CA50   [°]')

IS THE SAME PROCEDURE FOR ALL THE  PLOTS
```

Program 3

```
%_______GRAPHICS OF DIFFERENT LOADS REGARDING FUELA AND CA50________%

clear all,close all
cd('D:\Rafa\Fast computer\Press_mat\turbo1000 rpm');       %Do not forget change the directory in the line 90 as well
 files=dir('*.mat');
 rpm=[];  timing=[]; carga=[];  % carga = load
 long = length(files);
for i=1:long
    rpm=[rpm;str2num(files(i).name(8:11))];
    carga=[carga;str2num(files(i).name(13:14))];
    timing=[timing;str2num(files(i).name(16:17))];
end
dat=[rpm carga timing];
[arrangedat,index] = sortrows(dat,[1:3]);
for i=1:long
     load(files(index(i)).name);
     medCA50(i)=mean(CA50);
     medFuelA(i)=mean(FuelA);
     medTau(i)=mean(CA80-CA20);
     medNetindiceff(i)=mean(IMEPn./FuelMEP);
end
color = ['r' 'b' 'g' 'y'];
j=1;k=1;d=1;
for i=1:(long-1)
    if arrangedat(i,2)==arrangedat(i+1,2)
        j=j+1;
    else plot(medCA50(k:k+j-1),medFuelA(k:k+j-1),color(d));hold on
        k=k+j;
        j=1;
        d=d+1;
    end
end
plot(medCA50(k:k+j-1),medFuelA(k:k+j-1),'m');hold on
xlabel('CA50   [°]')
ylabel('FuelA   [%]')
legend('2 IMEP','3 IMEP','4 IMEP','5 IMEP','6 IMEP',4)
```

IS THE SAME PROCEDURE FOR ALL THE PLOTS

Program 4

```
%__CONTOURS MAPS OF GAS EXCHANGE EFFICIENCY  REGARDING LOAD AND CA50__%

clear all, close all
%_________________________Constants____________________________%
%Mesurement resolution
CADstep=0.2;
%%Dimentions (Bore)(m)
B = 0.127;
%cranshaft radius (Stroke)(m)
r = 0.154/2;
%connecting rod length (m)
l = 0.255;
%compression ratio
rc = 18;
%File type for .bin files
ftype='int16';
%Number of cycles in the process
nbrCycles=500;
%Valve timing (temperature mass lock interval)
IVC=243;
EVO=458;
%Mass lock variables (related to IVC)
MassLWind=10;
MassLoffs=-5;
%CAD sensor offset
offs=-0.2;
%Pressure tranducer constant
preConst=305;
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
cycleSize=round(720/CADstep);     %=3600,means one cycle( two loops)
CAD=0:CADstep:720;                % two loops
MassLWind=round(MassLWind/CADstep);
MassLoffs=round(MassLoffs/CADstep);
IVC=round(IVC/CADstep);
EVO=round(EVO/CADstep);
offs=round(offs/CADstep);
%Displacement volume (m^3)
Vd = pi*B*B/4*2*r;
%Volume as a function of CAD
theta = CAD*(pi/180);
Vs = pi*B*B/4.*(l + r - r.*cos(theta) - sqrt(l*l - (r.*sin(theta)).^2));
Vc = pi*B*B/4*2*r/(rc-1);
V = (Vc + Vs)';

%Inlet pressure and Intake temperature
cd('D:\Rafa\Fast computer\Press_mat\1000 rpm');   %Do not forget change the directory in the line 89 as well
 files=dir('*.mat');
 rpm=[];
 carg=[];  % carga = load
 timing=[];
 long = length(files);
```

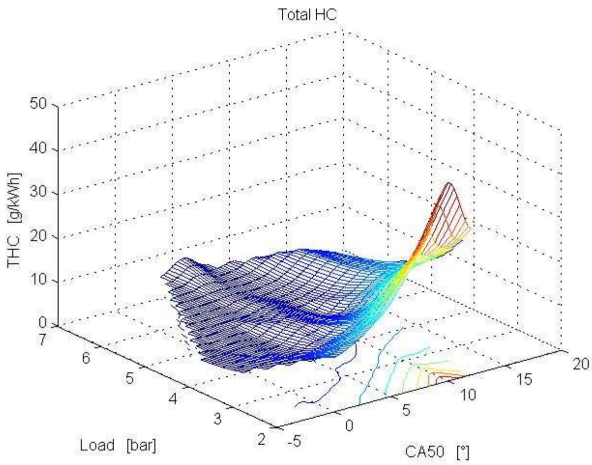
```
for i=1:long
    rpm=[rpm;str2num(files(i).name(8:11))];
    carg=[carg;str2num(files(i).name(13:14))];
    timing=[timing;str2num(files(i).name(16:17))];
end
dat=[rpm carg timing];
[arrangedat,index] = sortrows(dat,[1:3]);
fid=fopen('datas2.txt');
dat=char(fread(fid,'char')');
fclose(fid);
jj=0;
for i=1:long
    cd('D:\Rafa\Fast computer\Press_mat\1000 rpm');                          %Choose the speed you want to plot
    speed=dat(jj+1:jj+4);
    carga=dat(jj+5:jj+6);
    ca50=dat(jj+7:jj+8);
    fecha=dat(jj+9:jj+14);
    hora=dat(jj+15:jj+18);
    load([fecha '_' speed '_' carga '_' ca50 '_Status.mat']);
    Tinsug(i)=mean(TIn)+273.13;
    pInl(i)=mean(PIn)*1e5;

%%%%%%%%%%%%%%%%%%%Reads pressure and peggs%%%%%%%%%%%%%%%%%%%
cd('D:\Rafa\Fast computer');
for j=1:6
    preFile=[fecha hora 'Cyl Press' num2str(j) '.bin'];
    binfile=fopen([preFile],'r');
    cylPress=fread(binfile,[round(720/CADstep),nbrCycles],ftype);
    fclose(binfile);
    if offs>0,
        cylPress=[cylPress(offs:cycleSize,:); cylPress(1:offs,:)];
    elseif offs<0,
        cylPress=[cylPress((cycleSize+offs):cycleSize,:); cylPress(1:(cycleSize+offs),:)];
    else
        cylPress=[cylPress; cylPress(cycleSize,:)]; end
    cylPress(cycleSize+1,:)=cylPress(1,:);
    %Calculate pressure from AD convertion
    cylPress=preConst.*cylPress;
    %Pegg inlet pressure
    pGp=mean(cylPress(1:IVC,:));
    pOffs=pInl(i)-pGp;
    for a=1:nbrCycles,
        cylPress(:,a)=cylPress(:,a)+pOffs(a)
    end
    medcylPress1(:,j)=mean(cylPress,2);
end
medcylPress2(:,1)=mean(medcylPress1,2);
IMEPg(i)=sum(medcylPress2(901:2701).*gradient(V(901:2701)));
IMEPnt(i)=sum(medcylPress2(1:3601).*gradient(V(1:3601)));
PMEP(i)=IMEPg(i)-IMEPnt(i);
ngc(i)=IMEPnt(i)./IMEPg(i);
jj=jj+18;

 %%%%%%%%%%%%%%%%%%%%%%Calculate temperature%%%%%%%%%%%%%%%%%%%%%
    for a=1:nbrCycles
        mR(a)=mean(cylPress((IVC+MassLoffs-round(MassLWind/2)):(IVC+MassLoffs+round(MassLWind/2)),a).*V((IVC+MassLoffs-
round(MassLWind/2)):(IVC+MassLoffs+round(MassLWind/2)))./Tinsug(i));
    end
        t=ones(size(cylPress));
    t(1:IVC,:)=t(1:IVC,:).*Tinsug(i);
    for a=1:nbrCycles,
        t(IVC:EVO-1,a)=cylPress(IVC:EVO-1,a).*V(IVC:EVO-1)./mR(a);
        t(EVO:3601,a)=t(EVO:3601,a).*t(EVO-1,a);
    end
        tmed(:,1)=mean(t,2);
    tmax(i)=max(tmed);
      end
 datas=[arrangedat tmax' ngc' PMEP'];

% ______PLOTS_______  %

%%__Gas exchange efficiency__%%
figure
[XI, YI]=meshgrid(-4:.25:16, 2:.25:6);
ZI = griddata(arrangedat(:,3),arrangedat(:,2),ngc,XI,YI, 'cubic');
meshc(XI,YI,ZI), hold
title('Gas exchange efficiency')
xlabel('CA50   [°]')
ylabel('Load   [bar]')
zlabel('nge')
figure
[C,h]=contour(XI, YI,round(ZI),20);
clabel(C,h);
title('Gas exchange efficiency')
xlabel('CA50   [°]')
ylabel('Load   [bar]')
```

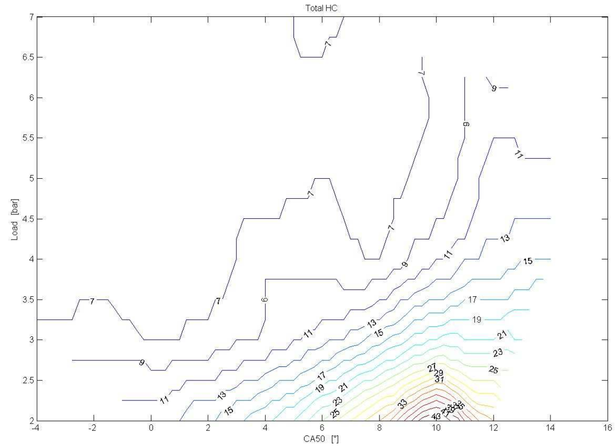
A.2 Figures overview

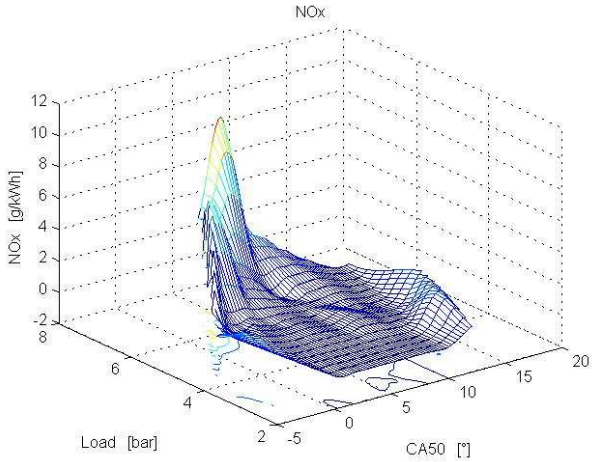
The emissions graphics for all sweeps of the measurement campaign are presented. First are shown the 3D plots and the contour maps in each engine speed. Afterwards the 2D plots are represented in each speed as well. And finally the 3D plots and the contour maps when the engine is running in the maximum net indicated efficiency condition are attached.

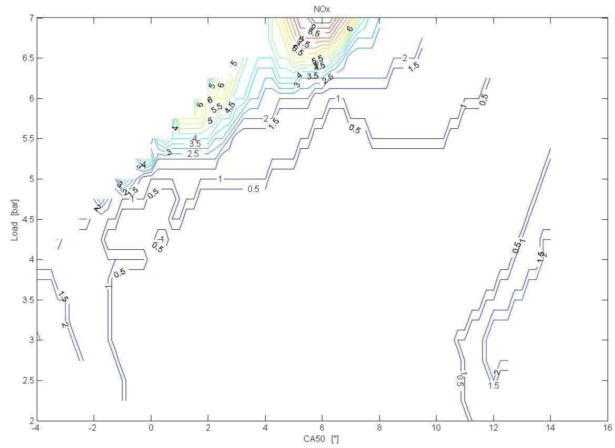
These sweeps are discussed in the results chapter.

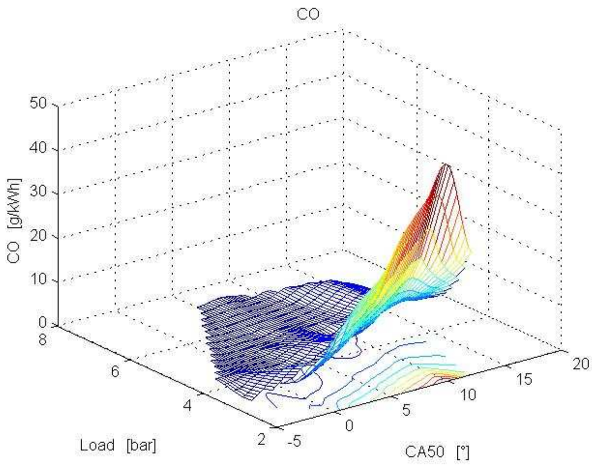
800 rpm

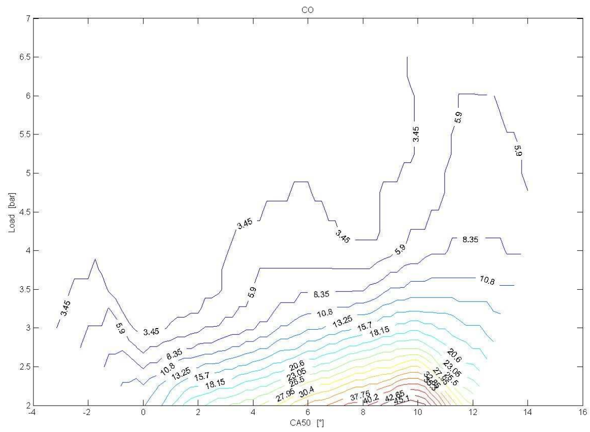

Total HC
THC [g/kWh]
Load [bar]
CA50 [°]

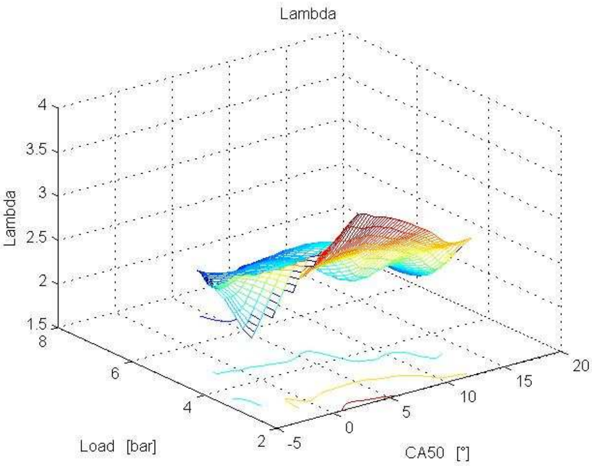

Total HC
Load [bar]
CA50 [°]

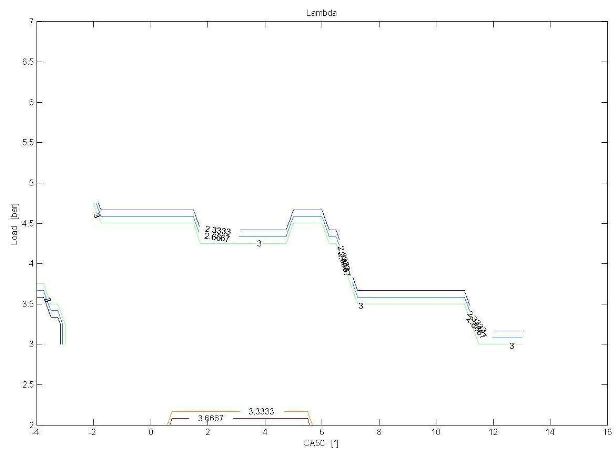

NOx
NOx [g/kWh]
Load [bar]
CA50 [°]


NOx
Load [bar]
CA50 [°]

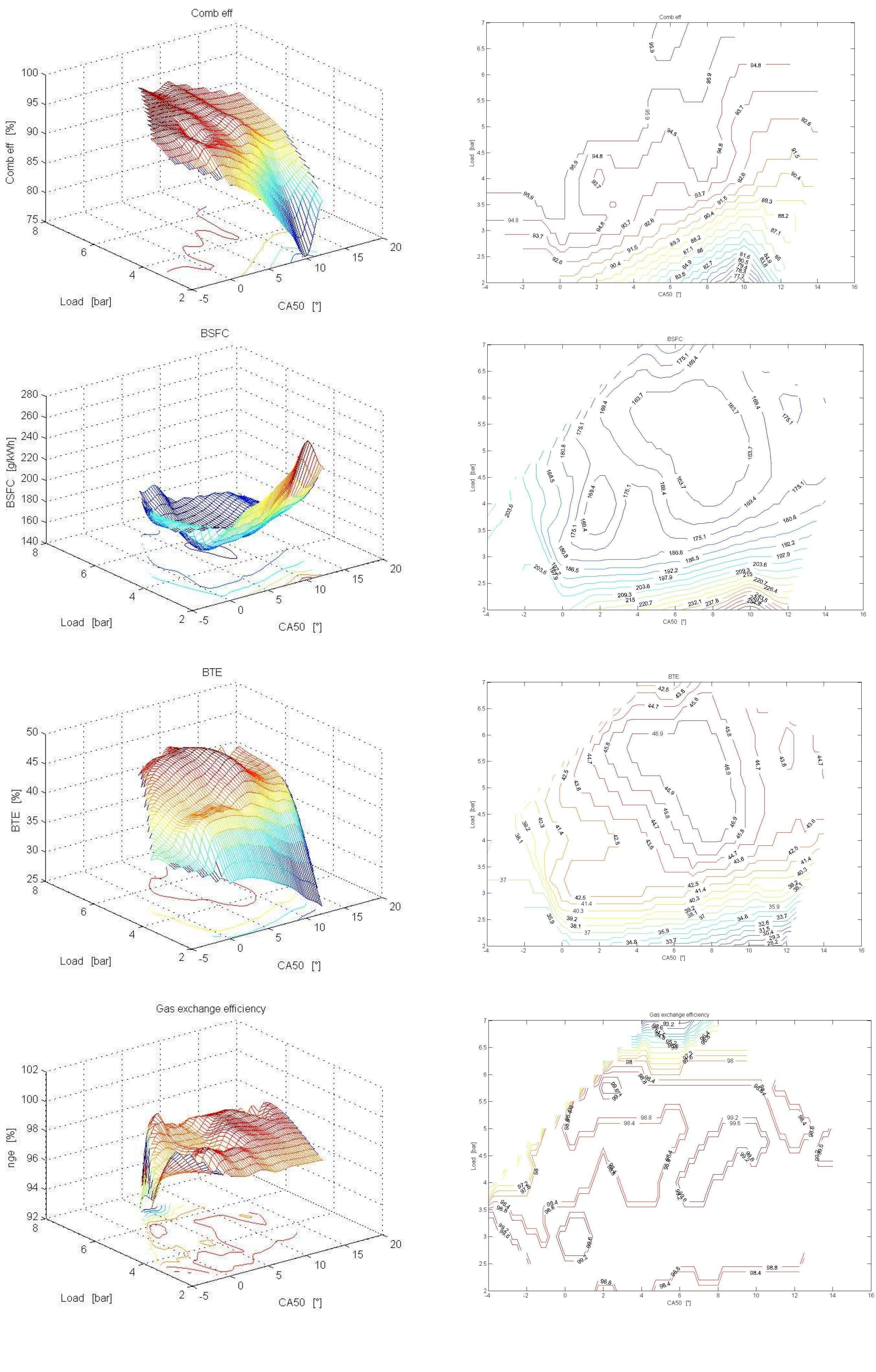

CO
CO [g/kWh]
Load [bar]
CA50 [°]


CO
Load [bar]
CA50 [°]

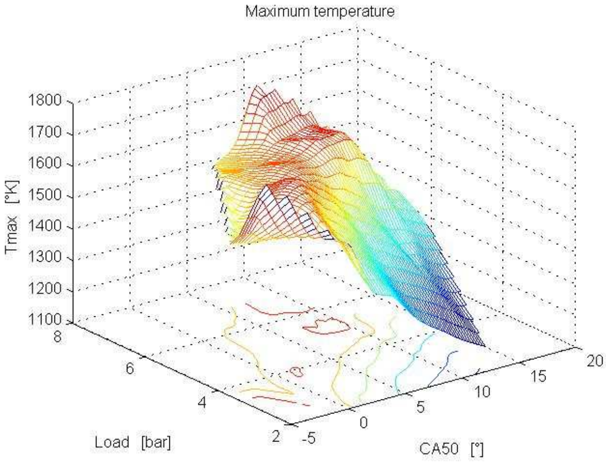

Lambda
Lambda
Load [bar]
CA50 [°]

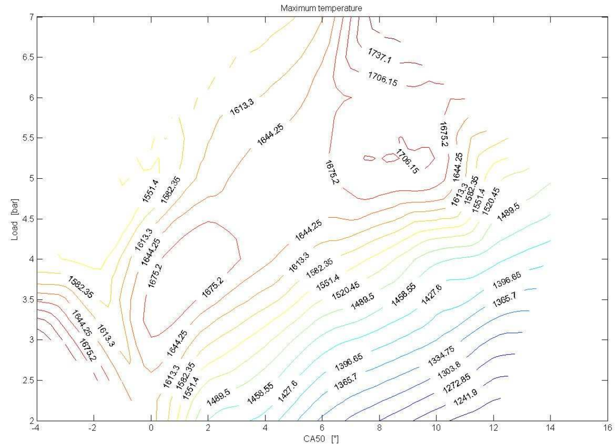

Lambda
Load [bar]
CA50 [°]

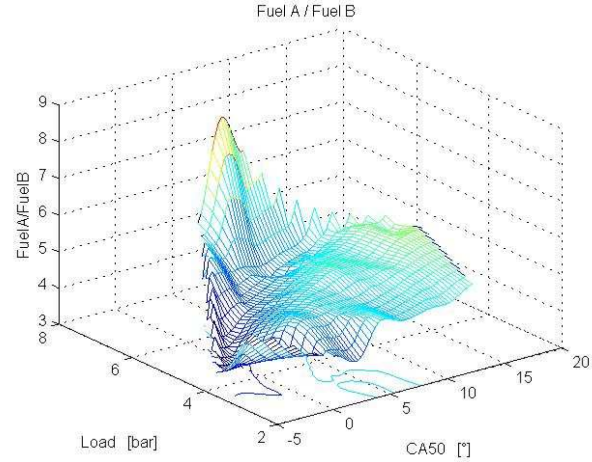
800 rpm

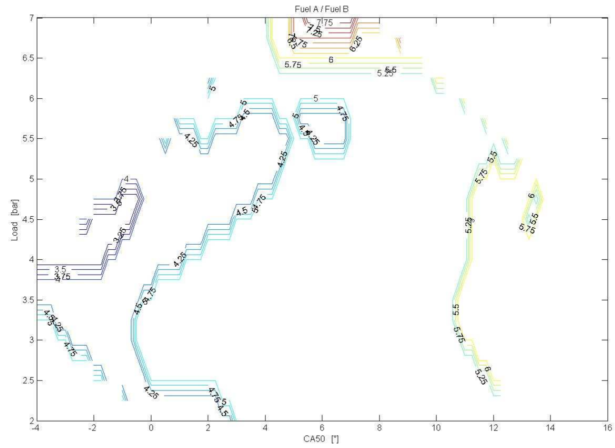

Comb eff
Comb eff [%]
Load [bar]
CA50 [°]
BSFC
BSFC [g/kWh]
BTE
BTE [%]
Gas exchange efficiency
nge [%]

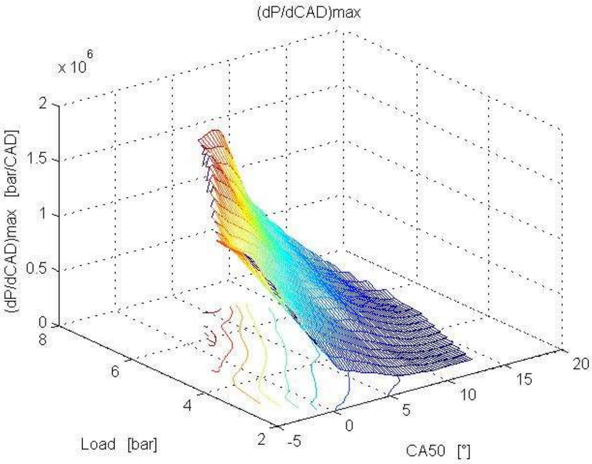
800 rpm

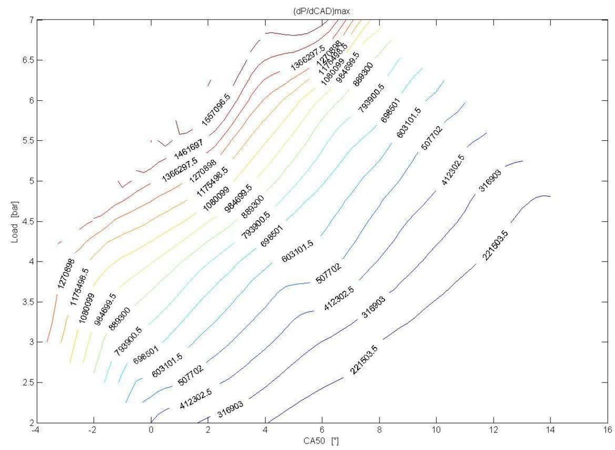

Maximum temperature
Tmax [°K]
Load [bar]
CA50 [°]

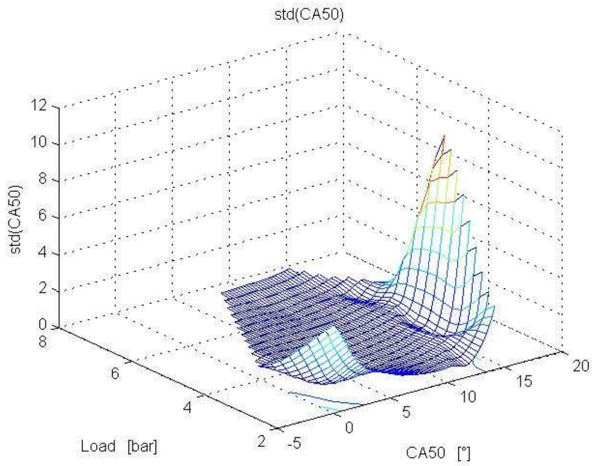

Maximum temperature
Load [bar]
CA50 [°]

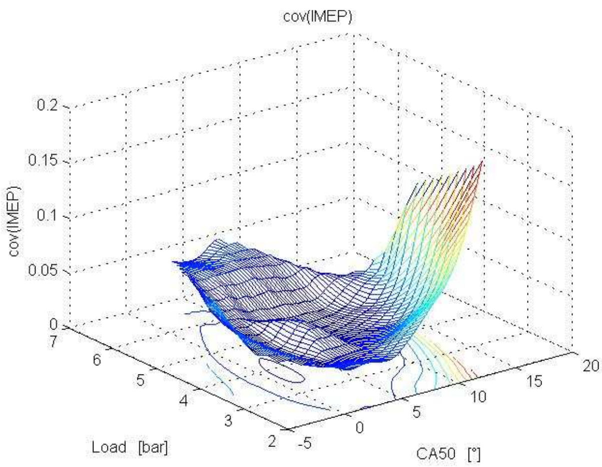

Fuel A / Fuel B
FuelA/FuelB
Load [bar]
CA50 [°]


Fuel A / Fuel B
Load [bar]
CA50 [°]

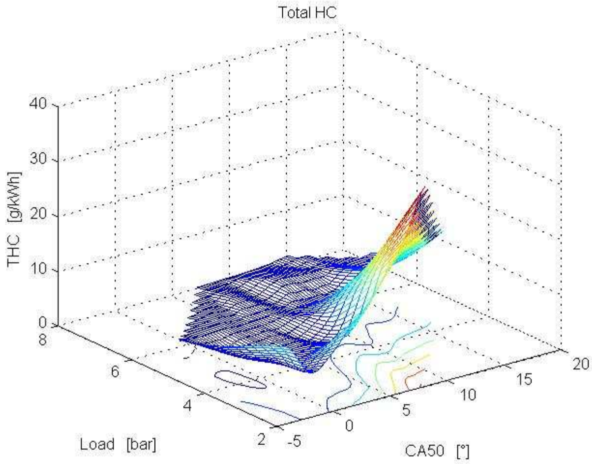

(dP/dCAD)max
x 10^6
(dP/dCAD)max [bar/CAD]
Load [bar]
CA50 [°]

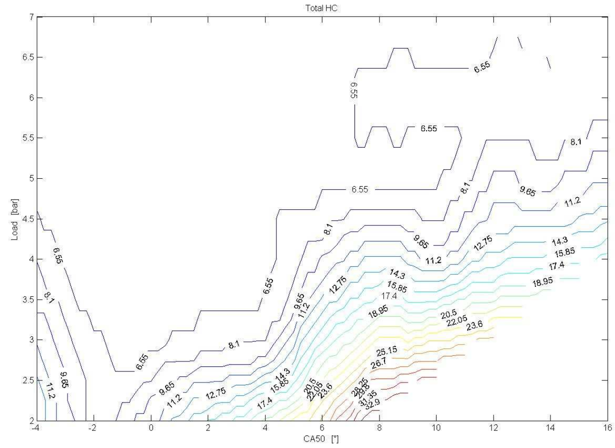

(dP/dCAD)max
Load [bar]
CA50 [°]

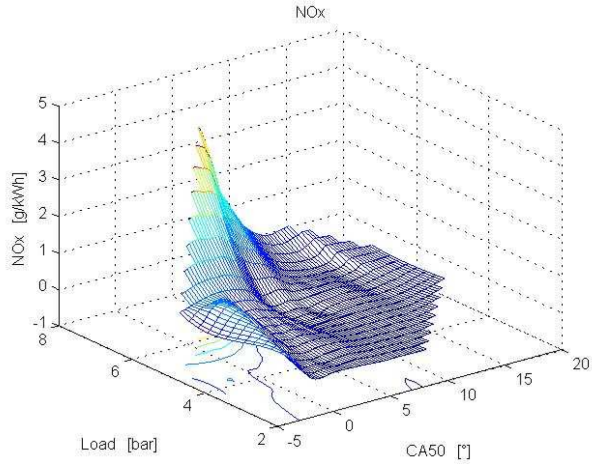

std(CA50)
std(CA50)
Load [bar]
CA50 [°]

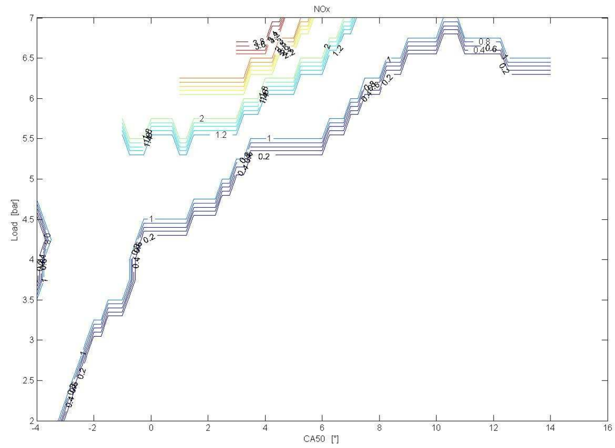

cov(IMEP)
cov(IMEP)
Load [bar]
CA50 [°]

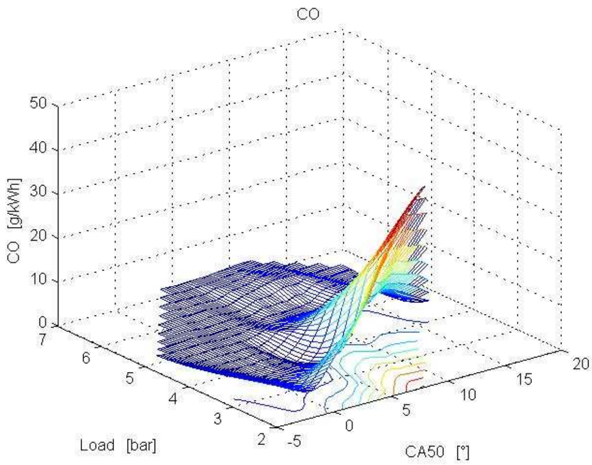
1000 rpm

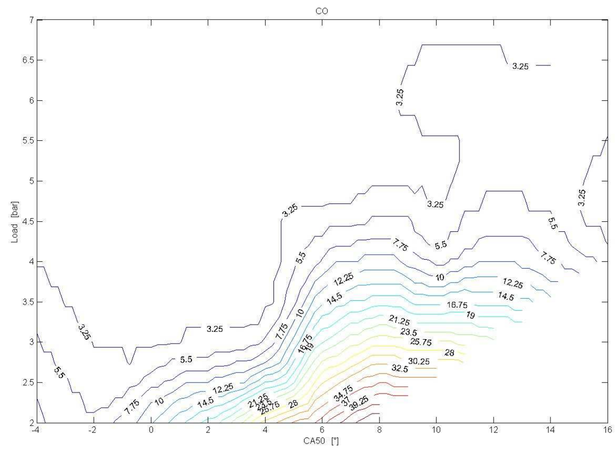

Total HC
THC [g/kWh]
Load [bar]
CA50 [°]

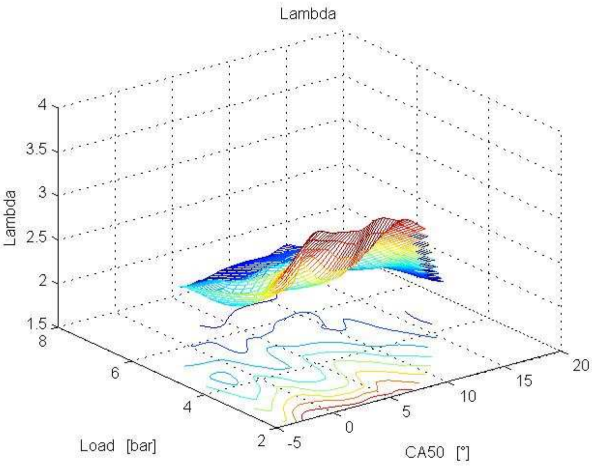

Total HC
Load [bar]
CA50 [°]

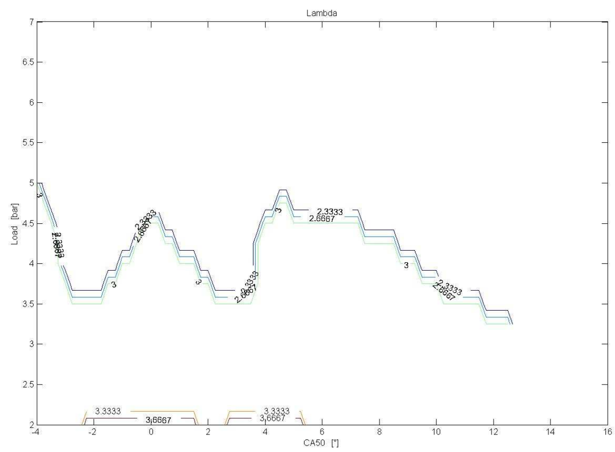

NOx
NOx [g/kWh]
Load [bar]
CA50 [°]


NOx
Load [bar]
CA50 [°]

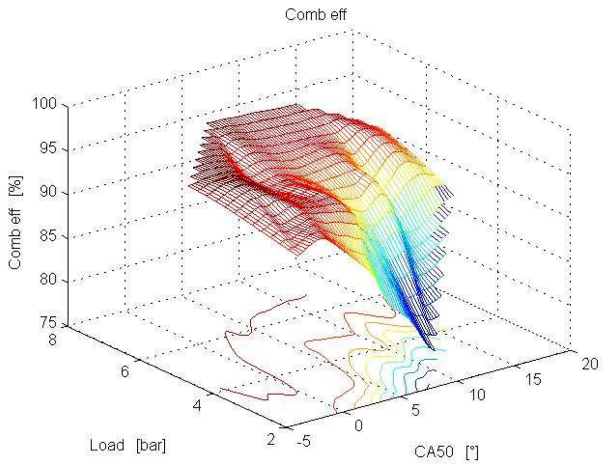

CO
CO [g/kWh]
Load [bar]
CA50 [°]

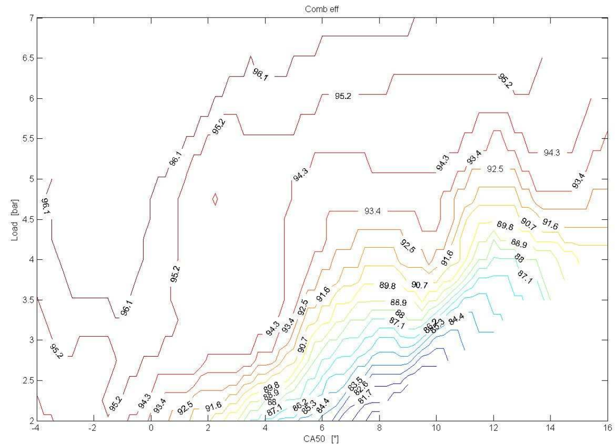

CO
Load [bar]
CA50 [°]

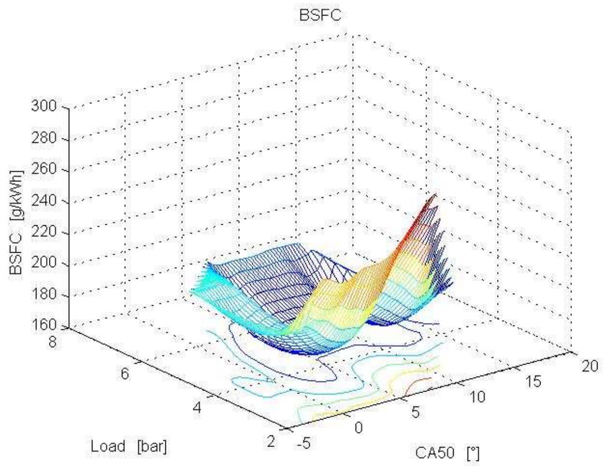

Lambda
Lambda
Load [bar]
CA50 [°]

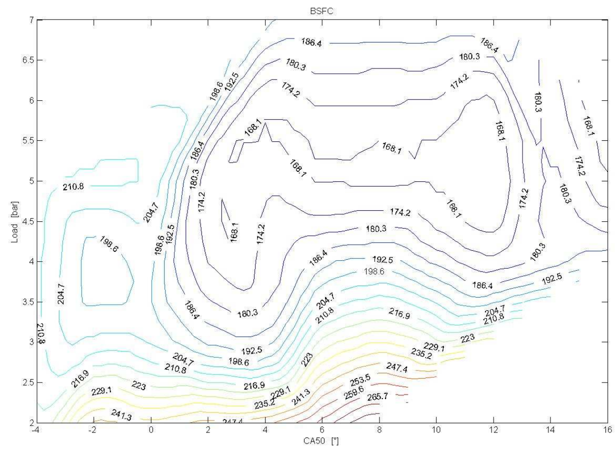

Lambda
Load [bar]
CA50 [°]

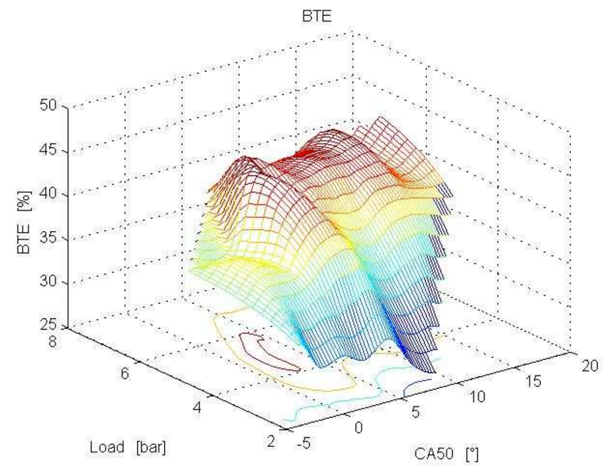
1000 rpm

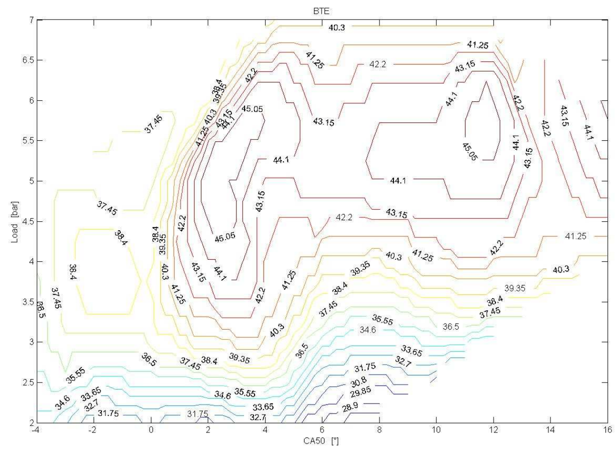

Comb eff
Comb eff [%]
Load [bar]
CA50 [°]

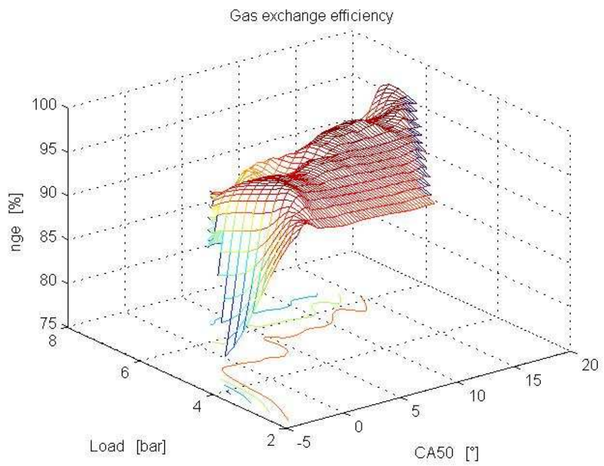

Comb eff
Load [bar]
CA50 [°]

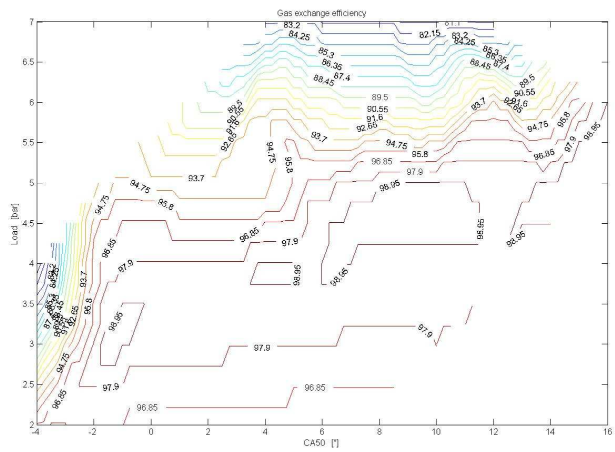

BSFC
BSFC [g/kWh]
Load [bar]
CA50 [°]


BSFC
Load [bar]
CA50 [°]

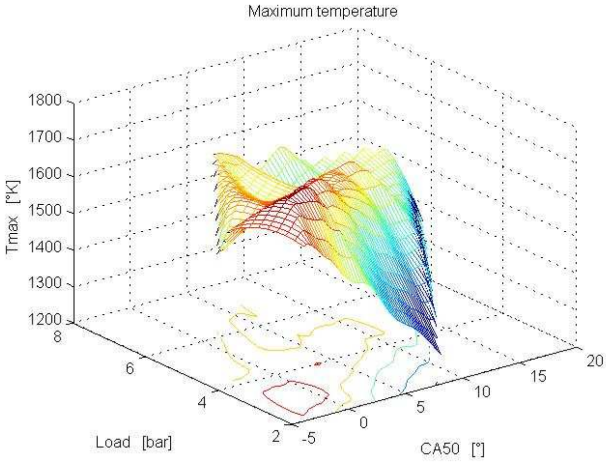

BTE
BTE [%]
Load [bar]
CA50 [°]

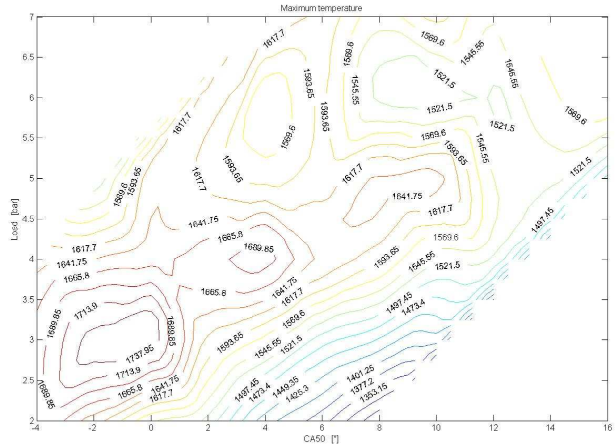

BTE
Load [bar]
CA50 [°]

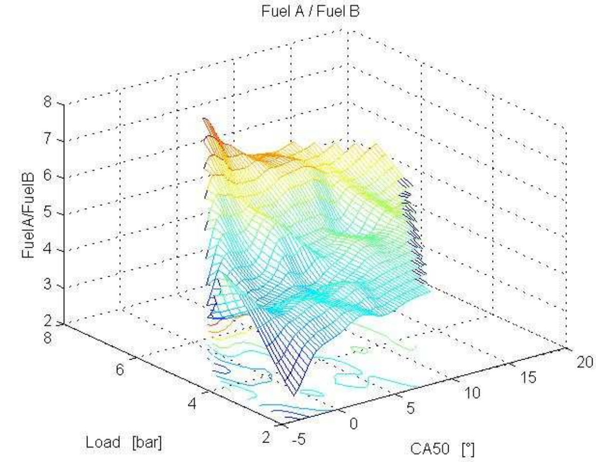

Gas exchange efficiency
nge [%]
Load [bar]
CA50 [°]

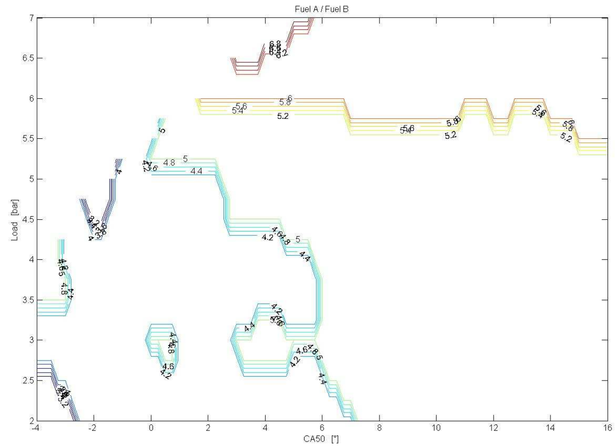

Gas exchange efficiency
Load [bar]
CA50 [°]

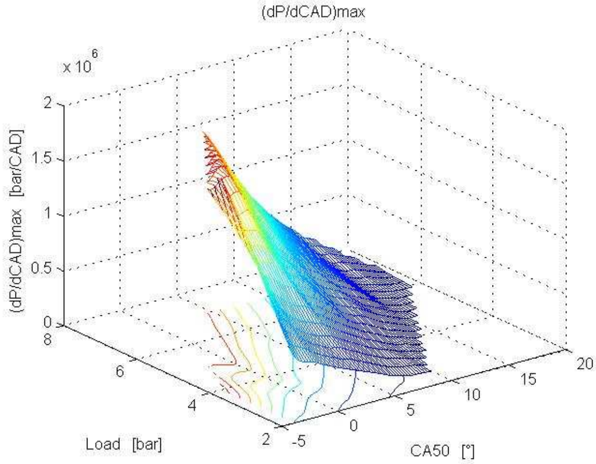
1000 rpm

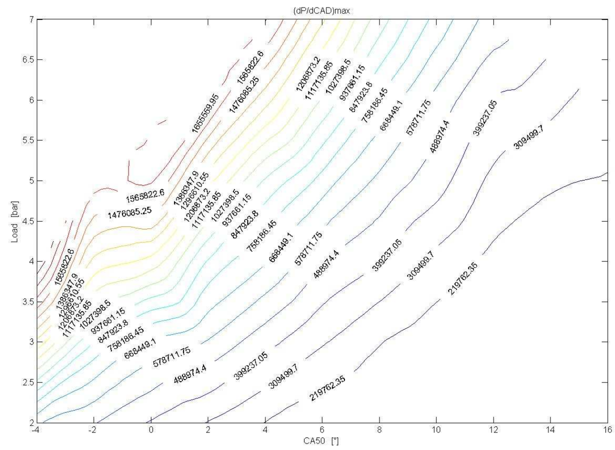

Maximum temperature
Tmax [°K]
Load [bar]
CA50 [°]

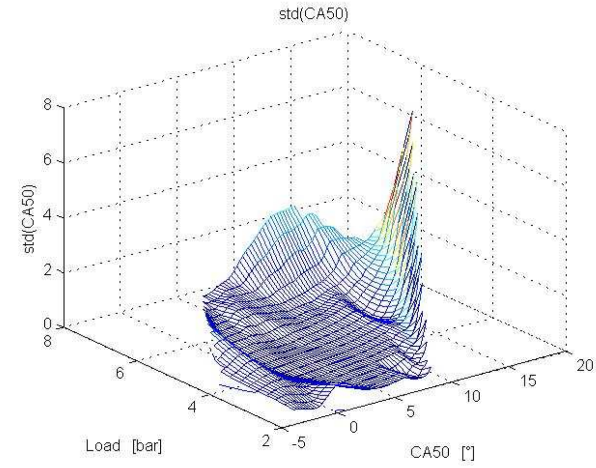

Maximum temperature
Load [bar]
CA50 [°]

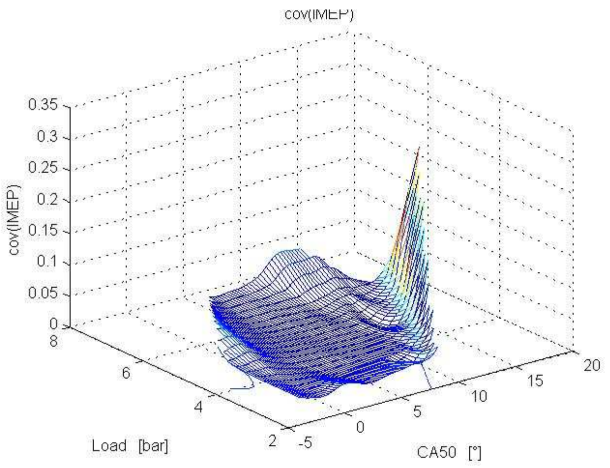

Fuel A / Fuel B
FuelA/FuelB
Load [bar]
CA50 [°]


Fuel A / Fuel B
Load [bar]
CA50 [°]

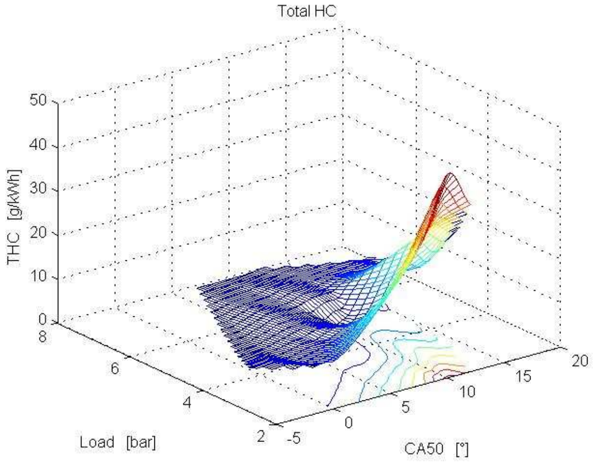

(dP/dCAD)max
x 10^6
(dP/dCAD)max [bar/CAD]
Load [bar]
CA50 [°]

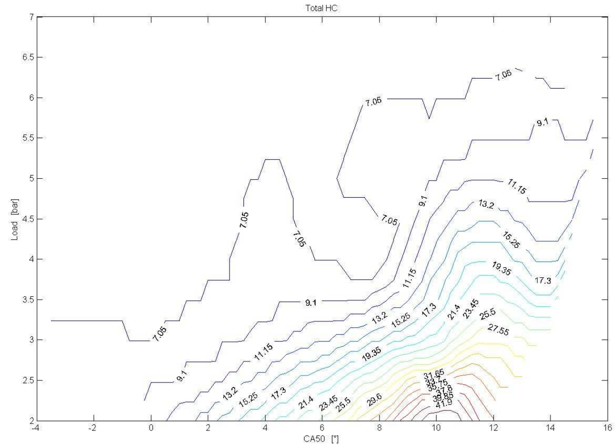

(dP/dCAD)max
Load [bar]
CA50 [°]

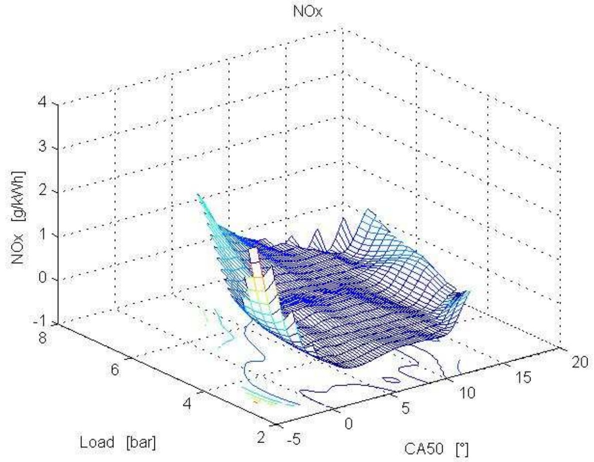

std(CA50)
std(CA50)
Load [bar]
CA50 [°]

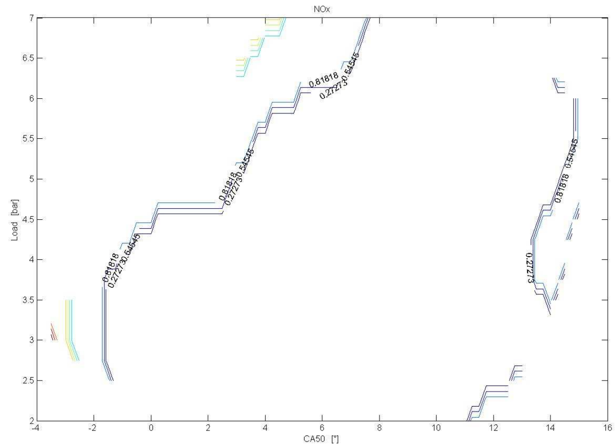

cov(IMEP)
cov(IMEP)
Load [bar]
CA50 [°]

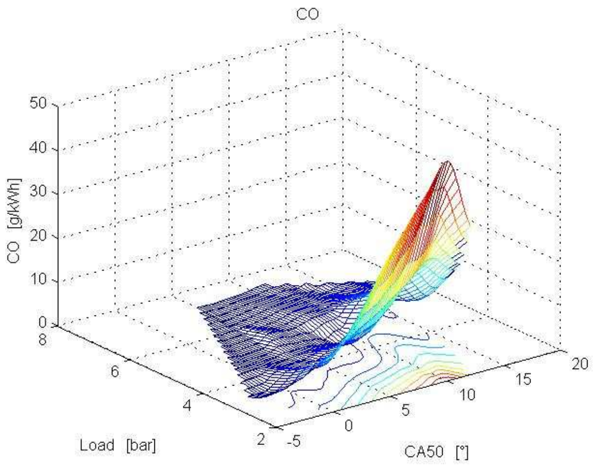
1200 rpm

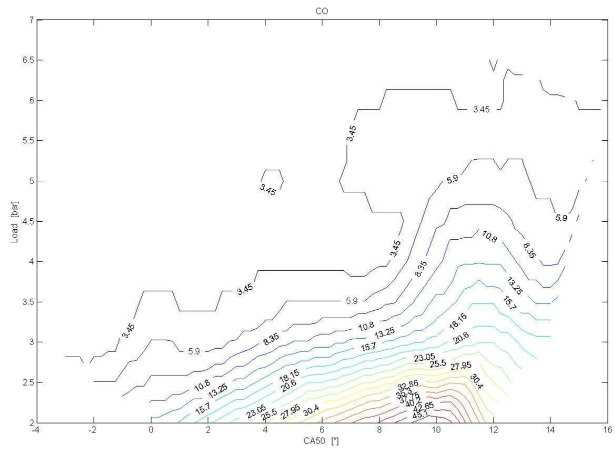

Total HC
THC [g/kWh]
Load [bar]
CA50 [°]

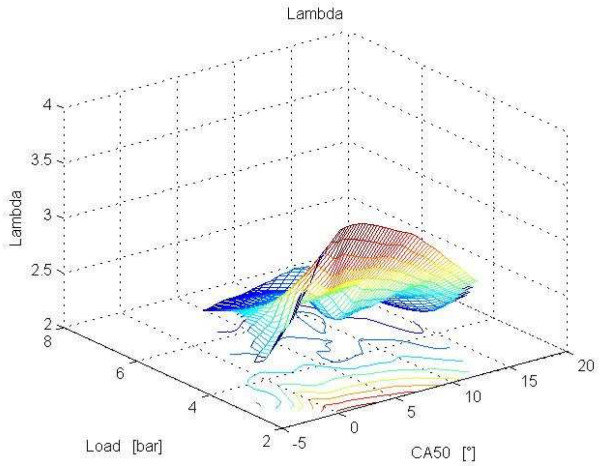

Total HC
Load [bar]
CA50 [°]

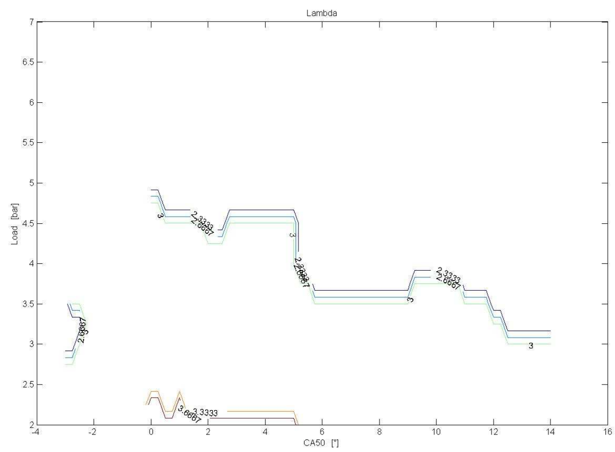

NOx
NOx [g/kWh]
Load [bar]
CA50 [°]


NOx
Load [bar]
CA50 [°]

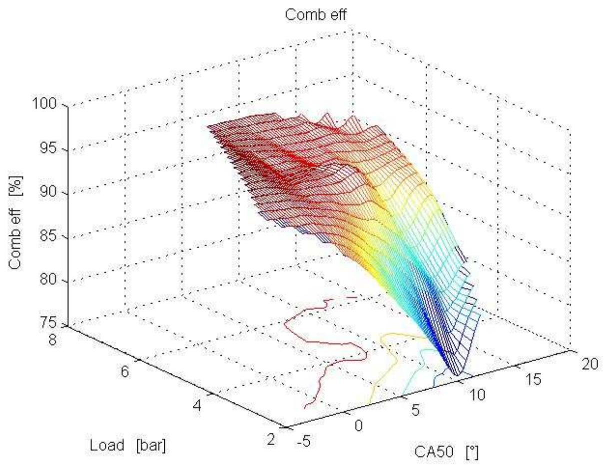

CO
CO [g/kWh]
Load [bar]
CA50 [°]

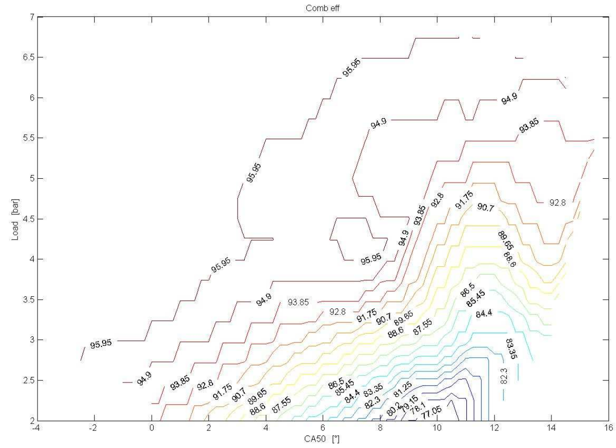

CO
Load [bar]
CA50 [°]

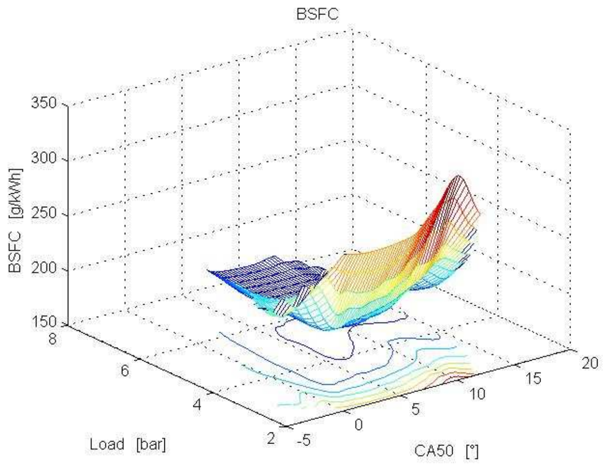

Lambda
Lambda
Load [bar]
CA50 [°]

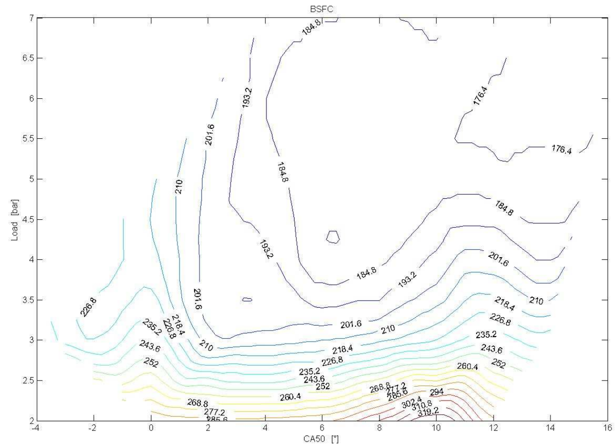

Lambda
Load [bar]
CA50 [°]

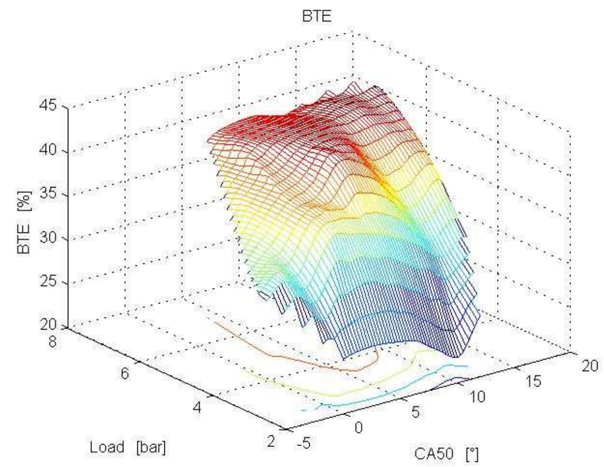
1200 rpm

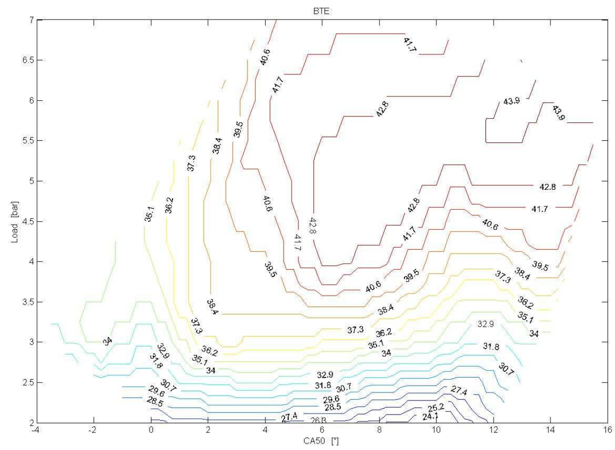

Comb eff
Comb eff [%]
Load [bar]
CA50 [°]

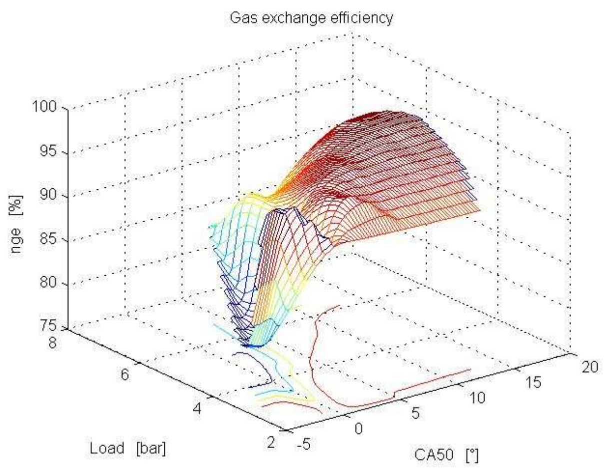

Comb eff
Load [bar]
CA50 [°]

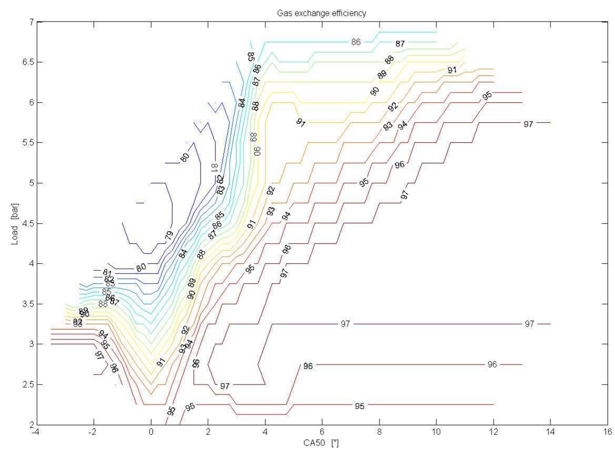

BSFC
BSFC [g/kWh]
Load [bar]
CA50 [°]


BSFC
Load [bar]
CA50 [°]

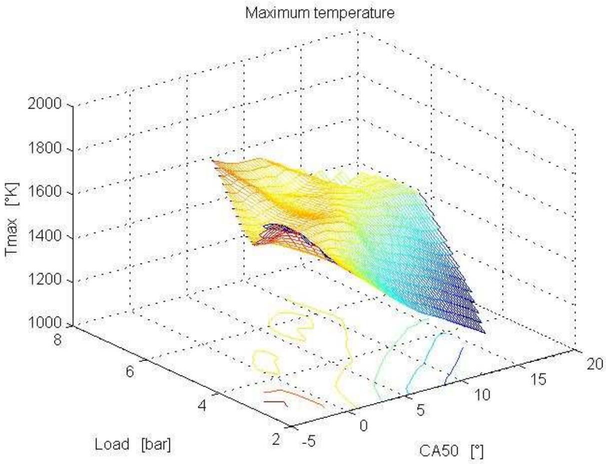

BTE
BTE [%]
Load [bar]
CA50 [°]

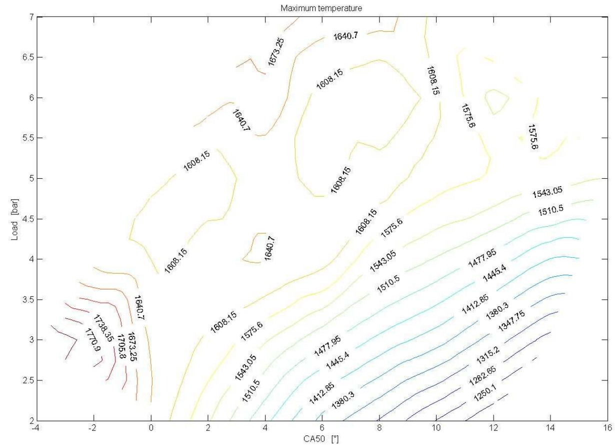

BTE
Load [bar]
CA50 [°]

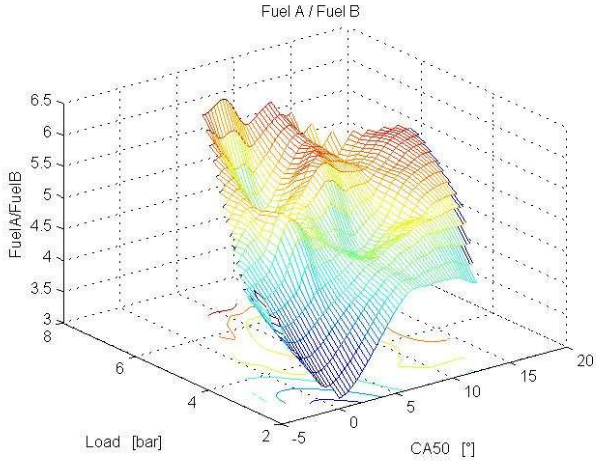

Gas exchange efficiency
nge [%]
Load [bar]
CA50 [°]

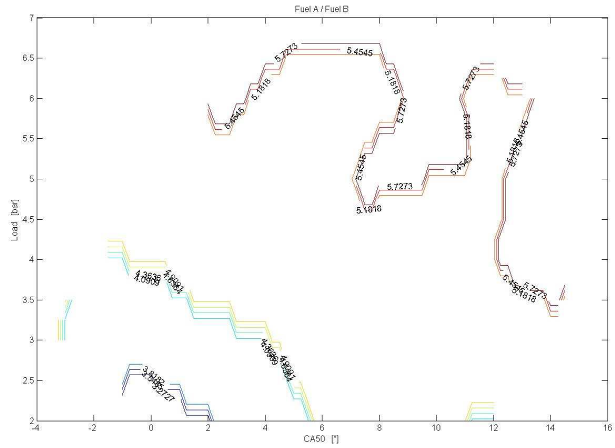

Gas exchange efficiency
Load [bar]
CA50 [°]

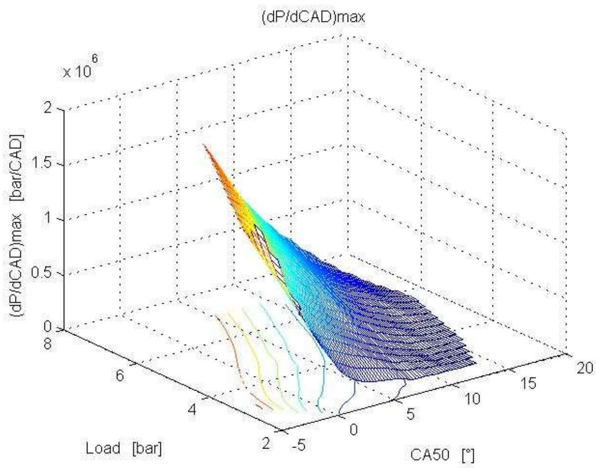
1200 rpm

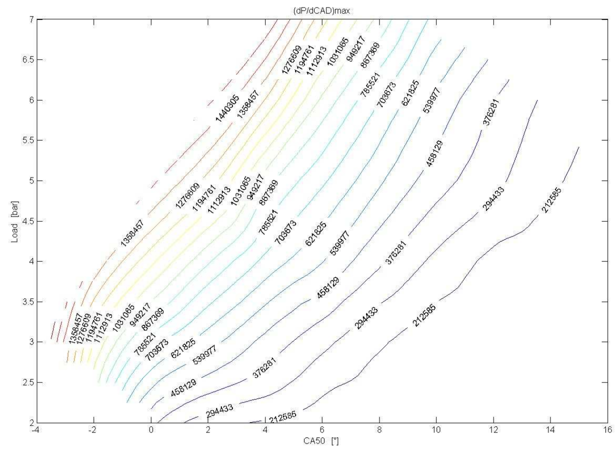

Maximum temperature
Tmax [°K]
Load [bar]
CA50 [°]

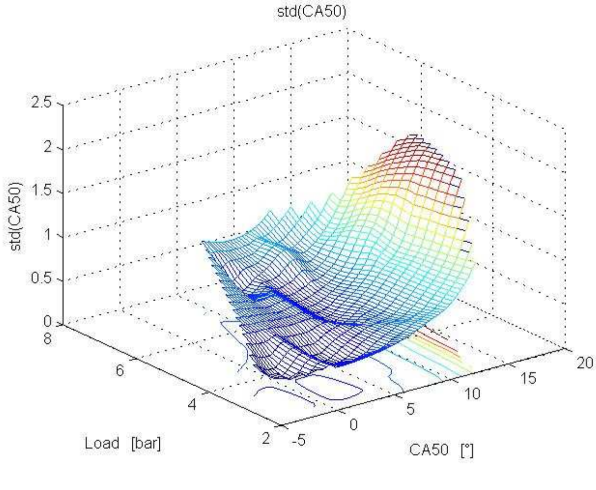

Maximum temperature
Load [bar]
CA50 [°]

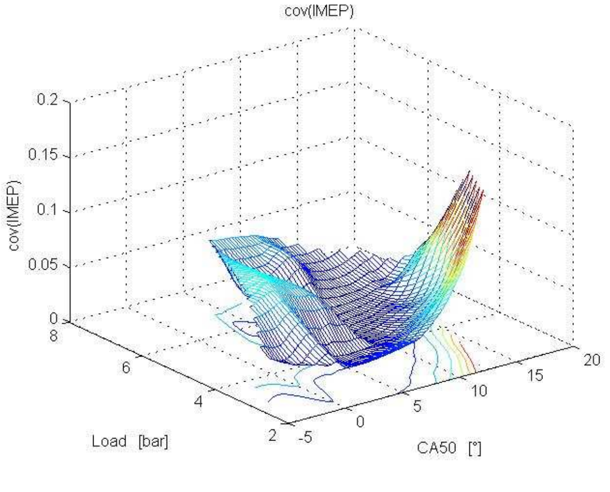

Fuel A / Fuel B
FuelA/FuelB
Load [bar]
CA50 [°]


Fuel A / Fuel B
Load [bar]
CA50 [°]

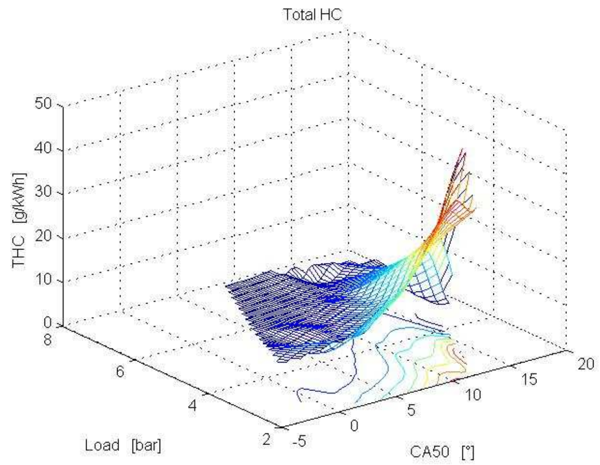

(dP/dCAD)max
x 10^6
(dP/dCAD)max [bar/CAD]
Load [bar]
CA50 [°]

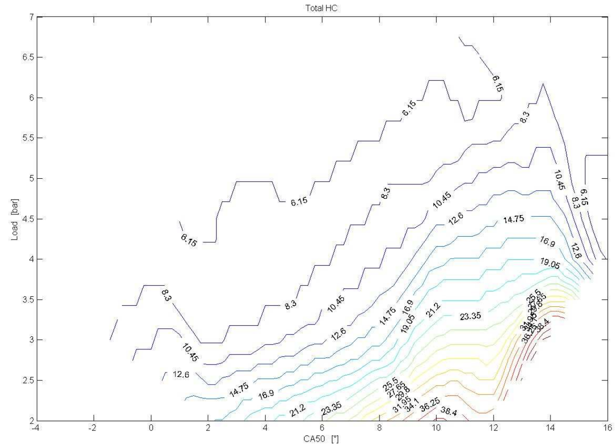

(dP/dCAD)max
Load [bar]
CA50 [°]

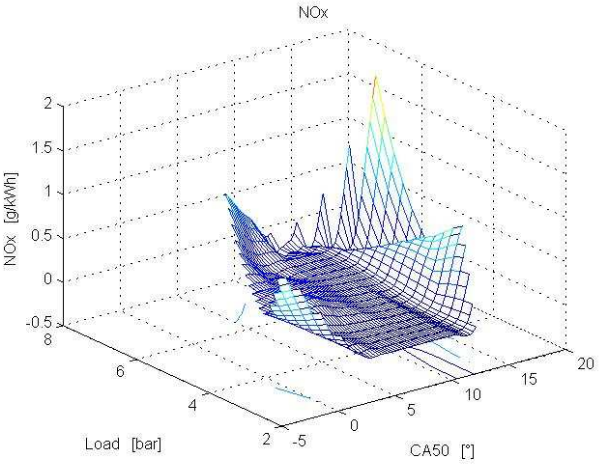

std(CA50)
std(CA50)
Load [bar]
CA50 [°]

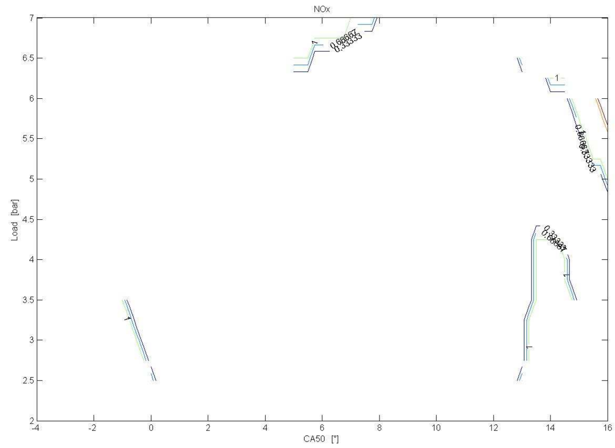

cov(IMEP)
cov(IMEP)
Load [bar]
CA50 [°]

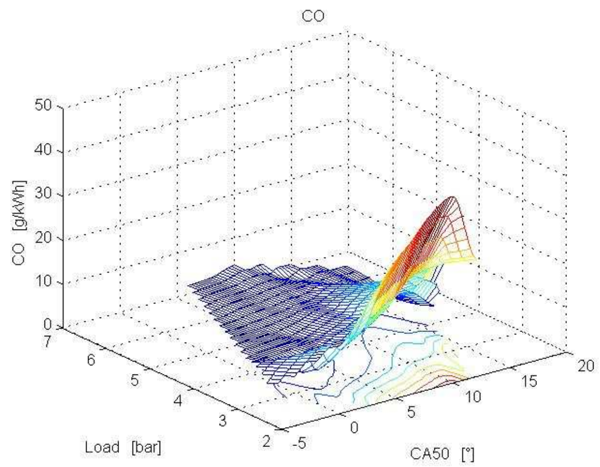
1400 rpm

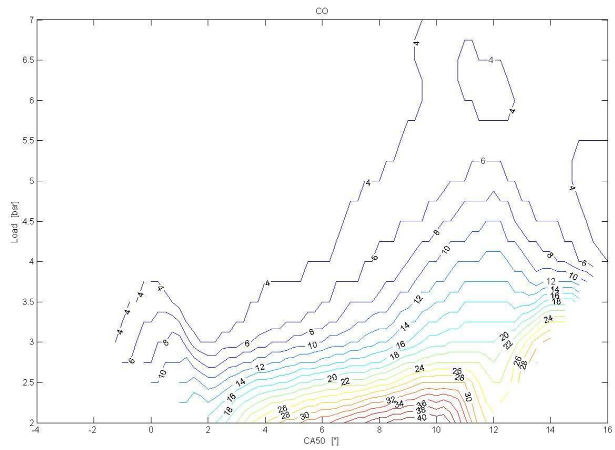

Total HC
THC [g/kWh]
Load [bar]
CA50 [°]

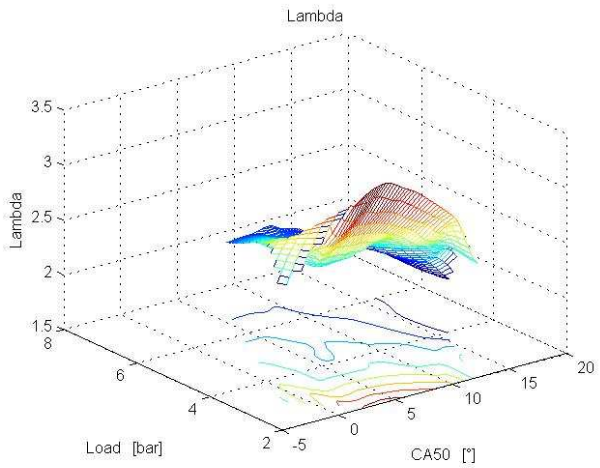

Total HC
Load [bar]
CA50 [°]

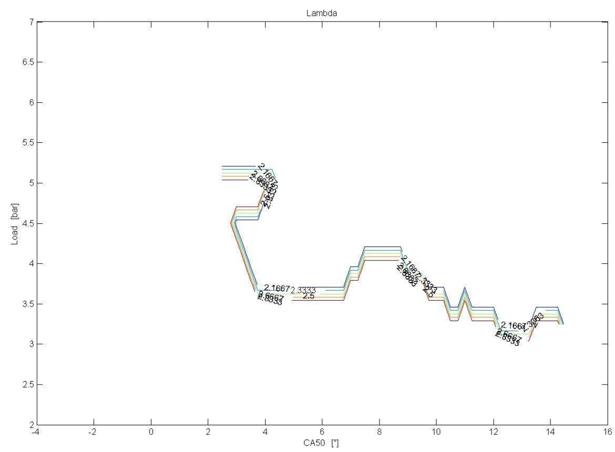

NOx
NOx [g/kWh]
Load [bar]
CA50 [°]


NOx
Load [bar]
CA50 [°]

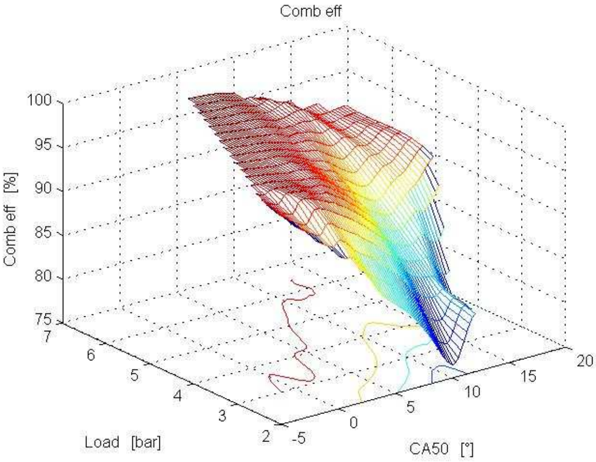

CO
CO [g/kWh]
Load [bar]
CA50 [°]

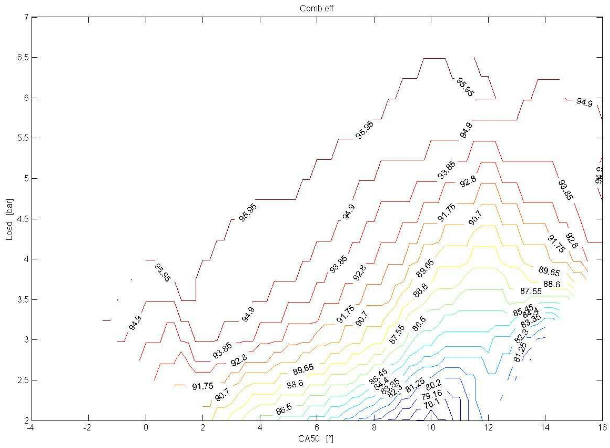

CO
Load [bar]
CA50 [°]

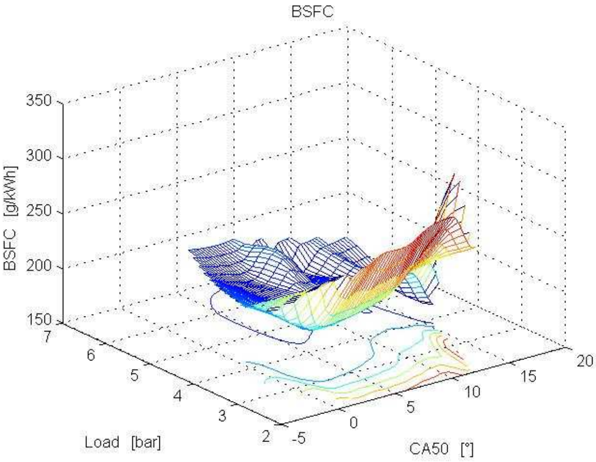

Lambda
Lambda
Load [bar]
CA50 [°]

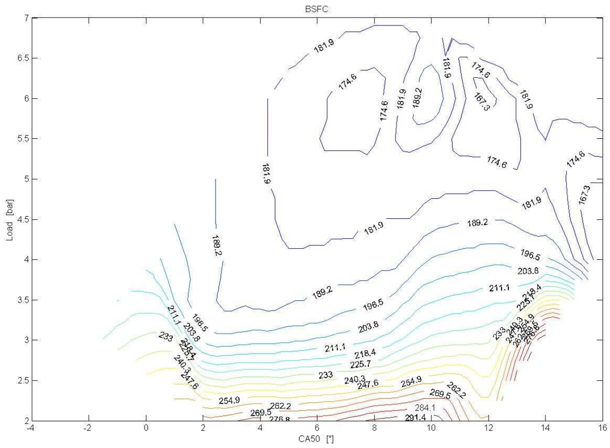

Lambda
Load [bar]
CA50 [°]

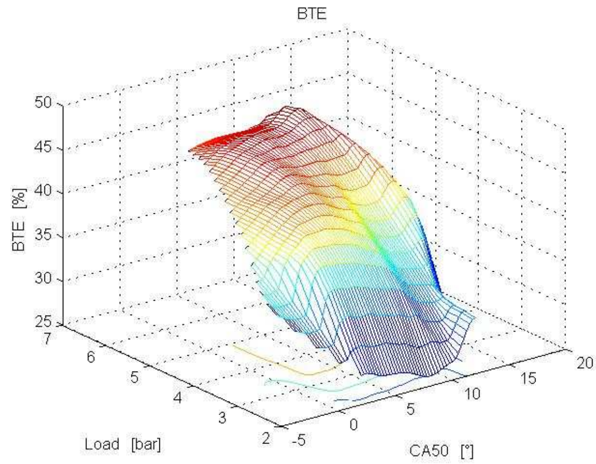
1400 rpm

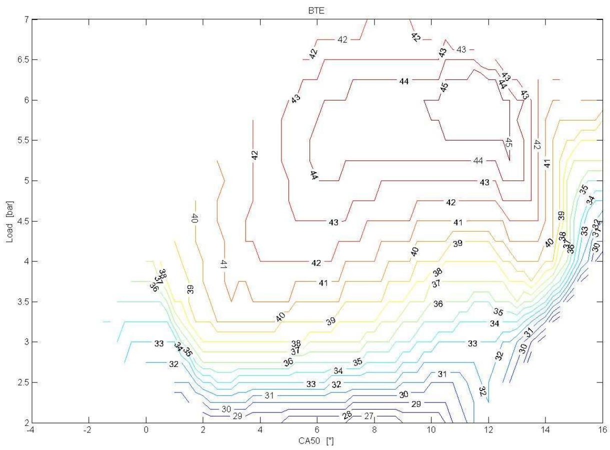

Comb eff
Comb eff [%]
Load [bar]
CA50 [°]

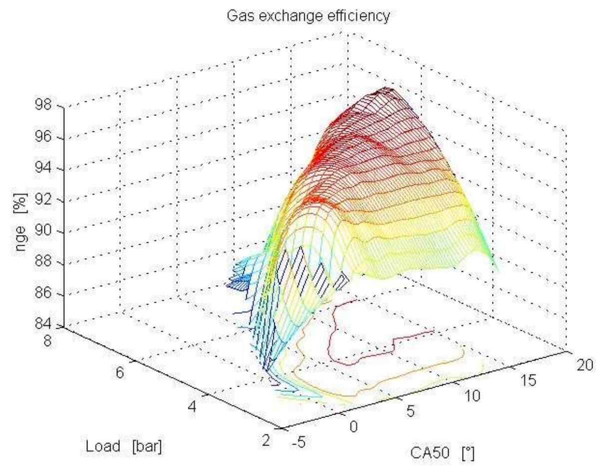

Comb eff
Load [bar]
CA50 [°]

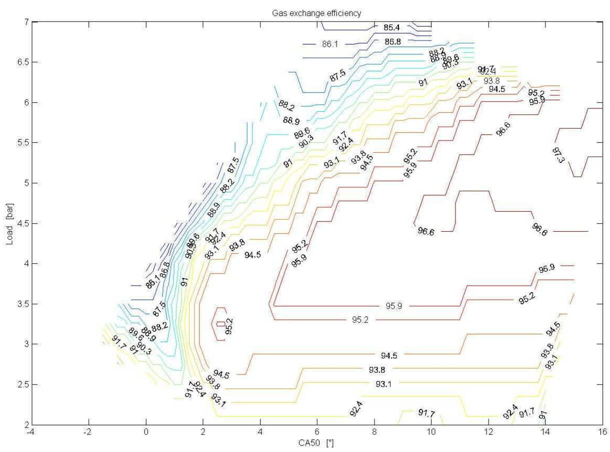

BSFC
BSFC [g/kWh]
Load [bar]
CA50 [°]


BSFC
Load [bar]
CA50 [°]

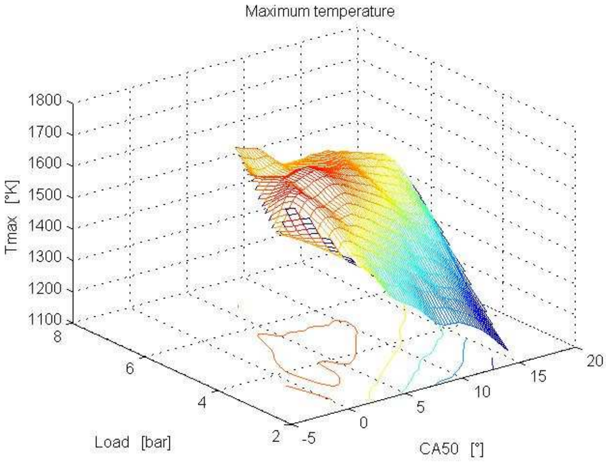

BTE
BTE [%]
Load [bar]
CA50 [°]

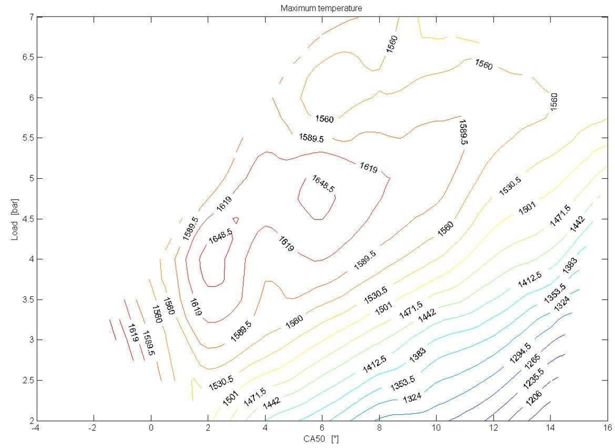

BTE
Load [bar]
CA50 [°]

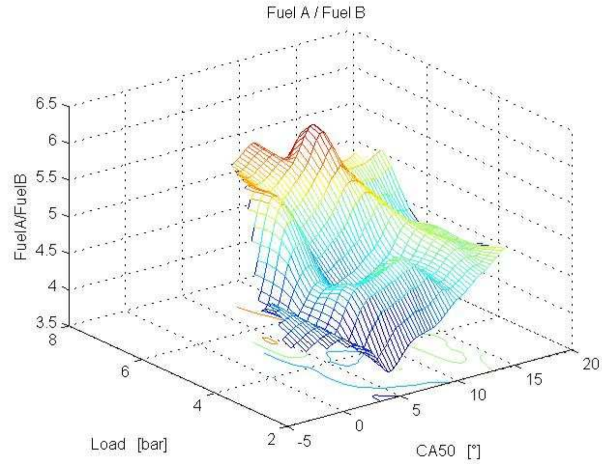

Gas exchange efficiency
nge [%]
Load [bar]
CA50 [°]

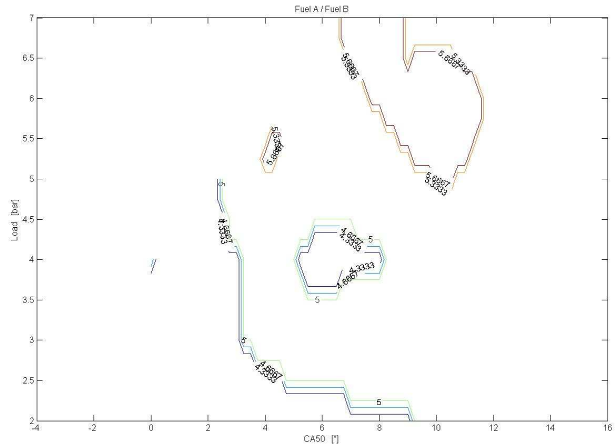

Gas exchange efficiency
Load [bar]
CA50 [°]

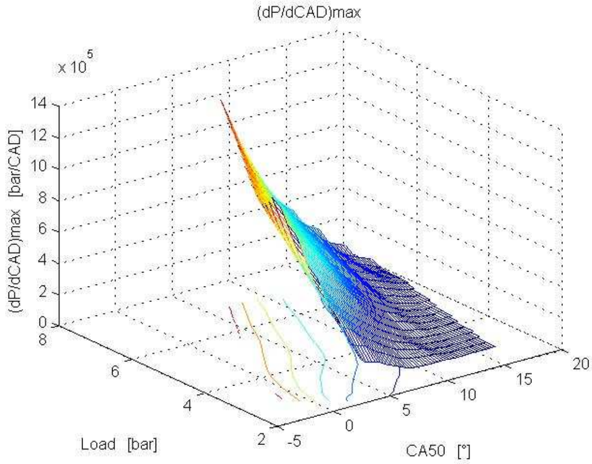
1400 rpm

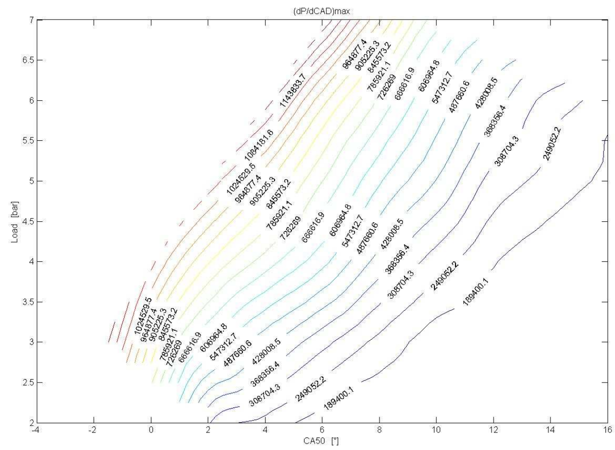

Maximum temperature
Tmax [°K]
Load [bar]
CA50 [°]

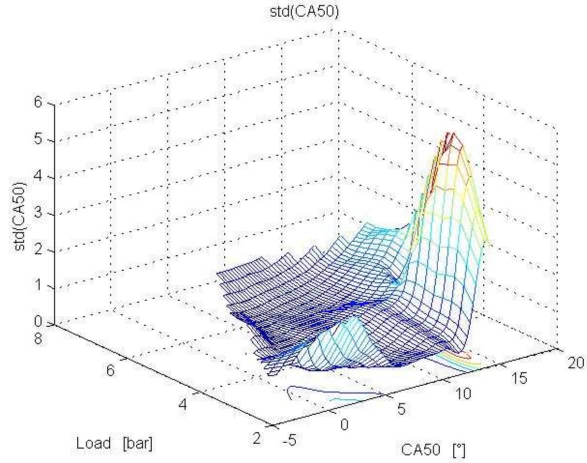

Maximum temperature
Load [bar]
CA50 [°]

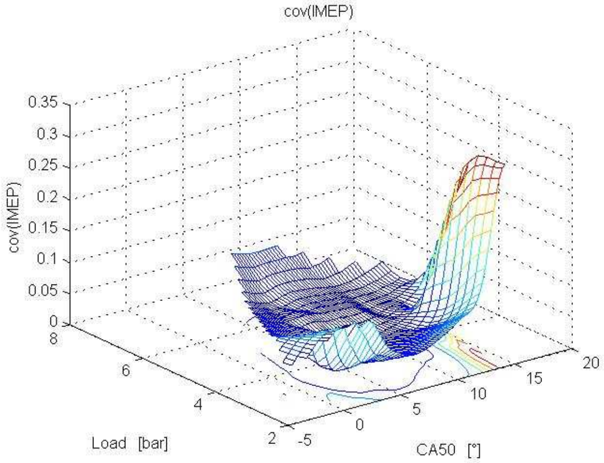

Fuel A / Fuel B
FuelA/FuelB
Load [bar]
CA50 [°]


Fuel A / Fuel B
Load [bar]
CA50 [°]

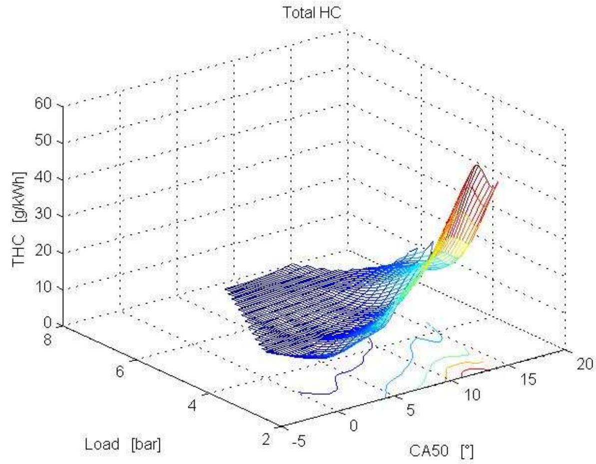

(dP/dCAD)max
x 10^5
(dP/dCAD)max [bar/CAD]
Load [bar]
CA50 [°]

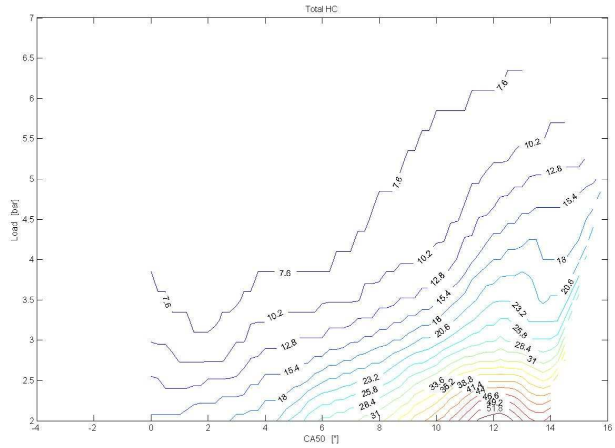

(dP/dCAD)max
Load [bar]
CA50 [°]

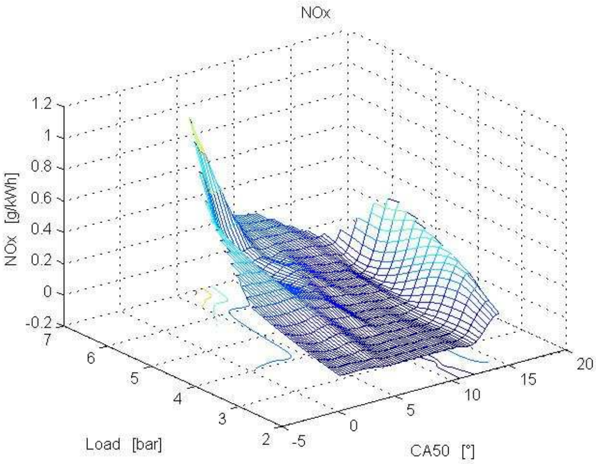

std(CA50)
std(CA50)
Load [bar]
CA50 [°]

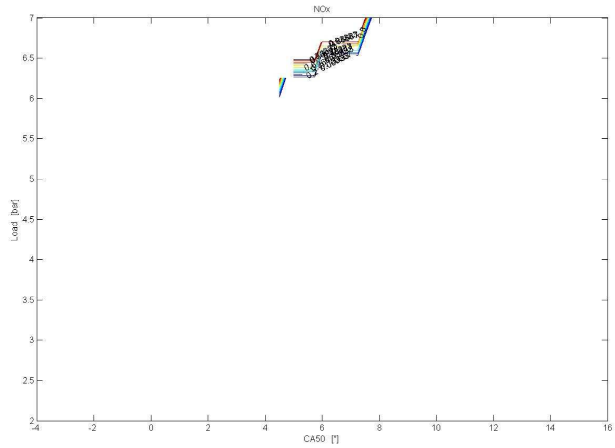

cov(IMEP)
cov(IMEP)
Load [bar]
CA50 [°]

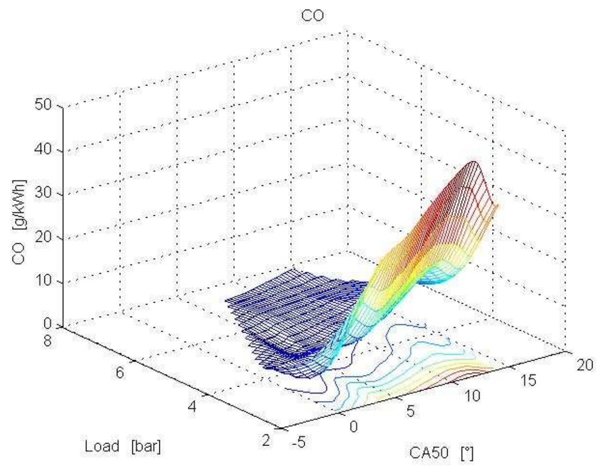
1600 rpm

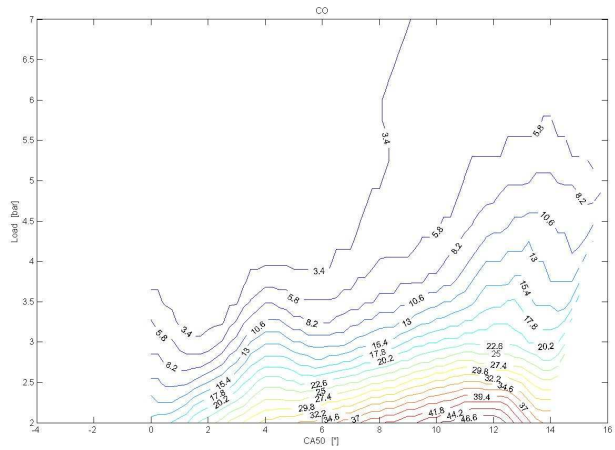

Total HC
THC [g/kWh]
Load [bar]
CA50 [°]

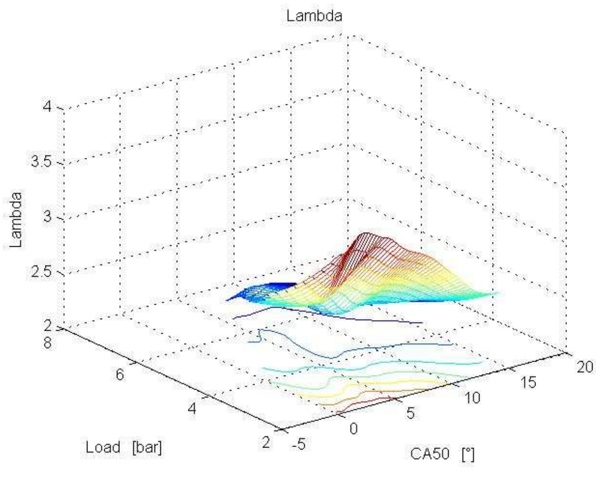

Total HC
Load [bar]
CA50 [°]

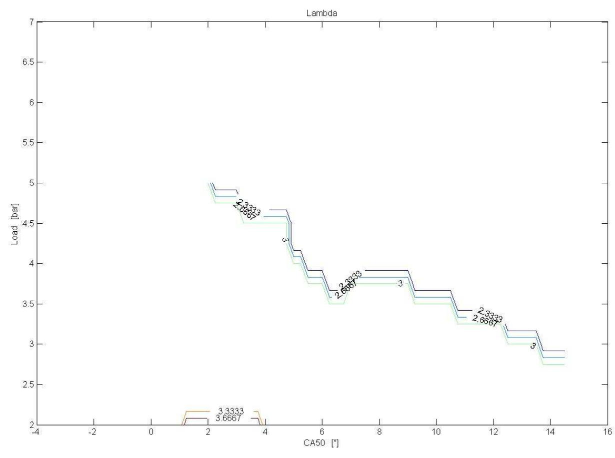

NOx
NOx [g/kWh]
Load [bar]
CA50 [°]


NOx
Load [bar]
CA50 [°]

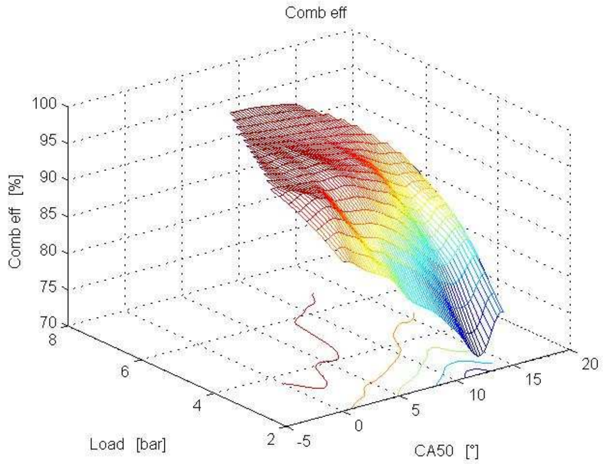

CO
CO [g/kWh]
Load [bar]
CA50 [°]

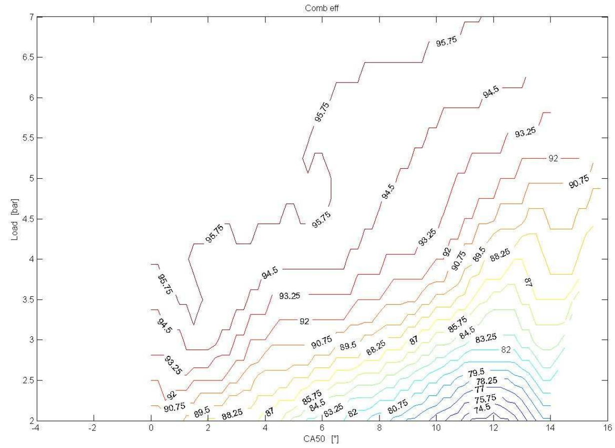

CO
Load [bar]
CA50 [°]

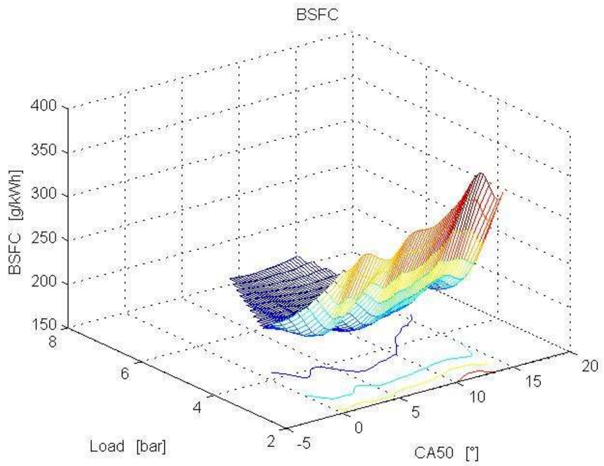

Lambda
Lambda
Load [bar]
CA50 [°]

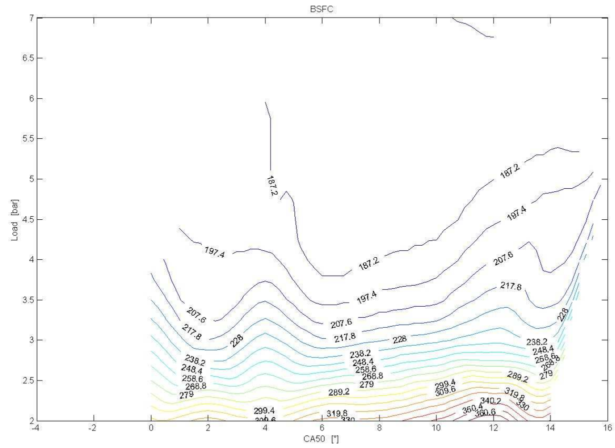

Lambda
Load [bar]
CA50 [°]

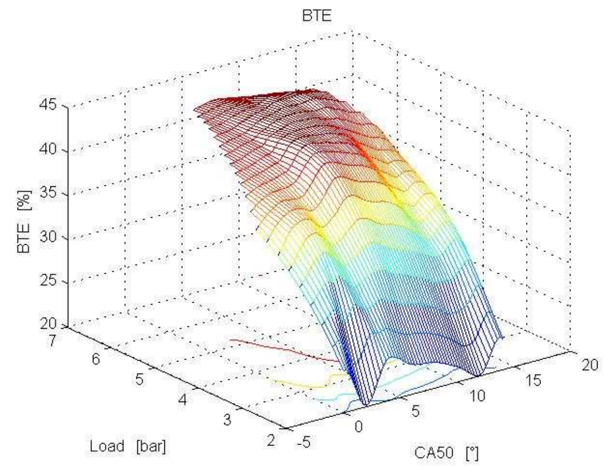
1600 rpm

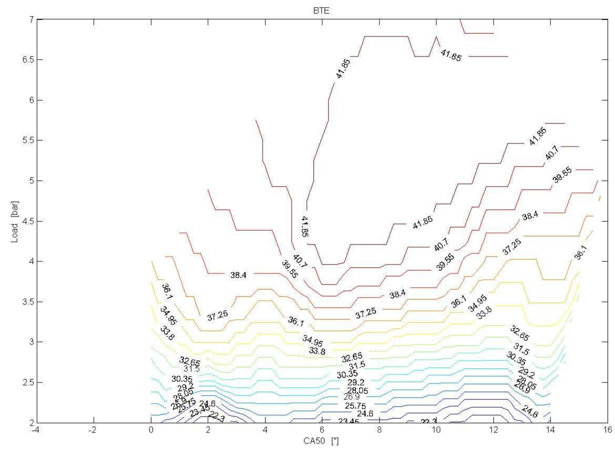

Comb eff
Comb eff [%]
Load [bar]
CA50 [°]

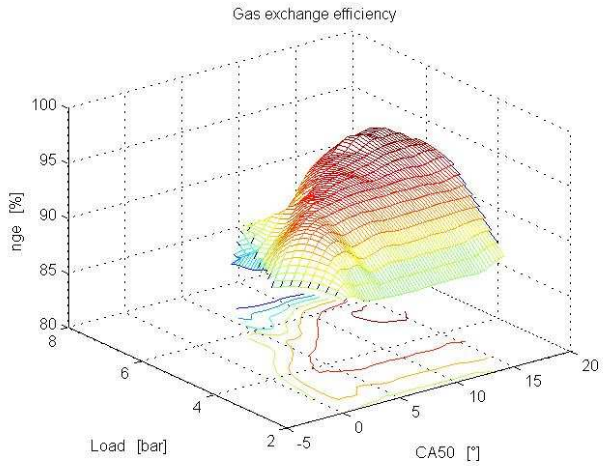

Comb eff
Load [bar]
CA50 [°]

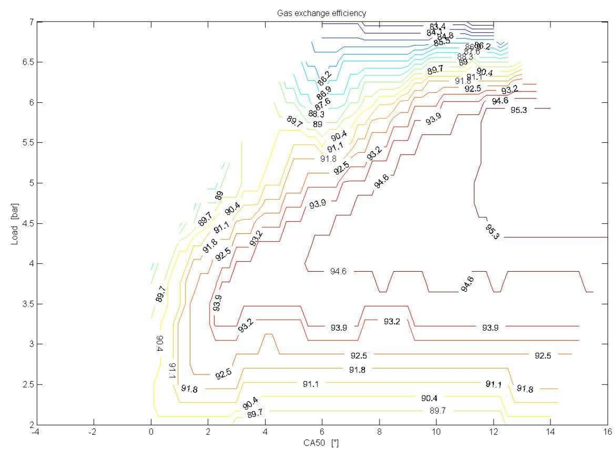

BSFC
BSFC [g/kWh]
Load [bar]
CA50 [°]


BSFC
Load [bar]
CA50 [°]

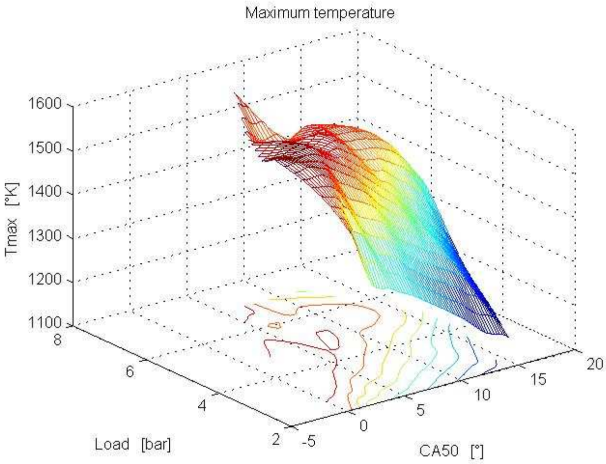

BTE
BTE [%]
Load [bar]
CA50 [°]

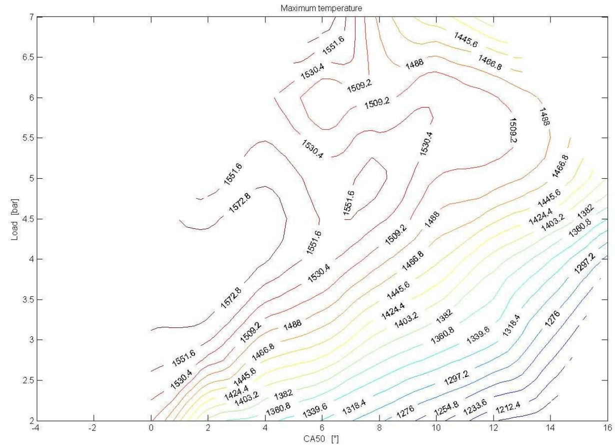

BTE
Load [bar]
CA50 [°]

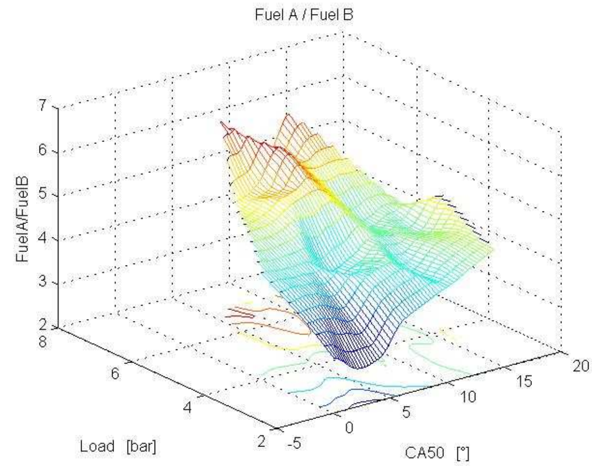

Gas exchange efficiency
nge [%]
Load [bar]
CA50 [°]

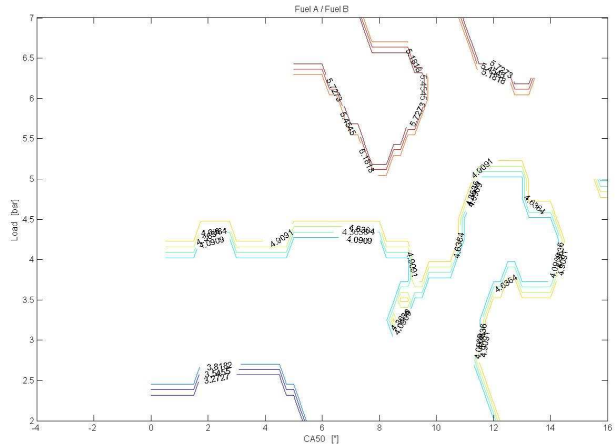

Gas exchange efficiency
Load [bar]
CA50 [°]

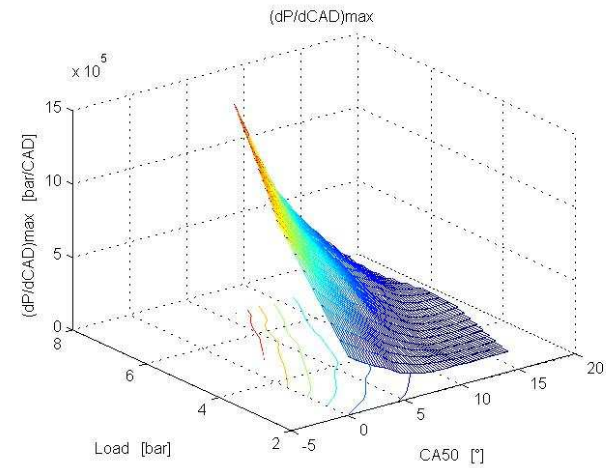
1600 rpm

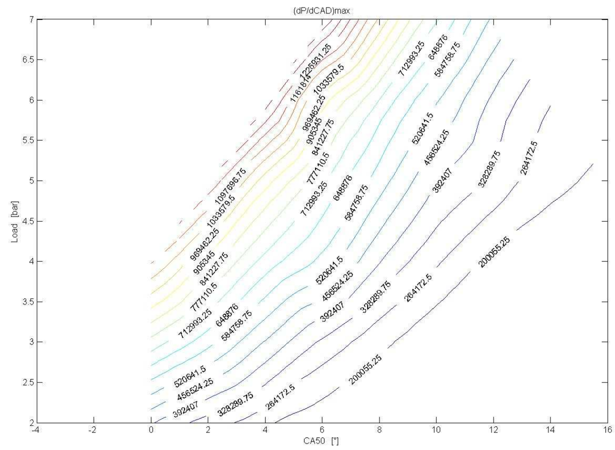

Maximum temperature
Tmax [°K]
Load [bar]
CA50 [°]

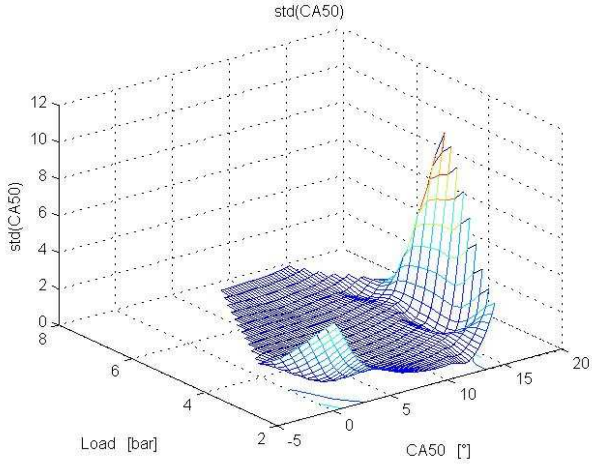

Maximum temperature
Load [bar]
CA50 [°]

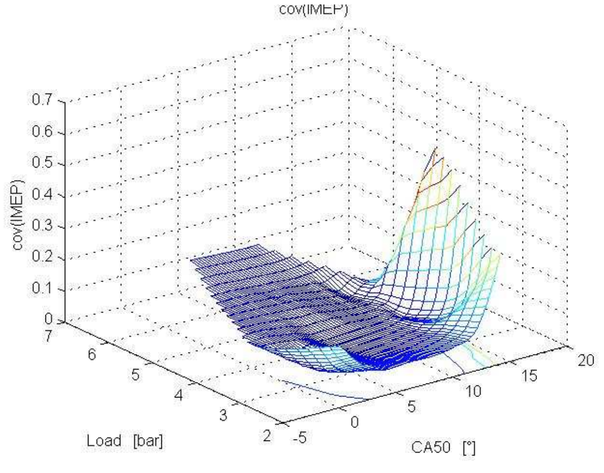

Fuel A / Fuel B
FuelA/FuelB
Load [bar]
CA50 [°]


Fuel A / Fuel B
Load [bar]
CA50 [°]

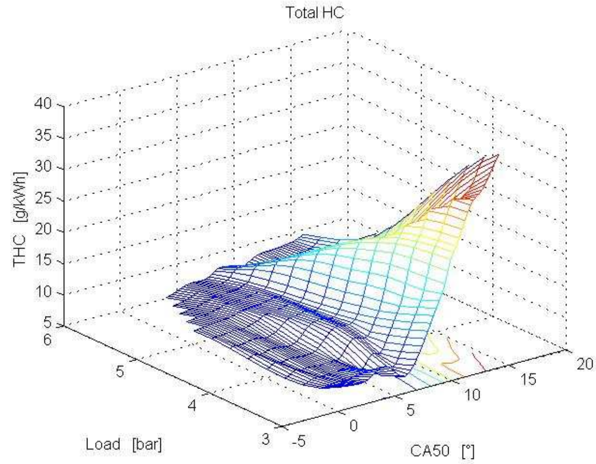

(dP/dCAD)max
x 10^5
(dP/dCAD)max [bar/CAD]
Load [bar]
CA50 [°]

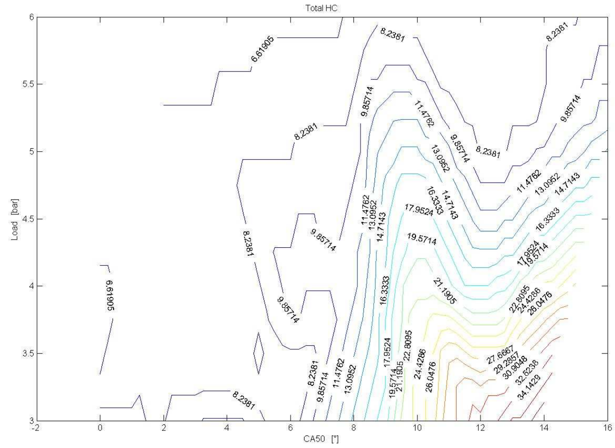

(dP/dCAD)max
Load [bar]
CA50 [°]

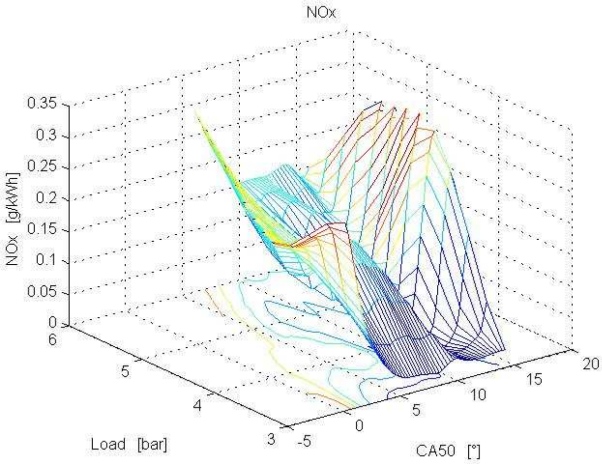

std(CA50)
std(CA50)
Load [bar]
CA50 [°]

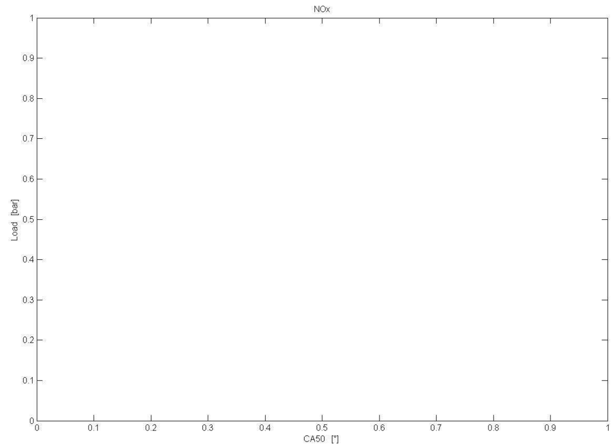

cov(IMEP)
cov(IMEP)
Load [bar]
CA50 [°]

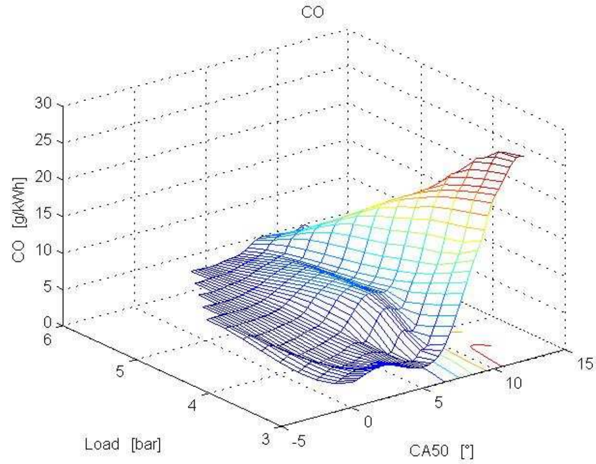
1800 rpm

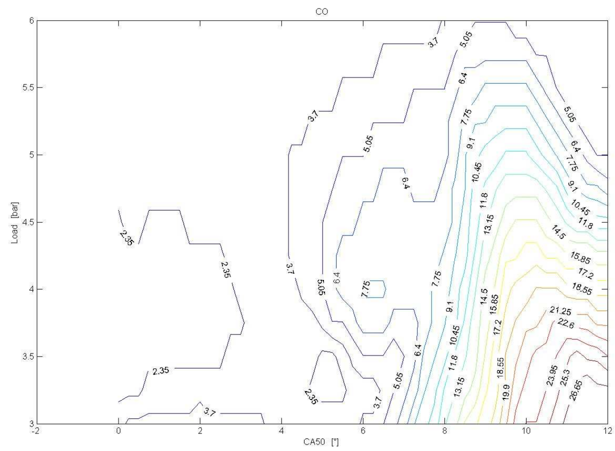

Total HC
THC [g/kWh]
Load [bar]
CA50 [°]

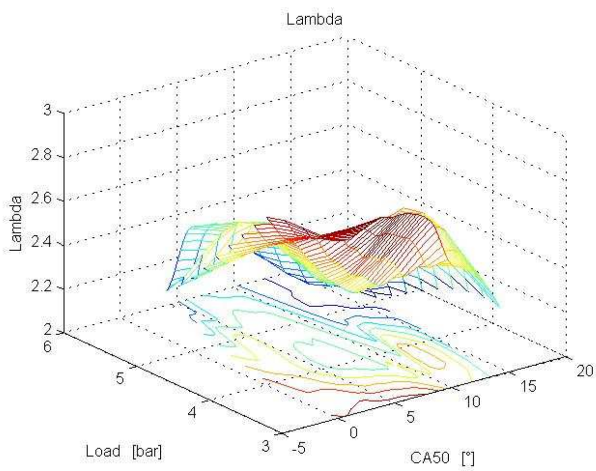

Total HC
Load [bar]
CA50 [°]

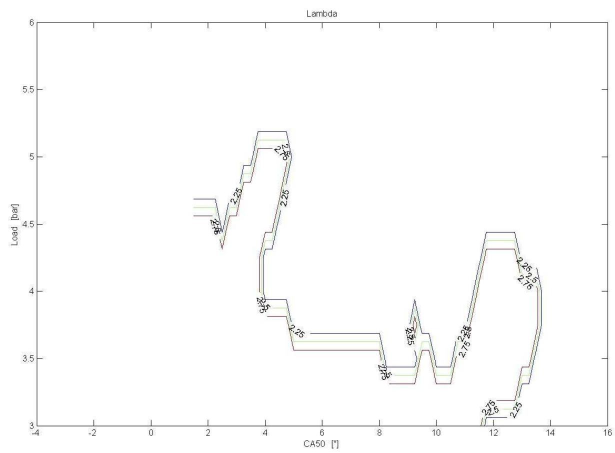

NOx
NOx [g/kWh]
Load [bar]
CA50 [°]


NOx
Load [bar]
CA50 [°]

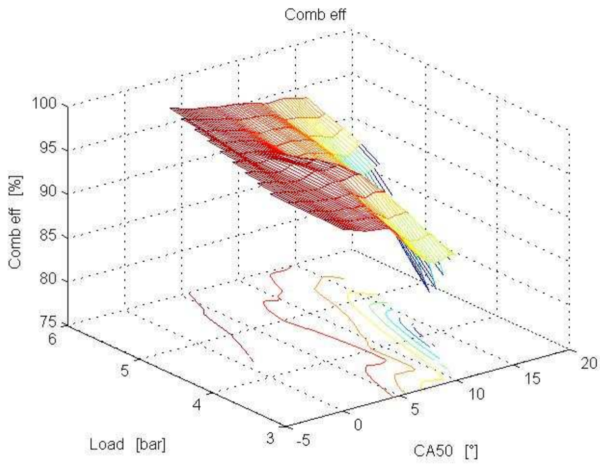

CO
CO [g/kWh]
Load [bar]
CA50 [°]

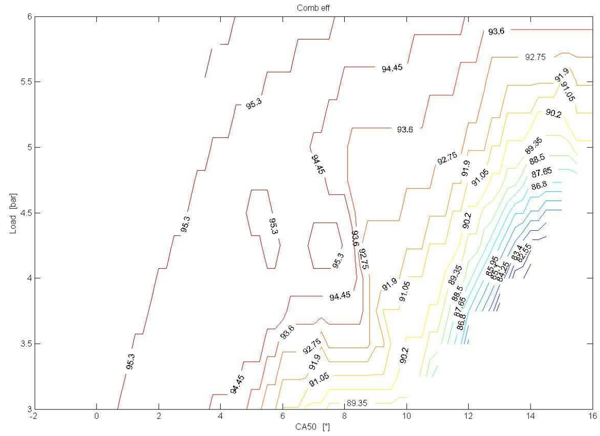

CO
Load [bar]
CA50 [°]

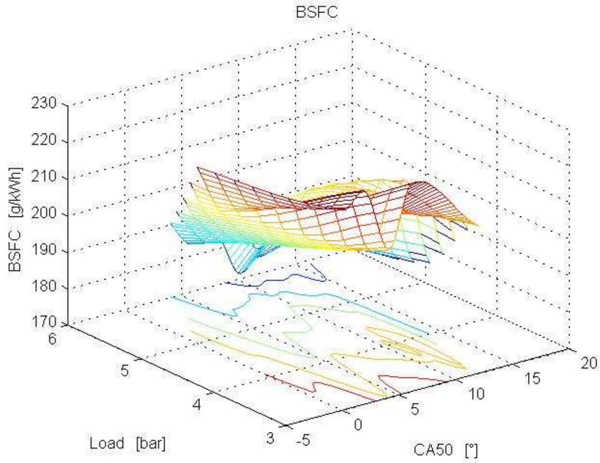

Lambda
Lambda
Load [bar]
CA50 [°]

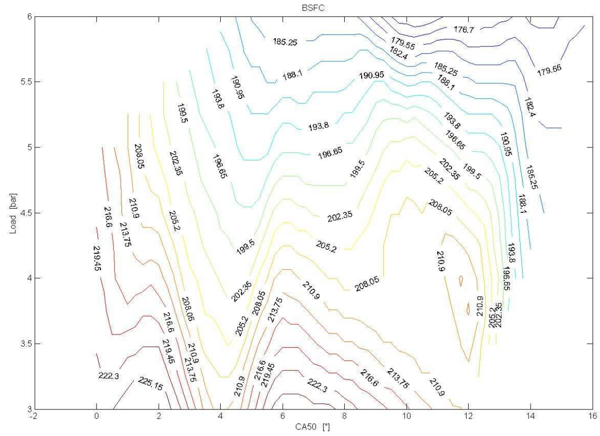

Lambda
Load [bar]
CA50 [°]

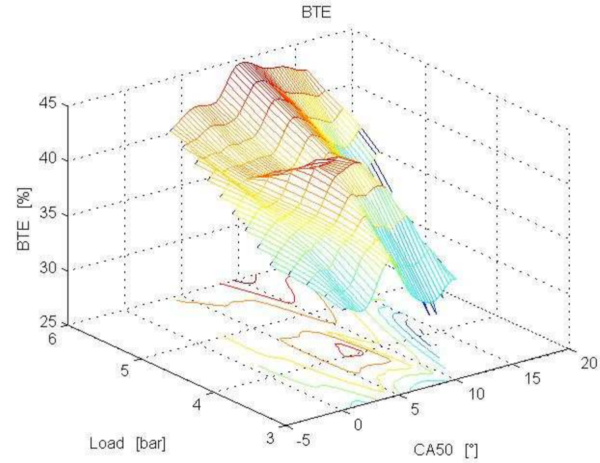
1800 rpm

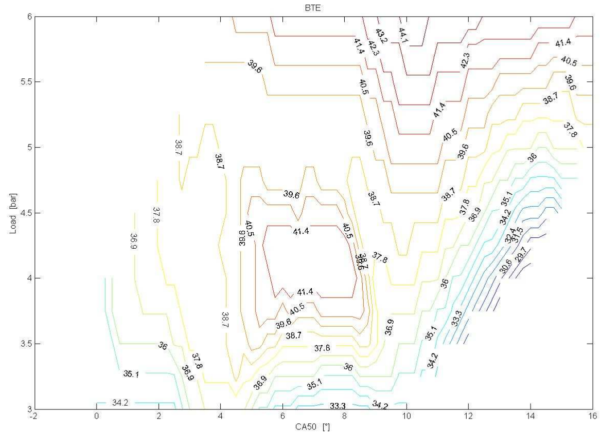

Comb eff
Comb eff [%]
Load [bar]
CA50 [°]

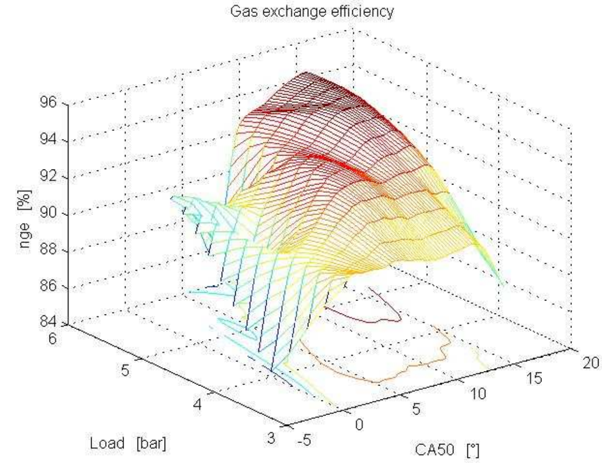

Comb eff
Load [bar]
CA50 [°]

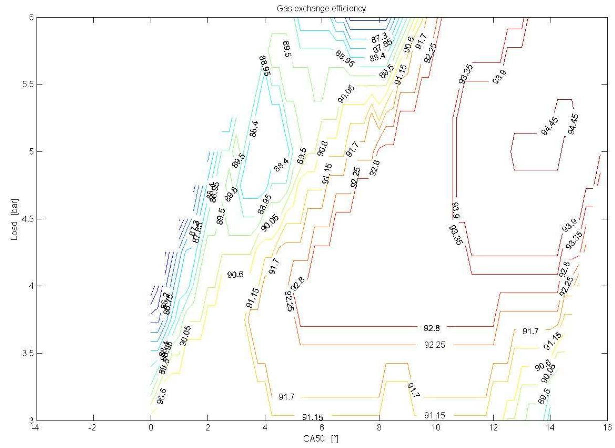

BSFC
BSFC [g/kWh]
Load [bar]
CA50 [°]


BSFC
Load [bar]
CA50 [°]

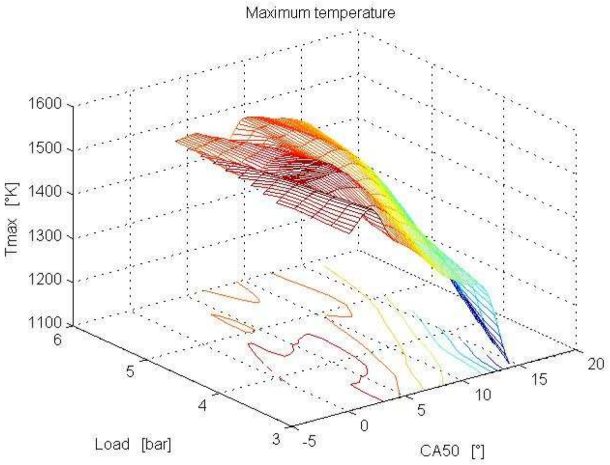

BTE
BTE [%]
Load [bar]
CA50 [°]

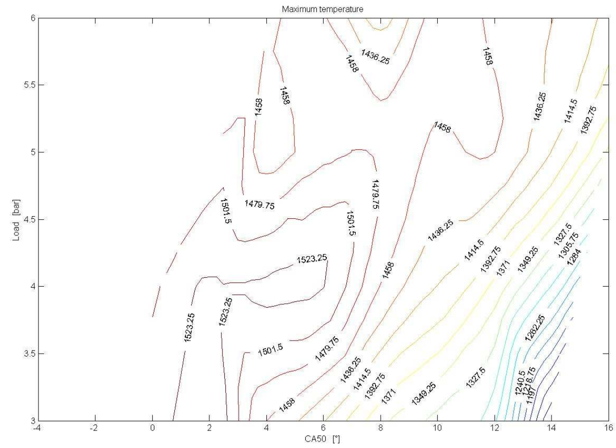

BTE
Load [bar]
CA50 [°]

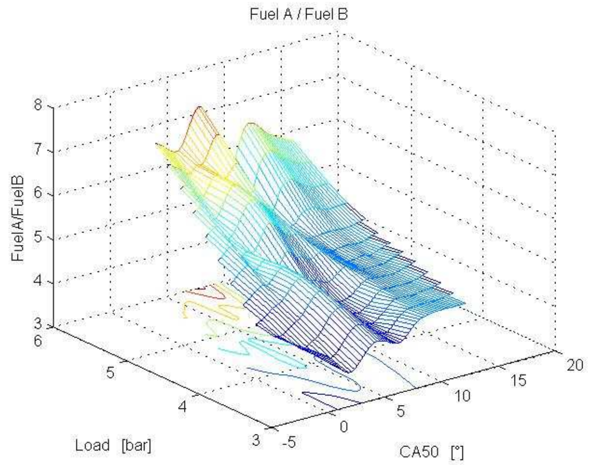

Gas exchange efficiency
nge [%]
Load [bar]
CA50 [°]

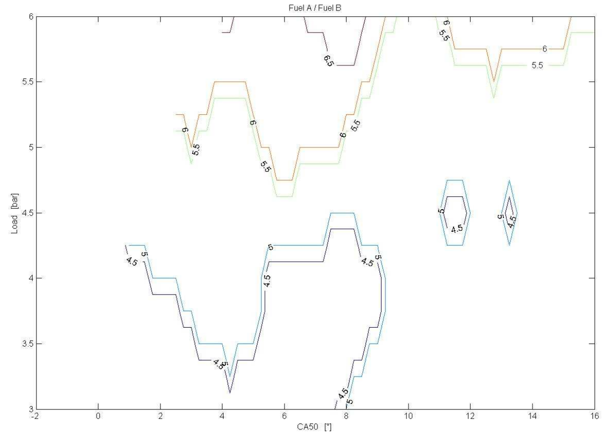

Gas exchange efficiency
Load [bar]
CA50 [°]

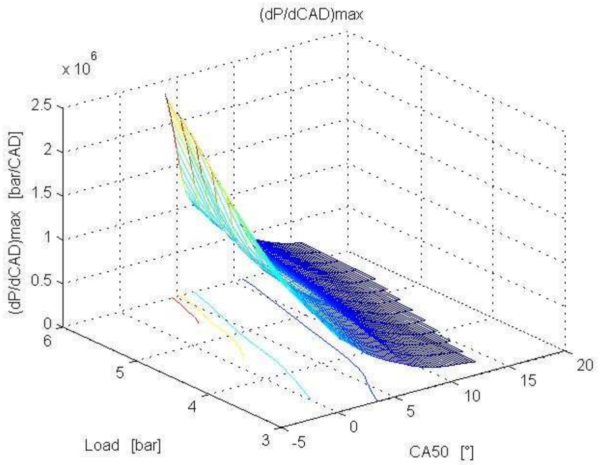
1800 rpm

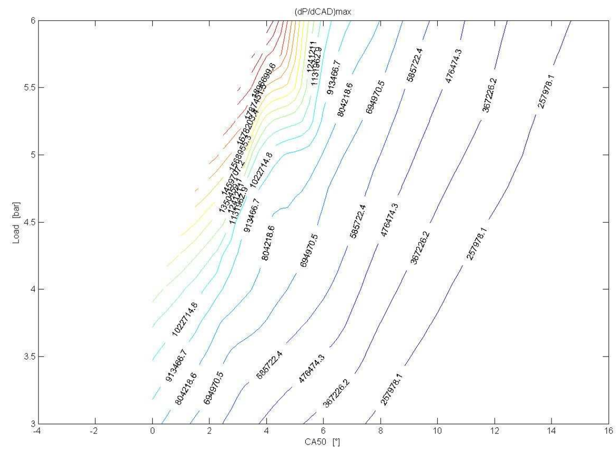

Maximum temperature
Tmax [°K]
Load [bar]
CA50 [°]

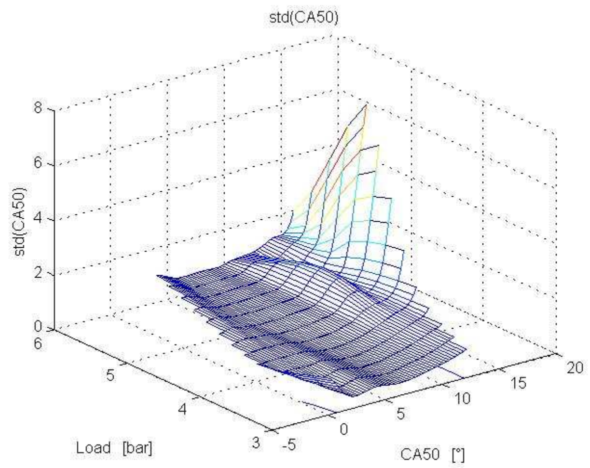

Maximum temperature
Load [bar]
CA50 [°]

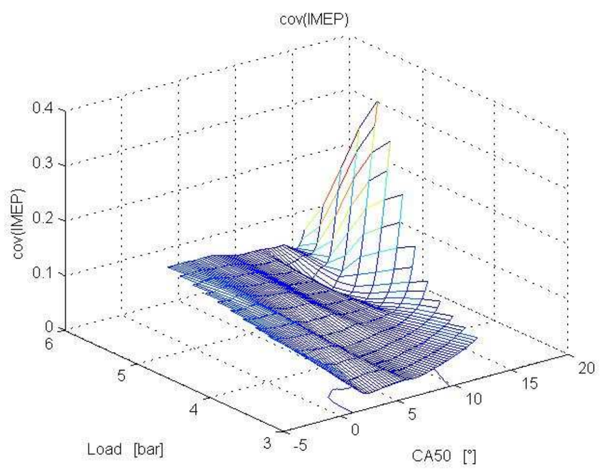

Fuel A / Fuel B
Fuel A/FuelB
Load [bar]
CA50 [°]


Fuel A / Fuel B
Load [bar]
CA50 [°]

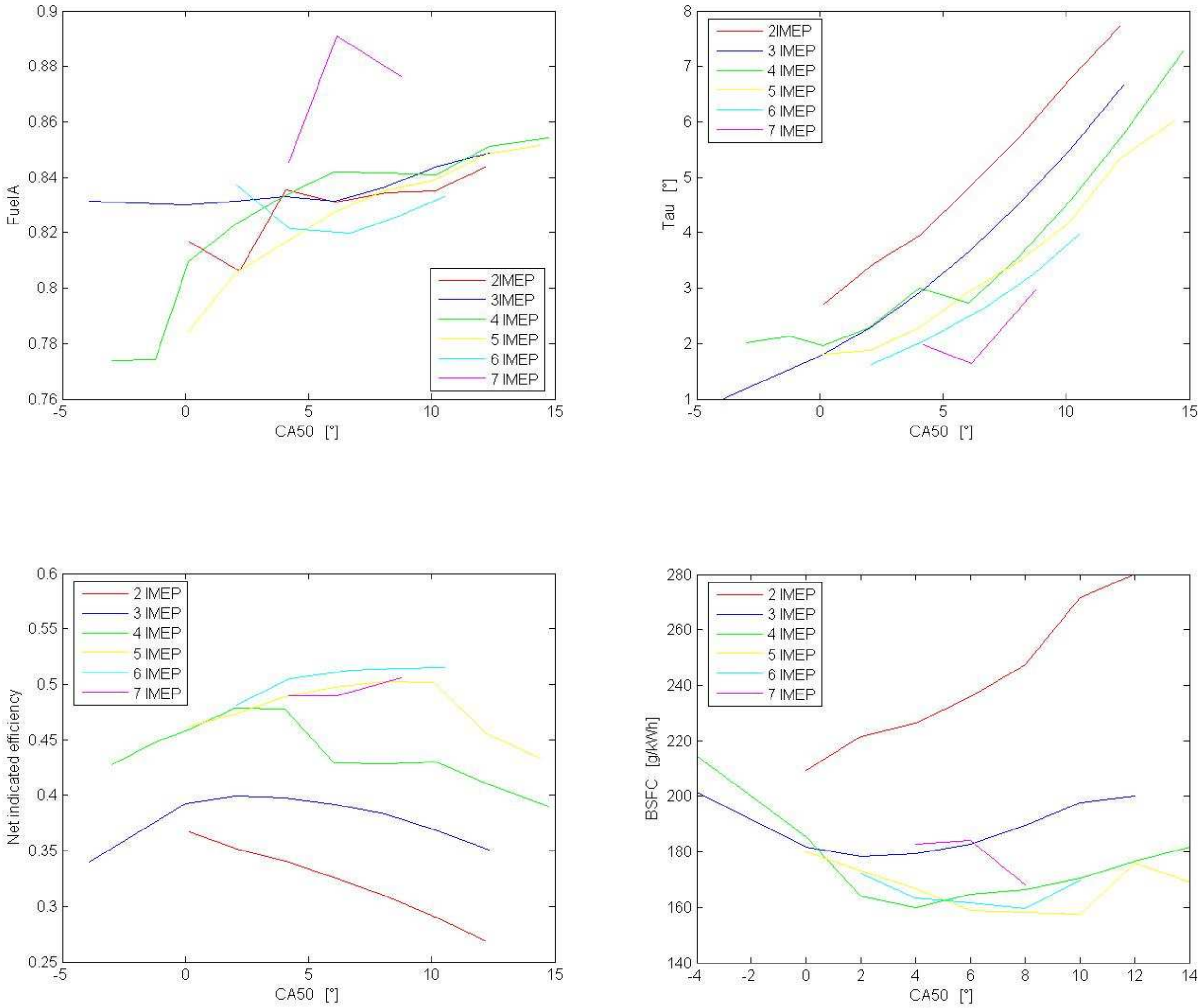

(dP/dCAD)max
x 10^6
(dP/dCAD)max [bar/CAD]
Load [bar]
CA50 [°]


(dP/dCAD)max
Load [bar]
CA50 [°]

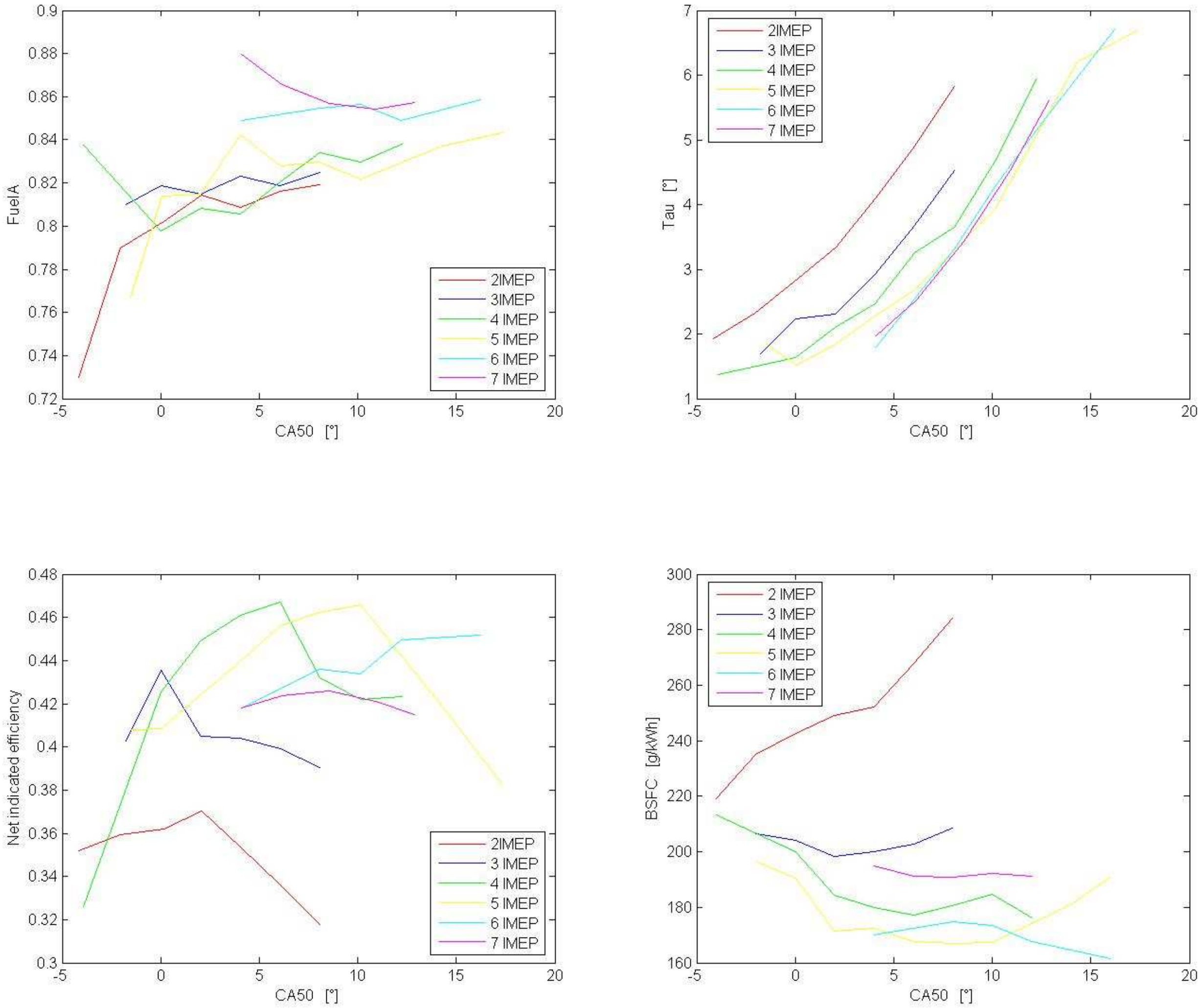

std(CA50)
std(CA50)
Load [bar]
CA50 [°]


cov(IMEP)
cov(IMEP)
Load [bar]
CA50 [°]

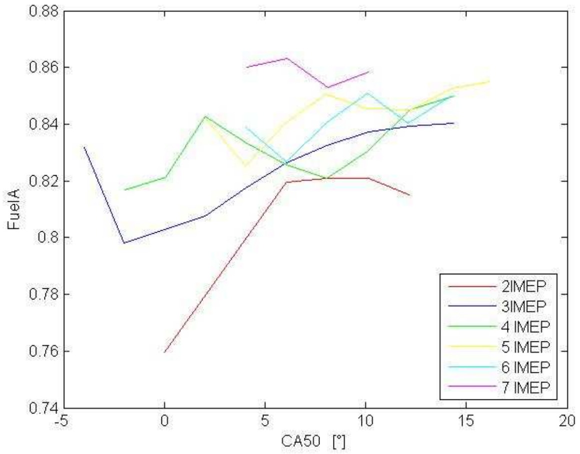
800 rpm

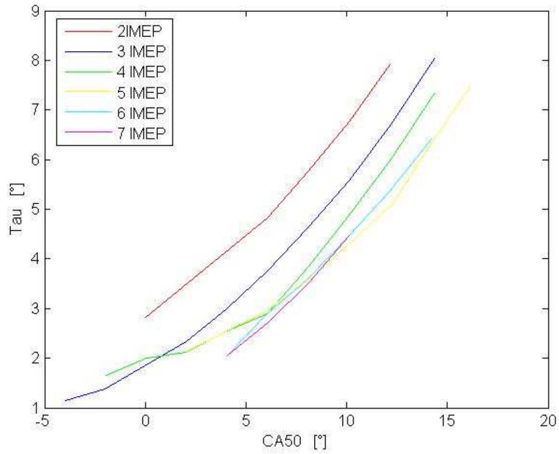

FuelA
CA50 [°]
2IMEP
3IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP
Tau [°]
CA50 [°]
2IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP
Net indicated efficiency
CA50 [°]
2 IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP
BSFC [g/kWh]
CA50 [°]
2 IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP

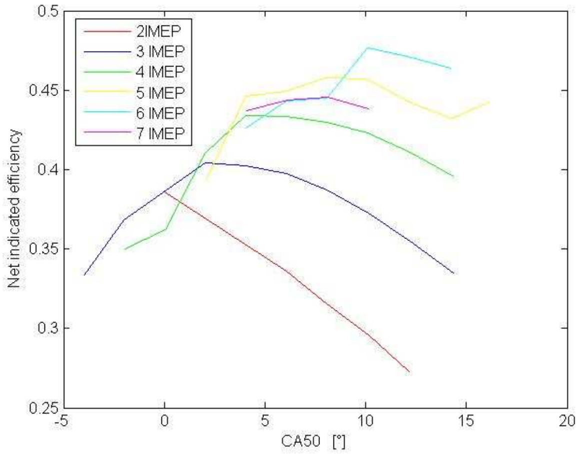
1000 rpm

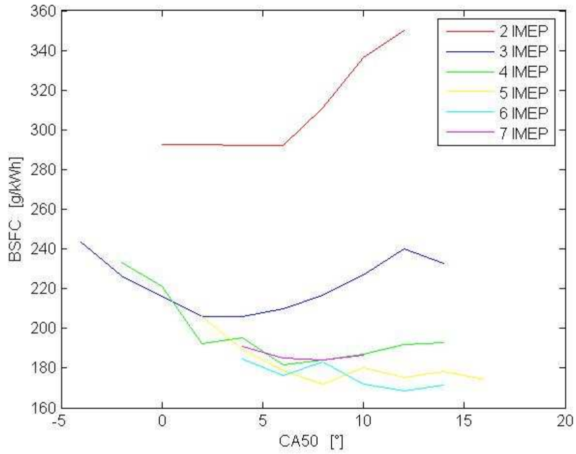

FuelA
CA50 [°]
2IMEP
3IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP
Tau [°]
CA50 [°]
2IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP
Net indicated efficiency
CA50 [°]
2IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP
BSFC [g/kWh]
CA50 [°]
2 IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP

1200 rpm

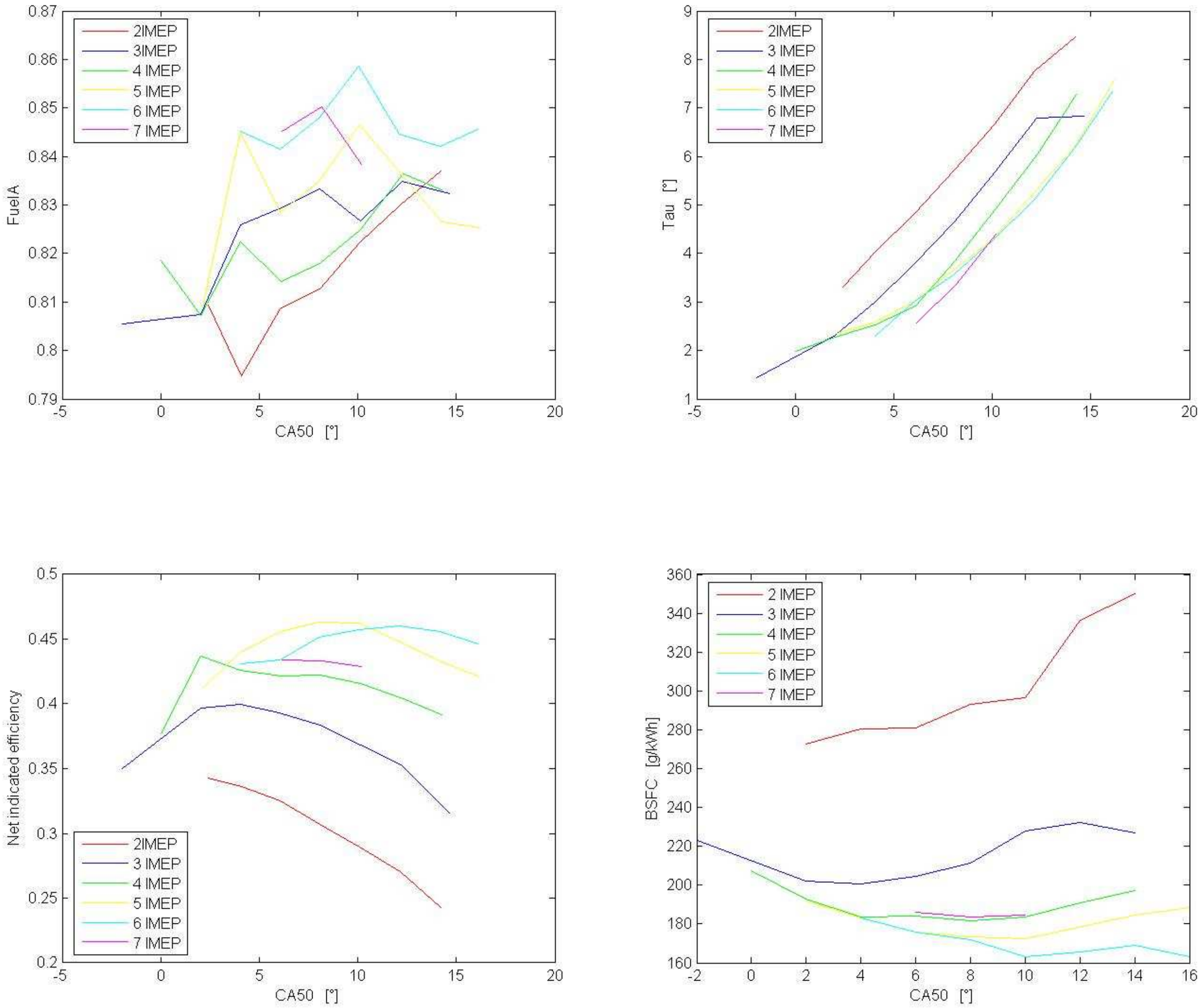

FuelA
CA50 [°]
2IMEP
3IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP


Tau [°]
CA50 [°]
2IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP

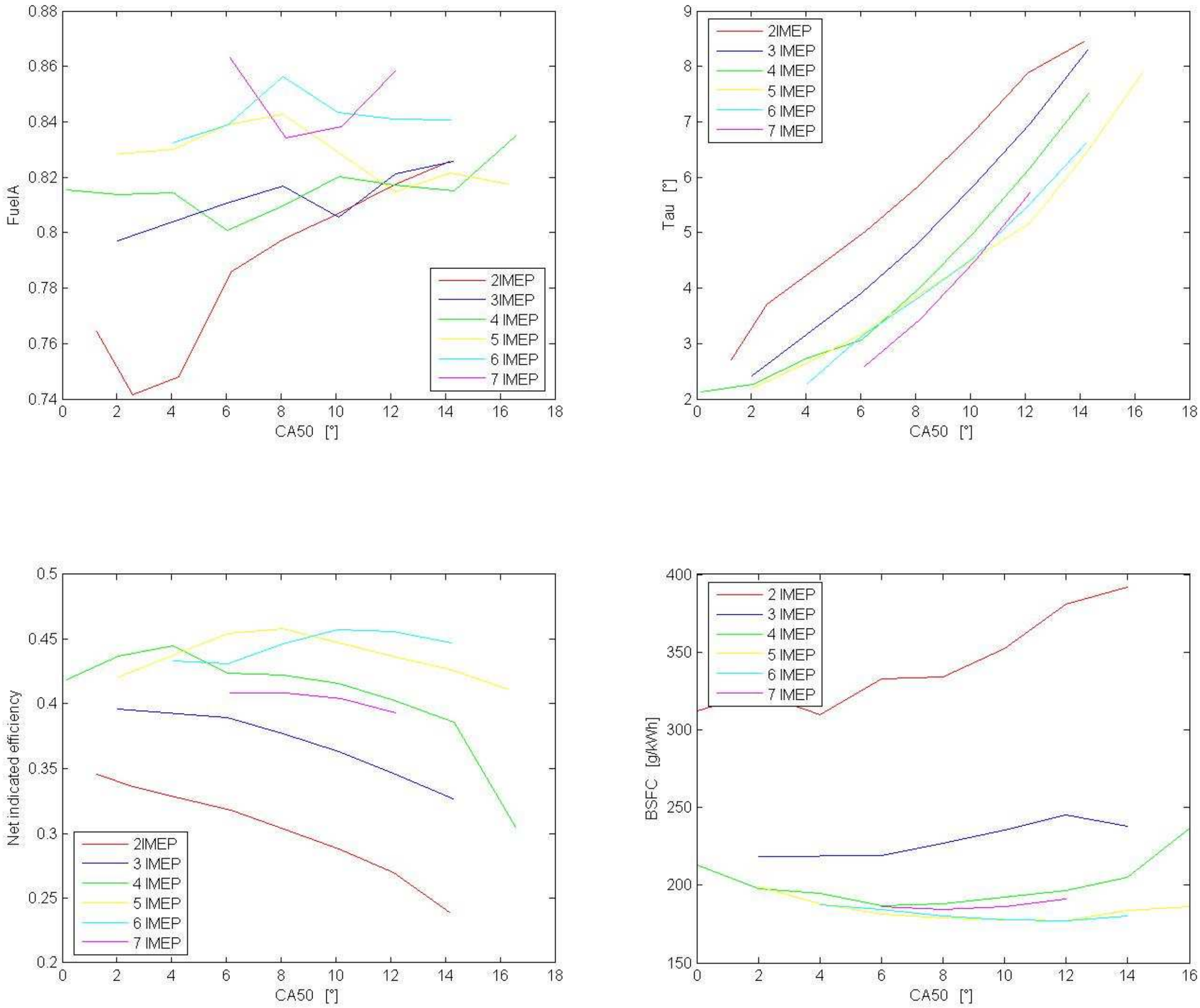

Net indicated efficiency
CA50 [°]
2IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP


BSFC [g/kWh]
CA50 [°]
2 IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP

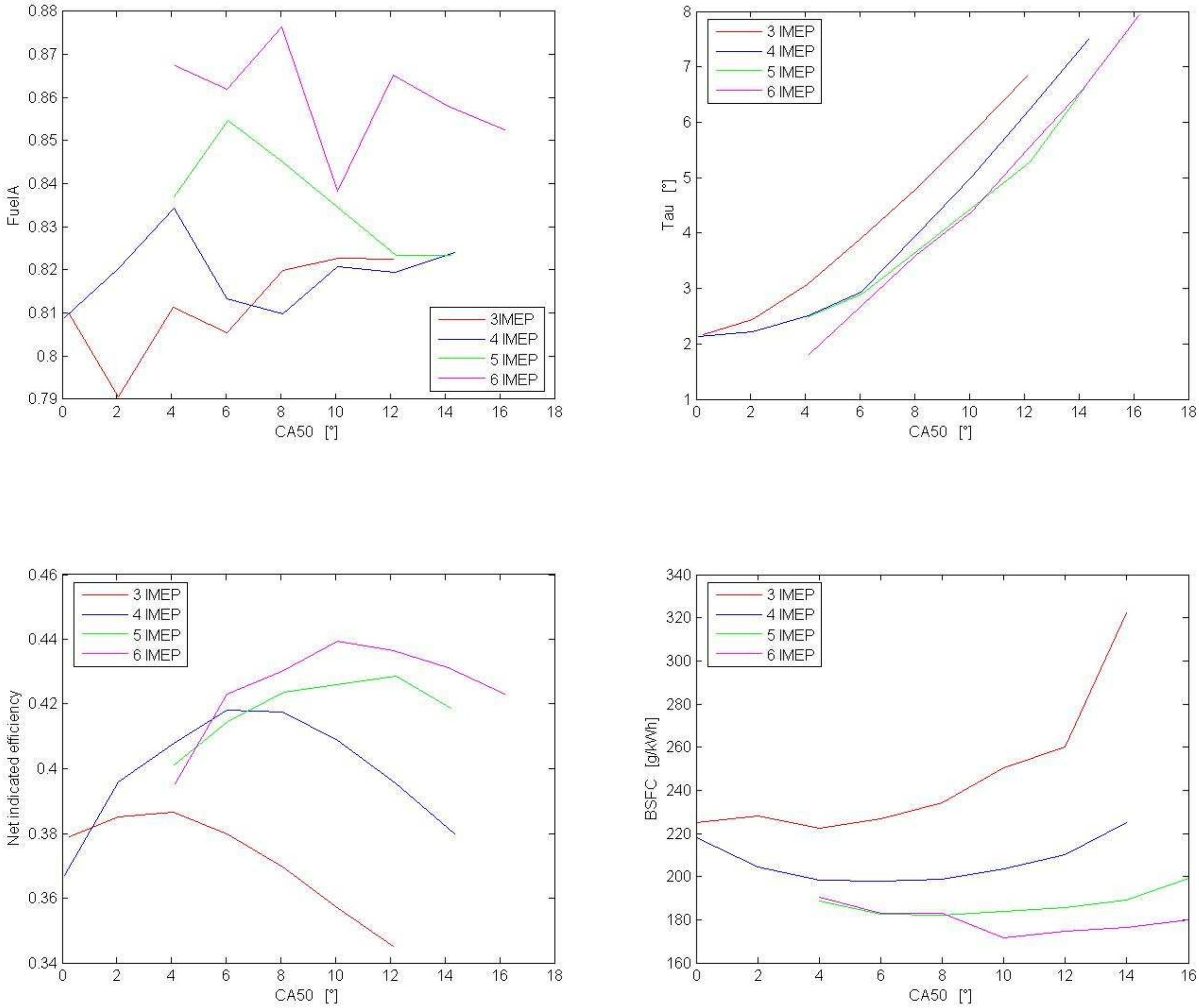
1400 rpm


FuelA
CA50 [°]
2IMEP
3IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP
Tau [°]
Net indicated efficiency
BSFC [g/kWh]
2 IMEP
3 IMEP

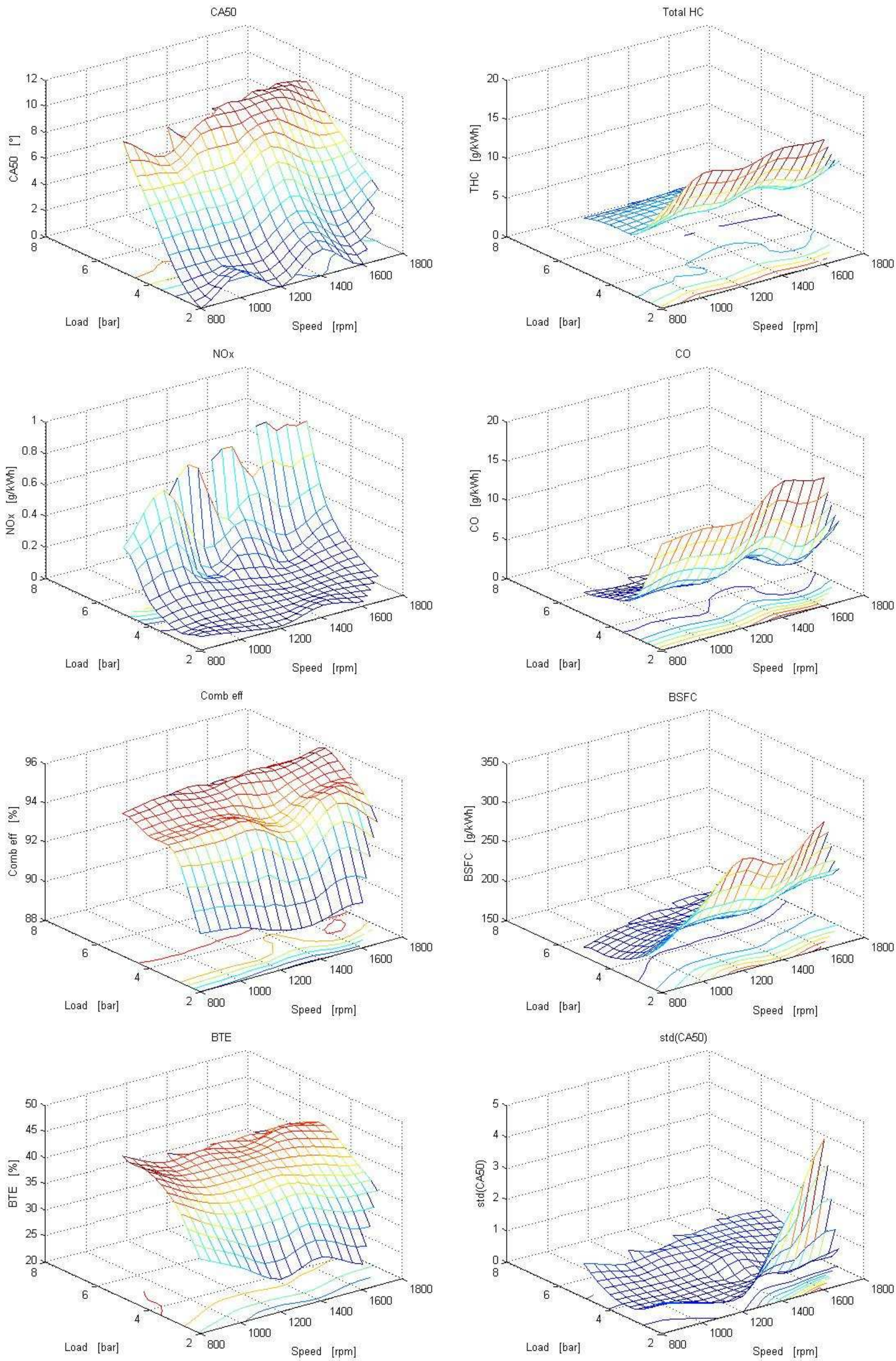
1600 rpm


FuelA
CA50 [°]
2IMEP
3IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP
Tau [°]
CA50 [°]
2IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP
Net indicated efficiency
CA50 [°]
2IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP
BSFC [g/kWh]
CA50 [°]
2 IMEP
3 IMEP
4 IMEP
5 IMEP
6 IMEP
7 IMEP

1800 rpm


FuelA
CA50 [°]
3IMEP
4 IMEP
5 IMEP
6 IMEP
Tau [°]
CA50 [°]
3 IMEP
4 IMEP
5 IMEP
6 IMEP
Net indicated efficiency
CA50 [°]
3 IMEP
4 IMEP
5 IMEP
6 IMEP
BSFC [g/kWh]
CA50 [°]
3 IMEP
4 IMEP
5 IMEP
6 IMEP

Conditions in the Maximum Net indicated efficiency



Conditions in the Maximum Net indicated efficiency