

THESIS FOR THE DEGREE OF LICENTIATE OF ENGINEERING

in

Thermo and Fluid Dynamics

An Experimental Study of Fischer-Tropsch  
Fuels in a Diesel Engine

by

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THESIS FOR LICENTIATE OF ENGINEERING no 2007:01  
ISSN 1652-8565

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Printed at Chalmers reproservice  
Göteborg, Sweden

## Abstract

Alternative fuels for Diesel engines include ester-based fuels (derived from sources such as rapeseed, soybeans and palm oil), dimethyl ether, and various synthetic fuels. Use of these fuels could contribute to a cleaner environment, reduce global warming and improve the human health by reducing levels of harmful emissions. One such fuel is Fischer-Tropsch (F-T) fuel, a synthetic fuel that can be produced from coal, natural gas or biomass, the properties of which heavily depend on the catalyst used in its production.

Various published studies have shown that use of F-T fuels could reduce emissions of soot,  $\text{NO}_x$ , HC and CO, to varying degrees, since they have a number of desirable properties, including low sulfur and aromatic contents, high cetane numbers and low density.

To further explore the performance of F-T fuels, and the emissions generated when they are used, engine tests were performed at Chalmers in both HCCI and conventional Diesel combustion modes. The engine used in the investigations was a single cylinder, 0.5 liter, research engine, and the performance of two F-T Diesel fuels derived from natural gas, produced by Statoil in Trondheim, was compared with that of two conventional Diesel fuels (Swedish low sulfur Diesel and European EN 590 Diesel).

In HCCI combustion mode the soot emissions were too low for meaningful comparison between the fuels. However, HC and  $\text{NO}_x$  emissions were lower when using the F-T fuels. There were no significant between-fuel differences in emissions of CO and  $\text{CO}_2$ , or fuel consumption. In HCCI combustion mode, the combustion was phased by using exhaust gas recirculation (EGR), thus there were no between-fuel differences in ignition delay, despite the early injection (which is a key feature of HCCI combustion). However, higher levels of EGR were needed for the F-T fuels than for the standard Diesel fuels, which reduced the thermal efficiency of the engine.

In conventional Diesel combustion mode the soot emissions were markedly lower (by up to 30 percent) when using the F-T fuels than when using the already clean Swedish low-sulfur Diesel. Reductions were also found in HC and CO emissions, while  $\text{CO}_2$  and fuel consumption values obtained with the different fuels were similar. The high cetane number of the F-T fuels resulted in shorter ignition delays for the pilot injections.

A challenge that had to be met before widespread use of F-T fuels could be economically viable was to minimize their production costs. However, these costs have fallen sharply, through a combination of intense research efforts and the construction of F-T plants with high production capacities, and they are now competitive with oil from fossil sources, so long as the price of the latter is higher than \$25/barrel.

The possibilities of producing Fischer-Tropsch fuels from several feed-stocks and using existing engines without modifications are major advantages with F-T fuels. Moreover, F-T fuels can be blended in any proportions with petroleum-based Diesel, and existing refueling and maintenance infrastructure can be used for vehicles running on them. All

these advantages, and the cleanliness of their exhaust gases, make F-T fuels attractive alternative Diesel fuels.

## Acknowledgement

The first person I would like to acknowledge is my supervisor and examiner, Ingemar Denbratt, for providing me the opportunity to work on a subject of great interest to us both. He has also freely shared his knowledge as the work has progressed, and I am very grateful for all the time he has given me in supervision and discussions. Whenever I needed help he provided it.

Further, I would like to thank the Swedish Road Administration and CERC for financial support during these years.

I am grateful for the support and advice given by all the members of the project group, including: Börje Gevert and Stylianos Nassos at Applied Surface Chemistry, Chalmers; Sivert Hiljemark and Erik Olsson at CERC; Magnus Nilsson at GM; Günther Kleinschek at Scania; Bjarne Lindberg and Åsa Håkansson at Statoil; Lisa Jacobsson at Volvo Cars Corporation; and Anders Røj at Volvo Technology.

Rolf Berg, Lars Jernqvist, Ingemar Johansson and Morgan Svensson, are acknowledged for their help in the lab with equipment and engine tests.

Sandra Arvidsson, Ulla Lindberg-Thieme, and Monika Orrbacke at Applied Mechanics, I am grateful for all your help with administrative problems.

Thanks to my colleagues at the department of Applied Mechanics for creating an enjoyable working environment, and to Arjan Helmantel for being an amusing roommate and sharing his knowledge concerning engines, and to Miriam Bergman for friendly support.

To my friends outside work, I appreciate sharing other interests, that are not work-related, with you.

I would also like to thank all of my family, especially my father, for sharing my interest in alternative fuels and supporting me in my studies.

Finally, to the most important person in my life, Staffan, thank you for everything.

## Abbreviations

ATDC	after top dead center
BTDC	before top dead center
BTL	biomass to liquid
CAD	crank angle degrees
CA50	crank angle where 50% of the heat is released
CFPP	cold filter plug point
CNG	compressed natural gas
Co	cobalt
CO	carbon monoxide
CO <sub>2</sub>	carbon dioxide
CTL	coal to liquid
DME	dimethyl ether
EGR	exhaust gas recycle
GHG	green house gases
GTL	natural gas to liquid
HC	hydrocarbons
FBP	final boiling point
FC	fuel consumption
F-T	Fischer-Tropsch
NO <sub>x</sub>	nitrogen oxides
PME	palm methyl ester
RME	rapeseed methyl ester
VCO	valve covers orifice

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# 1 Introduction

*In December 2005 the Swedish government appointed a Commission to formulate a program for reducing the dependency of Sweden on oil. The results of the Commission's consideration resulted in a report, published in June 2006, setting a goal to reduce fossil oil consumption in Sweden to zero by 2020. Meeting this target will be a huge challenge for Sweden and a revolution in energy management is required. Proposals by the Commission include (inter alia) phasing out the use of oil in the heating sector and replacing it with biomass fuels. In the transport sector the main proposal is to increase the use of Diesel vehicles (with aftertreatment systems), due to their high efficiencies. In addition, the Commission recommended use of hybrid vehicles in cities, which can reduce fuel consumption by up to 35%, and the provision of facilities for recharging their batteries with electricity from renewable energy sources (plug-in hybrids). The Commission also proposed use of Dimethyl ether (DME) in vehicles that can be refueled at just a few places, such as public city busses, garbage trucks etc. Moreover use of synthetic fuels, produced from biomass, is an alternative for Diesel engines. Such synthetic fuels are easy to handle, can be used in current, unmodified Diesel engines and existing infrastructure can be used for refueling and maintaining vehicles running on them. [1]*

## 1.1 Diesel engines

The Diesel engine was invented in 1892 by the German Rudolf Diesel, who aimed to create a highly efficient engine. He succeeded, and Diesel engines have higher efficiency than gasoline engines, since they have higher compression ratios (14:1-25:1 versus 8:1-12:1). The reason for this is that in the Diesel engine air is first compressed in the cylinders, fuel is injected close to top dead center and the charge is auto-ignited. In the gasoline engine, in contrast, air and fuel are compressed together, the charge is ignited by a spark and to avoid the risk of auto-ignition (knock) the compression ratio of the gasoline engine must be kept relatively low. In a Diesel engine the load is controlled by varying the amount of fuel injected in each cycle while the flow of air is unchanged for a given engine speed. In the gasoline engine, on the other hand, the engine is throttled at lower loads resulting in efficiency losses. [2]

## 1.2 Standards

Due to increasingly strict emission legislation the engine manufacturers are being obliged to develop less polluting engines that generate lower levels of harmful emissions. Their research and development efforts have resulted in Diesel engines that produce extremely low levels of emissions. However more can be improved and optimized.

The standards that have been applied in Europe to the emissions of particulate matter (PM) and nitric oxides (NO<sub>x</sub>) plus hydrocarbons (HC) from light-duty vehicles are presented in Figure 1. As shown in the figure, the limits have fallen remarkably from the first standards (Euro I, which came into force in 1992) to those due to be introduced in 2008 (EuroV).

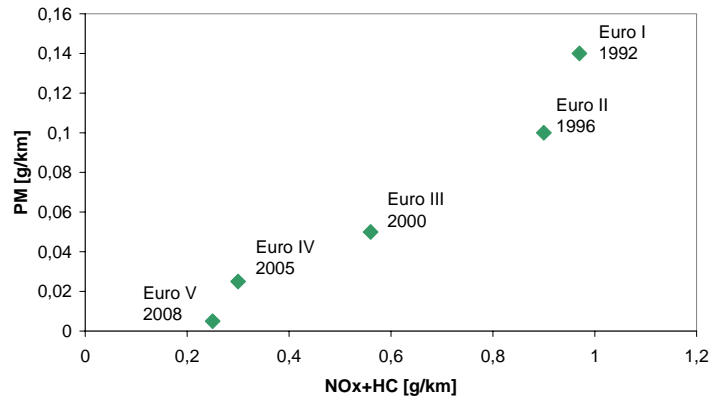


Figure 1. European emission standards for light-duty vehicles [3]

### 1.3 Oil reserves

Oil reserves are limited and concentrated in a few areas of the world. As shown in Figure 2, the largest reserves of oil at the end of 2005 were in the Middle East and it is predicted that by the year 2020, 54-67% of the oil produced globally will come from this area. [4]

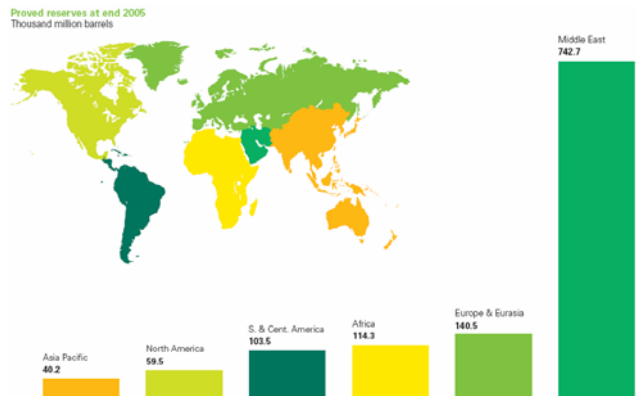


Figure 2. Oil reserves at end 2005 [5]

Oil production is expected to peak in the near future. After this point the reserves will not be empty, but production will begin to decline. Pessimistic and optimistic forecasts of when production will peak range from ca. three to ca. 15 years from now, respectively. [6]

## **1.4 Alternative fuels for Diesel engines**

*In this chapter some of the alternative fuels for Diesel engines are described and discussed. Vegetable oils are common feedstocks for alternative fuels, including virgin oil feedstock (rapeseed, soybeans, sunflower, palm oil etc), waste vegetable oils (from fast food restaurants, snack food factories etc.) and animal fats (tallow, lard, chicken fat etc.). Alternative fuel producers must compete for these feedstocks with the food and chemical industries. This is an important issue that may limit the availability of some alternative fuels, and it may therefore be necessary in the future to find ways to increase the use of other biomass fractions that are simply discarded today.*

### **1.4.1 Synthetic fuels**

Synthetic fuels are any liquid fuels produced from coal (CTL), natural gas (GTL), or biomass (BTL). These fuels are sulfur-free and contain extremely low levels of aromatics, resulting in clean combustion and low emissions of particulate matter. Generally, in order for synthetic fuels to be competitive with petroleum-based fuels, the price of crude oil must be relatively high. However, in countries like Sweden in which large amounts of biomass are available, it is possible to produce BTL fuels in large quantities, use of these fuels does not require any major modification of existing Diesel engines and offer opportunities to become self-sufficient in liquid fuels. [7]

### **1.4.2 Rapeseed methyl ester**

Biodiesel consists of mono-alkyl esters of long chain fatty acids usually derived from the esterification of vegetable oils with an alcohol using an appropriate catalyst. Rapeseed methyl ester (RME) is a Biodiesel produced from rapeseed oil, and is one of the feedstocks for which fuel producers must compete with the food industry. RME has been extensively tested by the engine manufacturers, for example Scania. RME fuels have been tested in engines with no modifications. Combustion has been found to result in slight increases in NO<sub>x</sub> emissions but lower CO and PM emissions than combustion of conventional Diesel fuel. Since the rapeseed cultivation capacity is insufficient to cover the needs of the transport industry, use of a blend of 5% RME in conventional Diesel is proposed for economic and environmental reasons. [8] A drawback of all “Biodiesel” fuels is their relatively high production costs. For small-scale production the cost of producing RME is \$2.5 per gallon and for large scale production \$1.5 per gallon. [9, 10]

### **1.4.3 Palm oil methyl ester**

In tropical countries, e.g. Malaysia, Indonesia, Thailand, and Colombia, production of palm oil Biodiesel is increasing. The vegetable palm oil is converted by a chemical process called transesterification to Biodiesel. Transesterification is the most economic of the possible routes to an alkyl ester. The resulting “PME Diesel” has several attractive properties, such as high cetane number and low aromatic contents. A disadvantage is that PME fuels have rather high cold filter plug point, making them difficult to handle in countries with Nordic climates. This is a general problem for vegetable oil fuels. However, in tropical countries the fuel can be used in Diesel engines without any

problems. Indeed, their use reduces soot-NO<sub>x</sub> trade-off points, and emissions of both HC and CO. [11]

### 1.4.4 Dimethyl ether

Like synthetic fuels, dimethyl ether (DME) fuel can be produced from coal, natural gas or biomass, depending on the regional resources. DME is a clean fuel and its combustion results in very low soot emissions. DME is a gas at ambient pressures and has much lower viscosity than conventional Diesel fuels. Therefore the injection systems of engines in which it is to be used must be modified. Moreover, existing fuel supply infrastructure would have to be reconstructed to enable widespread use of DME. However, DME fuel is a very attractive alternative to Diesel fuel in cities, for vehicles like buses and garbage trucks that can be refueled at just a few stations. [12]

## 1.5 Well-to-wheels analysis

In a well-to-wheels analysis, the energy inputs required to produce and use a fuel are calculated, and usually expressed in terms of MJ per 100 km traveled by a standard vehicle using it. Thus, such analyses provide comparative indications of the overall energy efficiency of different fuels. As shown in Figure 3, the energy inputs of synthetic fuels are higher those of conventional Diesel, especially synthetic fuel from farmed wood (for which the inputs are comparable to those of RME). However, production of synthetic Diesel from waste wood (and natural gas) is more energy-efficient than production of RME fuel.

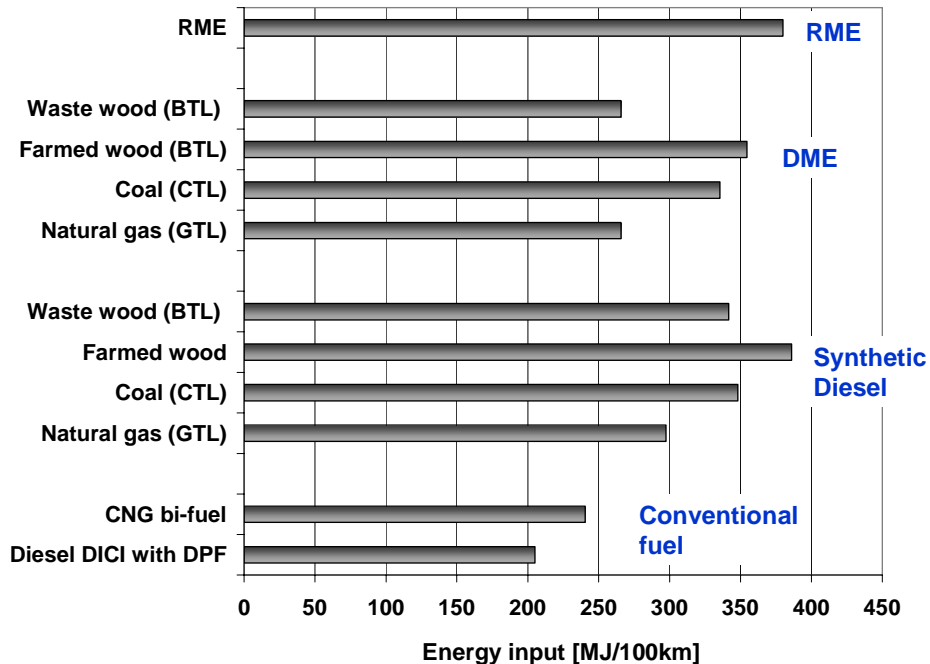


Figure 3. Well-to-wheels energy inputs for RME, DME, synthetic Diesel and conventional Diesel. [13]

There is growing consensus that human production of “greenhouse gases” is causing global warming, and that reducing their emission is a serious challenge that must be addressed. Well-to-wheel emissions of green-house gases resulting from production and use of RME, DME, synthetic Diesel and conventional Diesel in gCO<sub>2</sub>equivalenst/km [13] shows well-to-wheel levels of the main greenhouse gases generated by the combustion of RME, DME, synthetic Diesel and conventional fuels, in CO<sub>2</sub> equivalents. It can be seen that GHG emissions are highest for DME and synthetic Diesel produced from coal (double those for conventional fuels). These data imply that production of these fuels from coal would have a severe impact on global warming. The best fuels for minimizing emissions of GHG are DME and synthetic Diesel from waste and farmed wood.

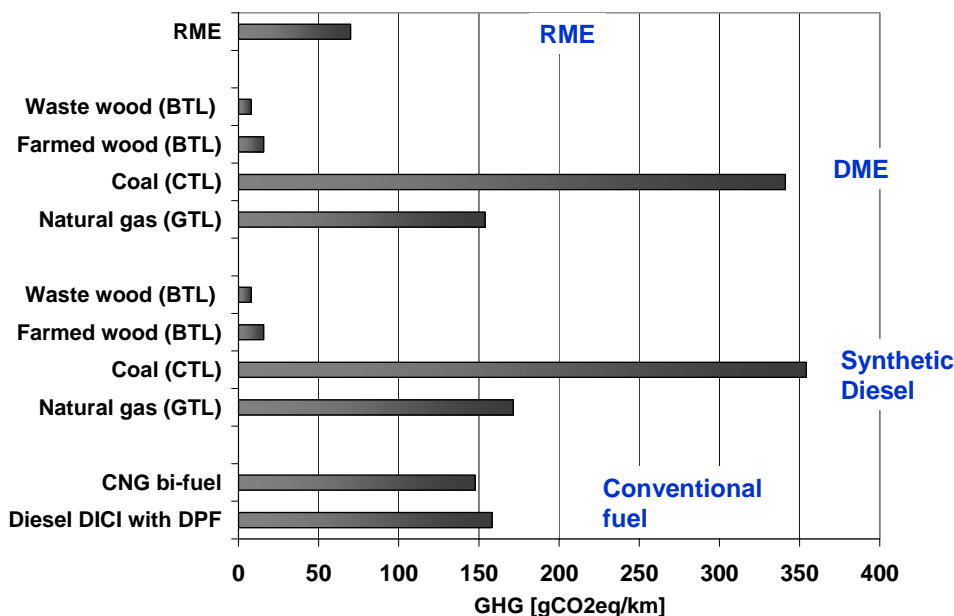


Figure 4. Well-to-wheel emissions of green-house gases resulting from production and use of RME, DME, synthetic Diesel and conventional Diesel in gCO<sub>2</sub>equivalenst/km [13]

An alternative solution for reducing CO<sub>2</sub> emissions and global warming would be to capture and store CO<sub>2</sub>, i.e. to separate CO<sub>2</sub> emissions from exhausts and other wastes generated in industrial and energy-related processes, then store them long-term in isolation from the atmosphere. CO<sub>2</sub> storage facilities can reduce CO<sub>2</sub> emissions from power plants by up to 80-90%, but they also increase their energy consumption by ca. 10-40%. [14]



## 2 The Fischer-Tropsch fuel

*This chapter reviews the history and production processes of synthetic fuel Fischer-Tropsch Diesel.*

### 2.1 History

Fischer-Tropsch synthesis was developed in the 1920's by the researchers Franz Fischer and Hans Tropsch in Germany, and production of liquid fuel from coal using the process began in the following years. In 1938, nine F-T plants were in operation in Germany and during World War II the Germans fueled their airplanes with F-T fuel. However, following the discovery of large oil reserves in the Middle East in the mid-1950's interest in F-T fuels declined. However, a plant to produce liquid fuel from coal was constructed in South Africa during the anti-Apartheid boycott in 1950, and further plants to produce it from natural gas were subsequently built. Today most of the buses in South Africa run on F-T fuel. [15,16]

### 2.2 From feedstock to fuel

*Fischer-Tropsch Diesel fuel can be produced from biomass, coal or natural gas. This section focuses on the processes involved in its production from natural-gas.*

#### 2.2.1 Production of Fischer-Tropsch diesel fuel

Producing Diesel fuel via the Fischer-Tropsch process involves several steps: conversion of natural gas to synthesis gas, the Fischer-Tropsch process and finally product purification.

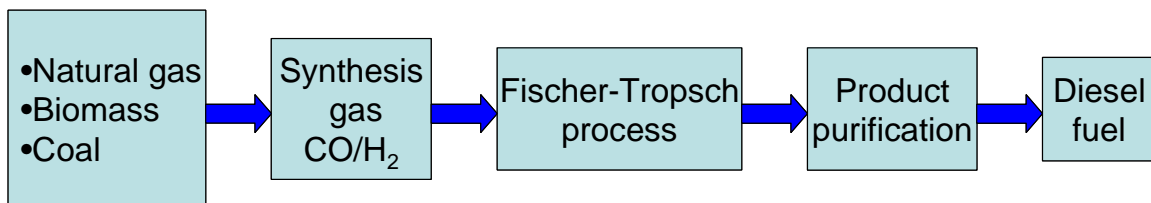


Figure 5. Feedstock to Diesel fuel.

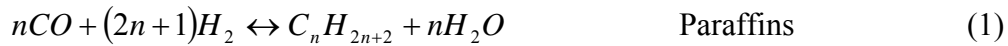
#### 2.2.2 Feedstock to synthesis gas

The most costly step in an F-T plant is the preparation of synthesis gas, which accounts for 60-70% of total costs. The synthesis gas can be produced from biomass, natural gas or coal. If natural gas (which consists largely of methane) or coal are used, they are first converted to H<sub>2</sub> and CO by steam-reformation or partial oxidization, and the H<sub>2</sub>/CO ratio is determined by the carbon source. Since methane has a higher hydrogen content than coal it has higher carbon efficiency; when coal is gasified to synthesis gas 50% of the carbon feed ends up in carbon-dioxide, and the corresponding figure for the steam reformation of methane is only 20%. [16,17]

### 2.2.3 Fischer-Tropsch process

The Fischer-Tropsch process involves transformation of the synthesis gas (CO and H<sub>2</sub>) to long chain hydrocarbons in a reactor. Either of two operational modes can be used: high-temperature F-T or low-temperature F-T. In high-temperature F-T two different types of reactors are used, fixed fluidized bed and circulating fluidized bed reactors, and the catalyst is Fe-based for both of these types of reactors. In low-temperature reactors, on the other hand, the catalyst can be either Fe- or Co-based and multitubular fixed-bed reactors or slurry phase reactors are used.

In the reactors the synthesis gas reacts on the catalyst surface and paraffins and olefins are produced. The process also includes the “water-gas-shift reaction” (WGS). The global reactions are shown below.



Active catalysts for the F-T synthesis are Ru and Co when synthesis gas is produced from natural gas. Co is the most common metal used in F-T synthesis due to its high activity and selectivity for long chain paraffins. It also has low water-gas shift activity, which is desirable when the feedstock is natural gas. Cobalt catalysts are also believed to deactivate less rapidly and yield higher fractions of linear alkenes than to iron catalysts. [15,17,18]

#### *Selectivity*

Selectivity is defined as the amount of desired product(s) obtained per unit consumed reactant. Fischer-Tropsch synthesis generates paraffins, olefins, and oxygenated products and its selectivity is influenced by several parameters, including (inter alia) the temperature, H<sub>2</sub>/CO ratio in the feed gas, pressure, and the type of catalyst used. [15]

Increases in temperature increase the proportions of products with relatively low carbon numbers when a Co catalyst is used. Thus, since products with high carbon numbers are desired when the Fischer-Tropsch process is used to make diesel fuel, low temperatures are required. [15] Increases in the H<sub>2</sub>/CO ratio of the feed gas result in undesirable increase in the proportions of short hydrocarbons and reductions in olefin contents. However, with too low H<sub>2</sub>/CO ratios, the CO saturates the catalyst surface, and hence the rate of CO consumption decreases. [19] Increasing total pressure up to 20 bar increases

the length of the hydrocarbon chains produced, but further increases in pressure have little additional effect. [15]

### *Supports*

Due to the high prices of catalysts it is important to minimize their use by increasing the available surface area of the metal used. Moreover, in catalytic processes it is also important to optimize the surface area in order to optimize activity, selectivity and reaction rates. To increase the surface area, the catalytic metal is distributed in very small particles on a supporting structure, which provides a larger total metal surface area than using a thin layer of metal.

In Fischer-Tropsch cobalt catalysts, the supports are usually  $\text{Al}_2\text{O}_3$ ,  $\text{SiO}_2$  or  $\text{TiO}_2$ . The effects of metal-support interaction, pore diameter, dispersion, acidity of the support, and other modifications of the support are important factors that affect the catalysts' performance.

During the course of its reaction with the support, cobalt is converted to  $\text{Co}_3\text{O}_4$ , and has to be reduced Co again. [15]

### **2.3.4 Promoters**

A promoter is a small amount of a second metal that is added to the catalyst in order to increase its reduction, activity and selectivity. In F-T catalysts the most common promoters are Pt, Ru and Re.  $\text{Co}_3\text{O}_4$  is normally reduced in two steps, first to  $\text{CoO}$  and then to  $\text{Co}$ . Pt and Ru are known to promote the reduction in both steps, and Re only the first step.

Use of Re as the promoter on a  $\text{Co}/\text{TiO}_2$  catalyst increases selectivity to production of higher hydrocarbons and conversion, and with an  $\text{Al}_2\text{O}_3$  supported Co catalyst it increases the selectivity to  $\text{C}_{5+}$  products. Ru improves F-T reaction rates of both  $\text{Co}/\text{SiO}_2$  and  $\text{Co}/\text{TiO}_2$  catalysts, and with  $\text{Co}/\text{Al}_2\text{O}_3$  catalysts it both increases the reaction rate catalyst and changes the product selectivity slightly. [15]

### **2.3.5 Effect of water**

During Fischer-Tropsch synthesis water is produced, and has varying effects on the Co-catalysts depending on the support and promoter used. In some cases adding water to the feed gas can have desirable effects on the reaction. However, Re-promoted  $\text{Co}/\text{Al}_2\text{O}_3$  are deactivated more quickly than unpromoted  $\text{Co}/\text{Al}_2\text{O}_3$  catalysts in the presence of high water partial pressures, due to the loss of active sites. [20]

In Pt-promoted  $\text{Co}/\text{Al}_2\text{O}_3$  catalysts small amounts of water reversibly deactivates the catalyst while addition of more than 25 % amount of water in the feed deactivates them permanently. [20]

High water partial pressures also result in rapid deactivation of  $\text{Co}/\text{SiO}_2$  catalyst, due to increase in pore diameter and pore volume, with associated reductions of BET area when cobalt silicates are produced. [20]

## 2.2.6 Product purification

In the product purification process the n-paraffins, in the form of waxes, are transformed to Diesel fuel with desired properties. The straight hydrocarbon chains, paraffins, are transformed in an isomerization process to branched hydrocarbon chains, isoparaffins. After the isomerization, components of the gas with boiling points higher and lower  $25^\circ\text{C}$  are condensed in a high temperature condenser and a low temperature condenser, respectively, and the condensates are then fractionated in a distillation column into fractions with desired boiling intervals. The product purification process is illustrated in Figure 6. [21]

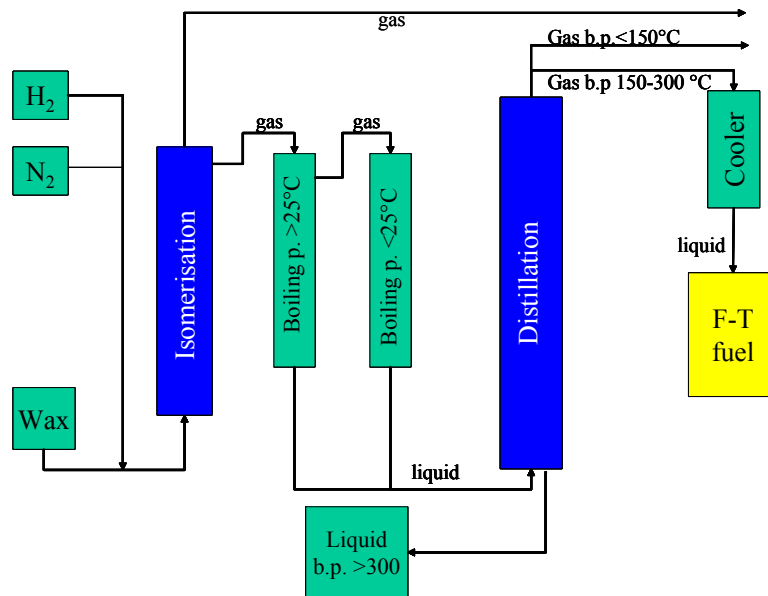


Figure 6. Purification of product paraffins to Fischer-Tropsch Diesel fuel.

## 2.3 Existing F-T plants

Currently operating F-T plants are located in South Africa, Malaysia and Qatar. In South Africa the oil company, Sasol owns three coal-based plants that have been operating since 1955 (Sasol I), 1980 (Sasol II) and 1982 (Sasol III) which collectively produce 6 000 000 tons fuel per year. An other South African oil company, Mossgas, started operating a natural gas-based plant in 1992 that has the capacity to produce 1 000 000 tons fuels per year. A third South African oil company, PetroSA, has also begun to

produce 35000 tons of F-T fuels per year in a pilot plant that was started up in 2005. In Malaysia, Shell has been operating a natural gas-based plant that produces 500 000 tons of fuel per year since 1993 and, finally, in collaboration with Qatar Petroleum, Shell has constructed a natural gas-based plant in Qatar, with a planned production capacity of 4 000 000 tons per year. [15,17,22,23]

## **2.4                      *Cost of producing F-T fuels***

The production of F-T fuels has high investment costs. However, calculations of the comparative costs of producing F-T fuels and conventional petroleum fuel indicate that, surprisingly, a GTL project based on current technology (assuming natural gas prices in the range \$0.5 to \$1.0 per million Btu) would be cost-competitive with petroleum if the price of crude oil is higher than \$25 per barrel; far lower than the current price of oil (\$54 per barrel, 2007-01-10). Furthermore, forecasts indicate that oil prices are unlikely to decline in the future, so investment in GTL fuels already appears to be profitable. [24, 25]



### 3 Emissions from Diesel engines

The regulated emissions from Diesel engines are particulates,  $NO_x$ , HC and CO emissions. This chapter considers emissions of these species and, in addition, emissions of  $CO_2$ .

#### 3.1 Particulate emissions

In Diesel engines particulate emissions are a major problem. Particulates are defined as any substances apart from water that can be filtered at 325 K in the exhaust gases and they are formed as a result of incomplete combustion of the fuel hydrocarbons. The fuel contains paraffins, olefins and aromatics, all of which contribute to particulate emissions, but the main sources are aromatics. Soot is formed from reactions in which particles containing ca.  $10^5$  carbons with H/C ratios of about 0.1 are formed from fuel molecules containing 12 to 22 carbon atoms with H/C ratios of about 2. Initially the fuel molecules start to oxidise and pyrolyze to products of acetylene and polycyclic aromatic hydrocarbons, which are the most likely precursors of soot in flames. The first recognizable soot particles are very small, with diameter less than 2 nm, and are often called nuclei. After formation of the nuclei, the particles start to grow, via processes including surface growth, coagulation, and aggregation. [2, 26]

In Figure 7 a temperature-equivalence ratio, calculated by Golovitchev et al [27], showing the soot and  $NO$  formation area is illustrated. The line in the figure illustrates the adiabatic flame temperature. As can be seen in the figure, soot formation takes place between 1650 and 2300 K, and the soot particles oxidize in the lean reaction zone at the periphery of the spray where  $NO_x$  is also formed. If the temperatures are high during the exhaust stroke then there will be increased oxidation and thus lower emissions of smoke and particulates. [28]

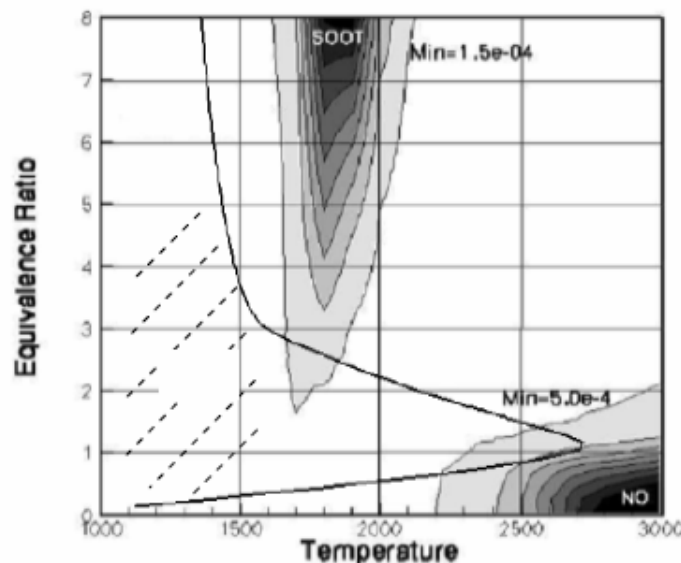


Figure 7. Temperature-equivalence map showing  $NO$  and soot formation regions [28]

### **3.2** *NO<sub>x</sub> emissions*

Nitric oxide (NO) and nitrogen dioxide (NO<sub>2</sub>) are grouped together as NO<sub>x</sub> emissions, NO being the dominant contributor. In Figure 7 it can be seen that NO<sub>x</sub> emissions are mainly formed at temperatures higher than 2200 K and equivalence ratios below 2.

NO<sub>x</sub> emissions are formed by four different mechanisms. The thermal (Zeldovich) mechanism occurs in the hot combustion gases (>1700 K). In the second, “prompt” mechanism CH radicals formed in the flame front react with nitrogen in the air, thereby forming hydrocyanic acid (HCN), which generates NO<sub>x</sub> in further reactions. The prompt mechanism requires a rich mixture and temperatures higher than 1000 K. The third mechanism, the nitrous oxide (N<sub>2</sub>O) mechanism, is important at low temperatures and is significant in lean pre-mixed laminar flames. Conversion of fuel nitrogen into NO<sub>x</sub> is the fourth mechanism, and is primarily observed in coal combustion, due to its chemically-bound nitrogen. [29]

NO<sub>x</sub> emissions are mostly formed during the diffusion combustion phase. Several methods can be used to reduce them, including retarding the start of injection (which lowers the in-cylinder temperature and thus NO<sub>x</sub> formation) and EGR (which replaces oxygen with residual gases and reduces both the combustion temperature and NO<sub>x</sub> emissions). However, methods for reducing NO<sub>x</sub> emissions normally cause an increase in soot emissions and, thus, methods for lowering NO<sub>x</sub>-soot trade-off points e.g. high swirl and high injection pressures, are important. [26]

### **3.3** *HC emissions*

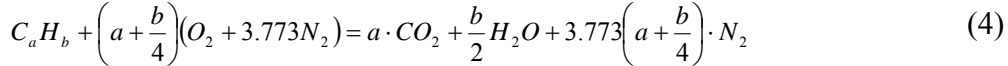
Emissions of hydrocarbon are a result of fuel escaping combustion, and over-leaning is the most important source of HC emissions in Diesel engines. During the ignition delay period, fuel and air are mixed rapidly locally to equivalence ratios lower than the flammability limit, and complete combustion is not possible. Moreover, under-mixing leads to locally over-rich mixtures, which also contribute to HC emissions. This occurs when the mixing is too slow and the mixture is too rich to burn, for example when the fuel in the nozzle orifice evaporates after the end of injection and leaves the injector at a very low velocity. It has been shown that increasing the ignition delay increases hydrocarbon emissions. To reduce the effects of under-mixing several different types of nozzles have been developed, including mini-sac and VCO (Valve Covers Orifice) nozzles etc. However, HC emissions from Diesel engines are not a major problem. [2]

### 3.4 CO and CO<sub>2</sub> emissions

Similarly to HC emissions, CO emissions from Diesel engines have not been considered to be a major problem. However, now that oxidation catalysts and low exhaust gas temperatures are becoming essential to meet increasingly strict legislative limits, CO emissions are starting to become an issue even for Diesel engines, especially when using high amounts of EGR to control NO<sub>x</sub> emissions. In SI engines the formation of CO emissions depends on the equivalence ratio,  $\Phi$ . Increases in  $\Phi$  can result in increases in CO emissions since there is insufficient oxygen for complete combustion. However, for fuel-lean mixtures, as in Diesel engines, the amount of CO emissions is only modestly affected by the equivalence ratio. The CO emissions are formed in the combustion process and are later oxidized to CO<sub>2</sub>. The reaction is not in equilibrium and the CO<sub>2</sub> molecules can be dissociated back to CO molecules and then re-oxidized to CO<sub>2</sub>. At 1500 K the oxidation reaction freezes and further oxidation is not possible. However, CO emissions can be reduced by increasing in-cylinder temperatures and/or the time available for oxidation.

The amounts of CO<sub>2</sub> emitted under ideal conditions can be estimated from the carbon:hydrogen ratio (C<sub>a</sub>H<sub>b</sub>) of the fuel and the stoichiometric air fuel ratio.

After the carbon:hydrogen constants of the fuel (a and b) have been calculated, the molar mass of the fuel can be calculated. Further, the energy per mole of fuel can be calculated by multiplying the heating value and the molar mass.



Equation (4) shows the numbers of moles of CO<sub>2</sub>, H<sub>2</sub>O and N<sub>2</sub> that each mole of fuel produces. The amounts produced per unit energy supplied can then be calculated. A higher H/C ratio results in a reduced amount of CO<sub>2</sub> emissions. [2] In Table 1 the amounts of CO<sub>2</sub> produced per unit energy are shown for the four fuels used in the engine tests underlying this thesis: Swedish low sulfur Diesel, European EN 590 Diesel and two Fischer-Tropsch fuels. For further description of the fuels, see Table 6. As the data show, the Fischer-Tropsch fuels produce lower CO<sub>2</sub> emissions under ideal conditions than the reference fuels.

**Table 1** calculated amounts of CO<sub>2</sub> produced from each of the fuels under ideal conditions.

	<b>Swedish low sulfur Diesel</b>	<b>European EN 590 Diesel</b>	<b>Fischer-Tropsch no 1</b>	<b>Fischer-Tropsch no 2</b>
H/C ratio	2.02	1.87	2.15	2.14
Heating value [MJ/kg]	43.5	42.8	43.7	43.6
CO <sub>2</sub> produced/unit energy [g CO <sub>2</sub> /MJ]	<b>0.03726</b>	<b>0.03828</b>	<b>0.03675</b>	<b>0.03684</b>



## 4 Engine tests with F-T fuels

*This chapter describes and discusses results from both the literature and engine tests performed at Chalmers.*

### 4.1 Results from the literature

F-T fuels have been extensively investigated, and compared in terms of emissions generated in engine tests and various other parameters, by workers based in diverse countries. The main results of some of these studies are briefly reviewed here. Clark compared emissions from various fuels including (inter alia) Malaysian F-T Diesel, soy-derived Biodiesel and a 49-state on-road Diesel fuel. He found that the F-T fuels gave lower levels of all the regulated emissions, but the Biodiesel yield higher NO<sub>x</sub> emissions than the conventional Diesel fuel. [30]

Schaberg found that HC, CO, NO<sub>x</sub> and PM emissions were significantly lower (30-40%, 40-60%, 10-40% and 40-60%, respectively) in heavy-duty engine tests with a natural gas-derived F-T fuel produced by Sasol in South Africa than with a US 2-D grade Diesel fuel. He also noted that the fuel consumption was slightly lower with the F-T fuel. [31,32]

Schaberg also compared European EN 590 Diesel with the Sasol-produced GTL fuel and concluded that HC and CO emissions were 90% lower, PM emissions 30% lower and CO<sub>2</sub> emissions lower with the GTL fuel, which the cited author attributed to its lower carbon content. Moreover, no significant difference was found between the fuels in terms of spray tip penetration, but the liquid penetration was found to be shorter with the GTL fuel, which was attributed to its lower final boiling point. [33]

Shell F-T Diesel and the Philips Chemical Company refined standard Diesel fuel were compared by McMillian. He concluded that NO<sub>x</sub> emissions and noise were lower with the F-T fuel, due to the lower heat release rate in the premixed combustion phase associated with the shorter ignition delay, while the C/H ratio was lower and the final boiling point lower due to its lower aromatic contents. He also noted that the F-T fuel gave a shorter burn duration than the standard Diesel fuel and that cetane number and density had the strongest effects on HC and CO emissions formation. The soot emissions were measured by opacity and were significantly lower for the F-T fuel. [34]

In 2001, Johnsson investigated F-T naphtha fuel, a fuel containing C<sub>5</sub> to C<sub>15</sub> molecules, an F-T fuel with a typical 160-370°C Diesel boiling range, and a US 2-D Diesel as a reference fuel. The F-T naphtha gave lower particulate matter emissions and lower NO<sub>x</sub> emissions compared to the reference fuel. However, the HC emissions were higher using F-T naphtha due to its lower cetane number. With the typical F-T fuel, emissions of HC, CO, NO<sub>x</sub> and PM were lower than those from the reference fuel. Moreover, no effect of final boiling point was found on the formation of PM emissions. [35]

Fanick compared F-T Diesel, Swedish low sulfur Diesel, and CARB Diesel with US 2-D Diesel in tests using both a light-duty and heavy-duty engine. He found that all regulated emissions were lower when using the F-T Diesel, Swedish low sulfur Diesel, and CARB

Diesel compared with the US 2-D Diesel. The largest reductions were recorded for the F-T Diesel. Moreover he found that the unregulated (and toxic) emissions of aldehydes and ketones were lowered when the engine was operated F-T fuel. Emissions of the greenhouse gases CO<sub>2</sub>, CH<sub>4</sub>, and N<sub>2</sub>O, were also lower for the F-T fuel compared to the US 2-D fuel, and the differences in observed CO<sub>2</sub> emissions was explained by the differences in carbon density of the fuels. In addition, particle size distribution measurements in the light-duty tests showed that most particles were in the 0.09-0.17 μm diameter class, and the F-T fuel yielded the lowest amounts of particles in the smallest measured size class (< 0.06 μm). [36]

Syntroleum GTL fuel was compared with US 2-D Diesel by Schubert in the experiments involving three different driving cycles: the federal test procedure (FTP), the highway fuel economy test (HFET) and the US06 supplemental federal test procedure (SFTP). The vehicle used in the experiments was a light-duty Diesel passenger car. In the FTP, HC and PM emissions were 25% lower, NO<sub>x</sub> emissions were 12% higher, and there were fuel consumption penalties with the GTL fuel compared to the US 2-D Diesel. In the HFET, PM emissions were 70% lower, NO<sub>x</sub> emissions were 4% higher, and again there were fuel consumption penalties with the GTL fuel. Finally in the SFTP, PM and HC emissions were 90% and 50% lower, respectively, and again there were NO<sub>x</sub> and fuel consumption penalties with the GTL fuel. [37]

Nord at Luleå University compared emissions from Swedish low sulfur Diesel and an F-T fuel named EcoPar<sup>TM</sup> in engine tests and concluded that emissions of HC were 15% lower, but NO<sub>x</sub> and CO emissions were only slightly affected, when using the F-T fuel instead of Swedish low sulfur Diesel. Analyses of specific hydrocarbon emissions also showed that acetaldehyde and benzaldehyde levels were reduced by 70% and acrolein levels by 90% when the F-T fuel was used. [38]

In a Japanese study Sasol GTL fuel was compared to both a current Japanese specification Diesel (500 ppm sulfur) and a future specification low sulfur Diesel (50 ppm sulfur) fuel. Tests were performed using both a heavy-duty and a light-duty engine. In the heavy-duty engine tests all the regulated emissions were lower, and the volumetric fuel consumption was higher, for the GTL fuel compared to the 50 ppm sulfur fuel. The GTL fuel also gave lower regulated emissions in the light-duty tests compared to the 500 ppm sulfur Diesel, although there were only small differences in HC and NO<sub>x</sub> emissions. [39]

The cetane number has been shown to affect emissions formation. Szybist, for instance, compared the NO<sub>x</sub> and CO emissions generated from fuels with different cetane number in HCCI combustion. It was concluded that NO<sub>x</sub> emissions are higher for lower cetane number (CN) fuels when timing is advanced, implying that F-T fuels are advantageous in this respect, since they usually have higher CN than conventional Diesel fuels. However, retarded combustion phasing resulted in higher CO emissions with high CN fuels, which is a weakness of the F-T fuels. [40]

A more detailed combustion analysis was performed by Lepperhoff. His investigation included EN 590 Diesel fuel and Sasol-produced GTL fuel. He concluded that the GTL fuel gave lower HC and CO emissions and higher thermal efficiency. He also found that the combustion of the pilot injection occurred earlier for the GTL fuel and resulted in a higher rate of heat-release; the main combustion occurred at the same time for both the tested fuels but the higher peak for the GTL fuel resulted in a higher efficiency. [41]

## 4.2 *The test engine*

The engine used in the engine tests at Chalmers is a Ricardo Hydra single-cylinder research engine, with the specifications shown in Table 2. It has a swept volume of 480 cm<sup>3</sup> and the same cylinder head and combustion system as a Volvo NED 5 production engine with four valves. When operating the engine in conventional Diesel combustion mode the geometrical compression ratio was 18:1 and in HCCI combustion mode the compression ratio was lowered to 15:1 by modifying of the piston.

**Table 2 Engine specification**

Engine type	Ricardo Hydra single cylinder
Displacement	480 cm <sup>3</sup>
Bore	81 mm
Stroke	92.3 mm
Cylinder head type	Volvo NED 5, 4 valves
Piston type	Toroidal bowl
Compression ratio	18 (for conventional Diesel operation) 15 (for HCCI operation)

The injection system is a Bosch common rail with a piezo injector and its specifications are listed in Table 3.

**Table 3 Injection system specifications**

Injection system	Bosch common rail
Injector type	Valve covered orifice, piezo
No. of nozzle holes	5
Nozzle hole diameter	0.158 mm
Umbrella angle	140°

### **4.3 Measuring equipment**

The emissions of HC, CO, CO<sub>2</sub>, NO<sub>x</sub>, and soot were measured with the equipment shown in Table 4.

**Table 4 Emission measuring equipment**

HC meter	J.U.M. HEATED FID-ANALYSER 3-300A
CO meter	Maihak infrared UNOR 6 N
CO <sub>2</sub> meter	Maihak infrared UNOR 6 N
CO <sub>2</sub> meter (for EGR)	Maihak infrared UNOR 6 N
NO <sub>x</sub> meter	ECOPHYSICS CLD 700 EL ht
Smoke meter	AVL 415S, variable sampling volume

The fuel consumption was measured using an AVL 730 fuel balance meter which measures the mass of fuel consumed over 99 seconds.

The in-cylinder pressure was measured (resolution 0.2 CAD) using an AVL GU12 piezo electric pressure sensor, and its signals were amplified by a Kistler amplifier. For the crank shaft position an AVL crank angle position sensor was used.

### **4.4 HCCI combustion mode**

Homogeneous charge compression ignition (HCCI) is a combustion mode in which the fuel is injected earlier in the compression stroke than in conventional Diesel combustion, to provide sufficient time for mixing in order to form a lean homogeneous mixture and thus generate lower soot emissions than in heterogeneous combustion. However, the early injection can cause advanced ignition timing and knocking can therefore be a problem. To minimize this problem, the compression ratio was lowered from 18:1 to 15:1 and only low load operational cases were investigated (2, 3 and 4 bar IMEP). Several different injection timings were also tested, with start of injection (SOI) at 70, 60, 50 and 40 CAD BTDC for 2 and 3 bar IMEP, and 70, 60 and 50 CAD BTDC for 4 bar IMEP. The injection was divided into six small injections. A typical in-cylinder pressure and needle lift trace is shown in Figure 8.

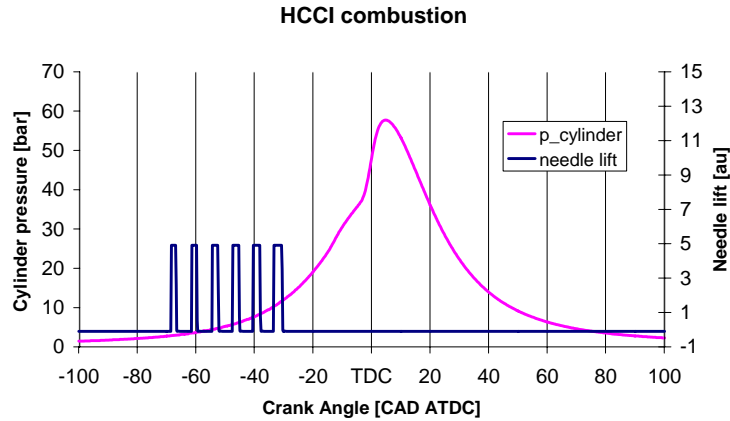


Figure 8. In-cylinder pressure and needle lift trace for HCCI combustion at 3 bar IMEP and SOI at 70 CAD BTDC

The combustion was phased so that CA50 was kept constant at TDC and the phasing was controlled by adjusting the amount of EGR.

#### 4.5 Conventional Diesel combustion mode

In conventional Diesel combustion mode the fuel is injected close to TDC. The injection is divided into a pilot injection, in which approximately 10% of the total fuel mass is injected, and a main injection. Two different loads were investigated, 3.6 and 6.6 bar IMEP. For each load two main injection and two pilot injection timings were investigated and three levels of EGR. In total, the design of experiments resulted in twelve operating points for each load. In Figure 9 the design of experiments is illustrated and in Table 5 the settings for every point can be seen.

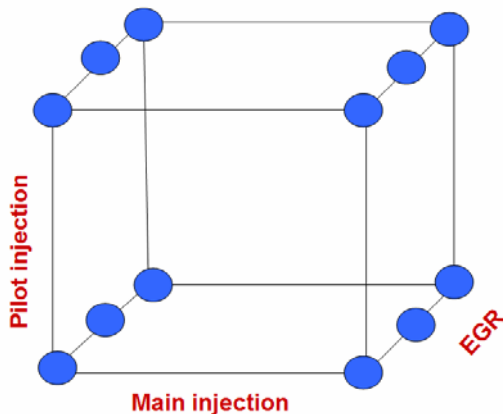


Figure 9. Design of experiments

Table 5. Operational cases in the conventional Diesel combustion mode investigations.

Load [bar IMEP]	EGR [%]	Start of pilot injection [CAD ATDC]	Start of main injection [CAD ATDC]
3.6	20	-20	-4
	24	-15	1
	30		
6.6	10	-26	-3
	16	-21	2
	20		

A typical in-cylinder pressure and needle lift trace for conventional Diesel combustion with a pilot and a main injection is shown in Figure 10 (operational point: 3.6 bar IMEP, pilot injection at 26 CAD BTDC, main injection at 4 CAD BTDC, 24% EGR).

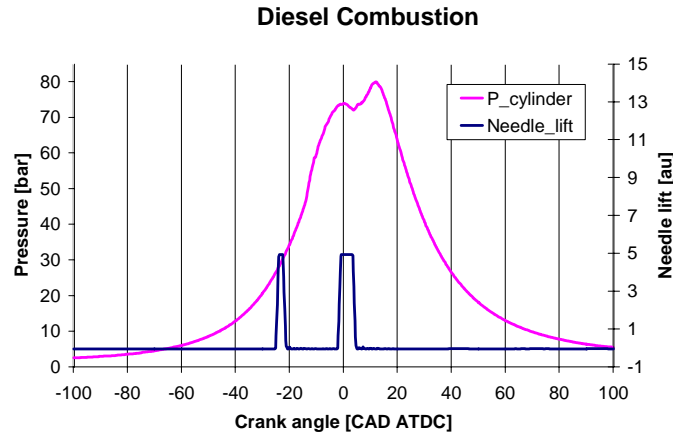


Figure 10. In-cylinder pressure and needle lift trace for conventional Diesel combustion

## 4.6 Fuels

The fuels included in the HCCI and conventional Diesel combustion investigations were Swedish low sulfur Diesel (MK1), European EN 590 (MK3) and two natural gas-derived Fischer-Tropsch fuels (FT1 and FT2). The F-T fuels were produced at a Statoil pilot plant in Trondheim, Norway. The specifications of the F-T and conventional fuels are shown in Table 6. The F-T fuels have lower densities ( $< 800 \text{ kg/m}^3$ ), lower contents of both sulfur and aromatics, higher cetane number and (thus) shorter ignition delays than MK1 and MK3. The final boiling point differs between the F-T fuels ( $297^\circ\text{C}$  for FT1, and  $341^\circ\text{C}$  for FT2). The heating values of all four fuels are very similar.

Table 6. Specification of the investigated fuels

	MK1 (Swedish low sulfur Diesel)	MK3 (European EN 590)	FT1	FT2
Density [ $\text{kg/m}^3$ ]	815	837	769	779
Sulfur [ $\text{mg/m}^3$ ]	1.4	6.5	<1	<1
Viscosity at $40^\circ\text{C}$ [ $\text{mm}^2/\text{s}$ ]	1.9	2.2	2.8	2.5
Cetane number	55	53	64	69
Aromatics[%]	5	24	<0.1	<0.1
Initial boiling point [ $^\circ\text{C}$ ]	184	184	177	183
Final boiling point [ $^\circ\text{C}$ ]	291	326	297	341
Heating value [ $\text{MJ/kg}$ ]	43.5	42.8	43.7	43.6
H/C ratio	2.02	1.87	2.15	2.14

## 4.7 Results of engine tests

This section briefly summarizes the results obtained from the HCCI and Diesel combustion tests. More detailed results can be found in Papers I and II.

### 4.7.1 Results, HCCI combustion

The mean results from the HCCI combustion investigation are presented in Figure 11, which shows that the F-T fuels gave ca. 60% lower NO<sub>x</sub> emissions than the Swedish low sulfur Diesel (MK1), which is attributed to the differences in the phasing of the combustion. The F-T fuels have higher cetane number than the conventional fuels and thus require larger amounts of EGR for a constant CA50 at TDC and EGR is known to reduce NO<sub>x</sub> emissions [2]. Moreover, the HC emissions are lower for the F-T fuels, which can be explained by their low density (especially for the FT1 fuel, the least dense of the tested fuels). No major between-fuel differences were observed in emissions of CO and CO<sub>2</sub> or fuel consumption.

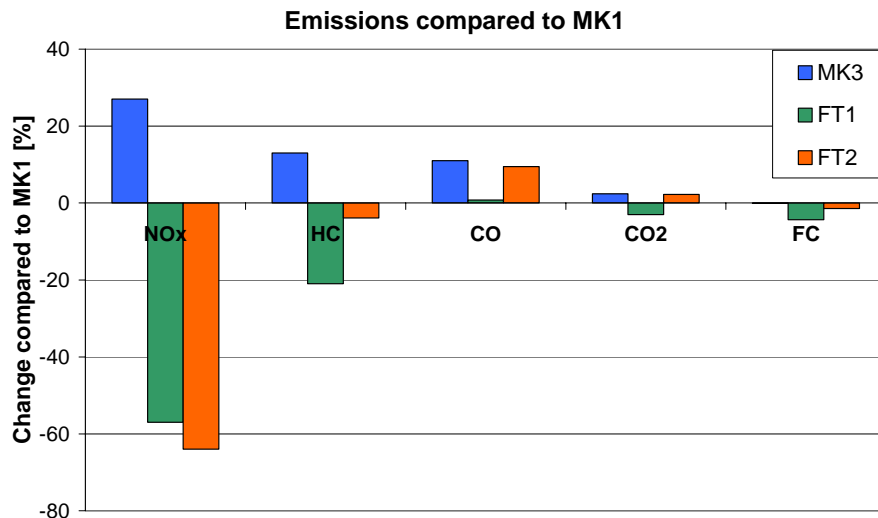


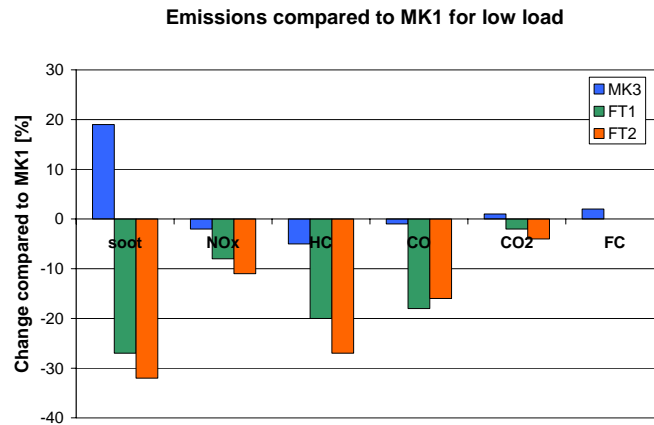
Figure 11. Emissions and fuel consumption in HCCI combustion mode. MK3, FT1 and FT2 are compared in relation to MK1.

The levels of soot generated in HCCI combustion mode were very low, below the detection limits of the measuring equipments so the fuels could not be compared in this respect.

It can be concluded that HCCI combustion is possible with F-T fuel even though they have higher cetane number than standard Diesel fuels and the charge will therefore ignite earlier, resulting in knocking at higher loads (and limiting to some extent the maximum obtainable load in HCCI mode). This limitation is minor in practice, and if the car manufacturers decide to use HCCI combustion mode in their engines, F-T fuel can still be used as an alternative fuel. Increased HC emissions is considered to be a general drawback of HCCI combustion. However with F-T fuels the HC emissions are lower than when using conventional Diesel fuels.

## 4.7.2 Results, Diesel combustion

Figure 12 illustrates some of the results obtained in the Diesel combustion tests. The graph shows average emissions obtained at all measuring points in the low load conditions. In Diesel combustion the soot emissions were high enough to be detectable by the measuring equipment so between-fuels comparisons were possible. As can be seen, the soot emissions ca. 30% lower with the F-T fuels than with the MK1 fuel, and the MK3 fuel gave the highest soot emissions. These differences are attributed to the differences in the aromatic contents of the fuels. The  $\text{NO}_x$  emissions were also lower for the F-T fuels at low load operation, while they were slightly increased at medium load (due to shorter ignition delay for these fuels associated with their higher cetane number, and thus higher in-cylinder pressures). The HC emissions were 20-30% lower for the F-T fuels, due to their shorter ignition delay and consequently lower over-leaning of the charge. Moreover the CO emissions are lower for the F-T fuels compared to the conventional fuels, and this can be explained by the differences in their fuel density, since FT1 has the lowest and MK3 the highest density. Low density leads to shorter fuel spray penetration and less fuel close to walls (less interaction) and thereby lower CO emissions. It can also be seen that there were no major differences between the fuels in terms of  $\text{CO}_2$  emissions and fuel consumption.



**Figure 12. Emissions and fuel consumption in Diesel combustion mode at low load in the tests with Data shown are levels obtained with MK3, FT1 and FT2 relative to MK1.**

## 5 Conclusions

Several alternative fuels for the Diesel engine have been developed. A very strong candidate for replacing conventional Diesel fuels is Fischer-Tropsch Diesel fuel; a synthetic fuel that can be produced from coal, natural gas or biomass.

The low aromatic contents of Fischer-Tropsch fuels contribute to low soot emissions; in our engine tests the soot emissions were up to 30% lower even than those obtained using environmentally friendly Swedish low sulfur Diesel.

The final boiling point was not, as expected, an important factor for formation of soot emissions.

The high cetane number of the tested F-T fuels shorten the ignition delay and reduce over-leaning, which results in lower HC emissions. Despite the high cetane number of the F-T fuels, HCCI combustion is possible with an increased amount of EGR.

The F-T fuels are less dense than conventional Diesel fuel, hence there are less spray-wall interactions when they are used, and consequently lower CO emissions.

Since F-T fuels have similar heating values to the conventional fuels, the specific fuel consumption is similar for all of the investigated fuels. This also means that no modification of the engine calibration is needed when using F-T fuel.

To reach the goal set by the oil Commission of reducing oil consumption in Sweden to zero by 2020 Fischer-Tropsch fuels are very good options. These fuels can be produced from several feed-stocks, existing engines and refueling infrastructure can be used, and they can be blended in any proportions with conventional Diesel fuels or Biodiesel. Most importantly; their use contributes to a cleaner environment.



## 6

## Summary of papers

### **Paper I**

“Comparison of Conventional Diesel and Fischer-Tropsch Diesel Fuels for HCCI Combustion”

In Paper I two Fischer-Tropsch fuels were compared in terms of emissions, fuels consumption and other combustion parameters in HCCI combustion mode to Swedish low sulfur Diesel and European EN590 Diesel. Since HCCI combustion results in low emissions of soot, the levels were too low to compare between the fuels. However the  $\text{NO}_x$  and the HC emissions were lower for the F-T fuels and only small differences were detected in fuel consumption between the fuels.

### **Paper II**

“An Experimental Investigation of Fischer-Tropsch Fuels in a Light-Duty Diesel Engine”

In Paper II the same fuels as in Paper I were compared (in terms of the same parameters) in conventional Diesel combustion mode. Engine settings such as pilot and main injection timing and EGR were varied to optimize the engine for low emissions and low fuel consumption. The F-T fuels showed, as expected, lower levels of soot, HC and CO emissions compared to the conventional Diesel fuels. As in HCCI combustion, there was little difference between the fuels in terms of fuel consumption in Diesel combustion mode. The ignition delay for the pilot injection was shortest for the fuel with highest cetane number and longest for the fuel with the lowest cetane number.



## 7

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# Paper I



# Comparison of Conventional Diesel and Fischer-Tropsch Diesel Fuels for HCCI Combustion

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Engine tests were performed with Swedish low sulfur Diesel, European EN 590 Diesel and two natural gas-derived Fischer-Tropsch Diesel fuels in a light-duty single-cylinder engine to compare their emission, fuel consumption, and combustion characteristics. The engine was operated in HCCI combustion mode with multiple injections and different injection timings. The combustion was phased to a constant CA50 for all operating points by adjusting the amount of EGR. Low soot emissions are expected with HCCI, and no significant differences between the various fuels were detected in this respect. However, NO<sub>x</sub> emissions were lower for the F-T fuels than the other fuels, due to their high cetane number and higher EGR requirements.

**Key Words: Fischer-Tropsch Diesel fuel, HCCI**

## 1 Introduction

The use of vehicles for transportation is constantly increasing, and almost all of these vehicles are currently driven by internal combustion engines using fuels derived from fossil oil such as Diesel or gasoline. The reasons for the popularity of combustion engines running on oil-based fuel are their high energy density, ease of handling, and well-established infrastructure. An additional advantage of fossil fuels is their relatively low price compared to alternative fuels. However, oil reserves are limited and concentrated in a few areas around the world [1]. In the near future the increased demand for these resources and the limited reserves will lead to world oil production peaking, then declining [2]. So there are likely to be severe shortages of oil in the future.

Moreover, emissions from internal combustion engines are known to have adverse effects on human health and the environment. Increasingly stringent emissions legislation has, over the years, led to the development of steadily less polluting vehicles. However, further legislative restrictions on emissions are anticipated.

A possible way to address the problems associated with limited oil reserves and to meet the even more rigorous emissions limits anticipated in the future is to develop alternative fuels and effective new combustion systems.

One possible alternative fuel is Fischer-Tropsch (F-T) fuel derived from natural gas, reserves of which are more widely spread around the world than oil reserves [1]. Another important advantage of F-T fuels is that they can be produced from coal, biomass and waste as well as from natural gas. Furthermore, these fuels can be used in contemporary diesel engines without any modifications and can be blended with conventional Diesel fuel without any complications. A further advantage of F-T fuels is that their properties can be adjusted to suit Diesel combustion by selecting appropriate catalysts for the Fischer-Tropsch process and by modifying the product upgrade processes. However, several challenges must be surmounted before Fischer-Tropsch fuels can be commercially produced. One of the challenges is to develop economically viable manufacturing processes, yielding F-T fuels with prices comparable to those for Diesel and gasoline fuels derived from crude oil.

The most fuel-efficient engine today is the Diesel engine, due to its high compression ratio. However, conventional Diesel combustion is known to produce large amounts of nitrogen oxides (NO<sub>x</sub>) and particulate matter (PM). These emissions can be reduced by controlling the equivalence ratio and in-cylinder temperature, avoiding soot and NO<sub>x</sub> formation zones. One way to accomplish this is to use the so-called Homogeneous Charge Compression Ignition (HCCI) combustion mode, where the fuel is injected early to

improve the mixing of fuel and air, ideally creating a nearly homogeneous mixture.

Experiments have shown that replacing Diesel fuel by F-T fuel in conventional Diesel engines can reduce exhaust emissions (by around 30 percent, for soot emissions [3, 4]). The main reason for this is the lower aromatic content of F-T fuels. The final boiling point has also been shown to be an important parameter for soot formation [5]. Furthermore, use of F-T fuels slightly reduces  $\text{NO}_x$  emissions, due to their high cetane numbers [6, 7].

The purpose of this study was to investigate the effects of using F-T fuel in an HCCI combustion system on combustion and emissions formation. HCCI combustion is known to produce very low soot emissions compared to conventional Diesel combustion [8]. However, a potential disadvantage of F-T fuel in this respect is its high cetane number, which reduces the ignition delay and complicates attempts to position the combustion (CA50) correctly, and thus may limit the scope to develop F-T HCCI combustion systems.

The studies presented here included engine tests with two different F-T fuels produced from natural gas. These fuels were compared to two conventional Diesel fuels (Swedish low sulfur Diesel and European EN 590 Diesel), in terms of the emissions, fuel consumption and combustion parameters obtained when they were combusted in the same engine in HCCI mode.

## 2. Experimental Equipment

### 2.1 Engine setup

Experiments were performed in a single-cylinder engine equipped with a common rail injection system. The specifications of the engine and the injection system are presented in Tables 1 and 2, respectively.

Table 1. Engine specification

Engine type	Ricardo Hydra single cylinder
Displacement	480 cm <sup>3</sup>
Bore	81 mm
Stroke	92.3 mm
Cylinder head type	Volvo NED 5, 4 valves
Piston type	Toroidal bowl
Compression ratio	15 (for HCCI)

Table 2. Injection system specification

Injection system	Bosch common rail
Injector type	Valve covered orifice, piezo
No. of nozzle holes	5
Nozzle hole diameter	0.158 mm
Umbrella angle	140°

### 2.2 Measuring equipment

The measuring instruments included emission analyzers, fuel balance, and a cylinder pressure transducer, see Table 3.

Table 3. Measuring equipment

HC meter	J.U.M. VE7 FID
CO meter	Maihak infrared
CO <sub>2</sub> meter	Maihak infrared
CO <sub>2</sub> meter (for EGR)	Maihak infrared
NO <sub>x</sub> meter	Ecophysics CLD 700 EL ht
Smoke meter	AVL 415S, variable sampling volume
Fuel flow meter	AVL 727
In-cylinder pressure sensor	AVL GU12
Data acquisition system	Sensetech

## 3. Fuels

The fuels tested in the experiments were two different Fischer-Tropsch diesel fuels (FT1 and FT2) and two conventional Diesel fuels; Swedish low sulfur Diesel fuel (mk 1) and European EN 590 diesel fuel (mk 3). The specifications of the fuels are presented in Table 4.

Table 4. Fuel specifications

	mk 1 (Swedish low sulfur Diesel)	mk 3 (European EN 590 Diesel)	FT1	FT2
Density [kg/m <sup>3</sup> ]	815	837	769	779
Sulfur [mg/m <sup>3</sup> ]	1.4	6.5	<1	<1
Viscosity at 40°C [mm <sup>2</sup> /s]	1.9	3.3	2.8	2.5
Cetane number	55	53	64	69
Aromatics [%]	5	24	<0.1	<0.1
Initial Boiling Point [°C]	184	184	177	183
Final Boiling Point [°C]	291	326	297	341
Heating value [MJ/kg]	43.5	42.8	43.7	43.6
H/C ratio	2.02	1.87	2.15	2.14

4. Experiments

The engine was operated at three different low loads: 2, 3, and 4 bar IMEP. The engine speed was kept constant at 2000 rpm in all the tests. The Start of Injection (SOI) was varied at 70, 60, 50, and 40 crank angles degrees before top dead center (CAD BTDC). Six injections were used in each experiment and the dwell time between the injections was 7 CAD for the two early injection timings (70 and 60 CAD BTDC) and 5 CAD for the two later injection timings (50 and 40 CAD BTDC.) The crank angle where 50% of the heat was released (CA50) was kept constant at TDC for all the tests and was controlled by adjusting the level of exhaust gas recirculation (EGR.). Experiments with each permutation of fuel and operating case (Table 5) were repeated three times, and exhaust emission, fuel consumption and cylinder pressure data were collected in each experiment, then mean values and errors for the key combustion parameters were calculated.

Table 5. Summary of operating conditions.

Measuring point	Load [bar]	SOI [CAD BTDC]	no. injections	dwell time	CA50
1	2	40	6	5	TDC
2		50	6	5	TDC
3		60	6	7	TDC
4		70	6	7	TDC
5		40	6	5	TDC
6		50	6	5	TDC
7	3	60	6	7	TDC
8		70	6	7	TDC
9		50	6	5	TDC
10	4	60	6	7	TDC
11		70	6	7	TDC

A cylinder pressure trace, and the fuel injection schedule, used for the 3 bar IMEP, 70 CAD BTDC SOI operational point are illustrated in Figure 1.

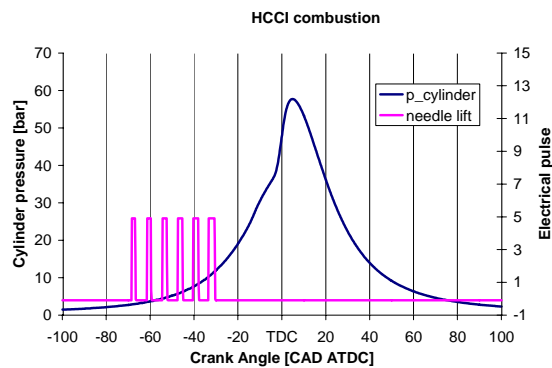


Fig.1. Cylinder pressure and needle lift traces for HCCI combustion at 3 bar IMEP and SOI at 70 CAD BTDC.

5. Results and Discussion

5.1 Emissions

The emissions obtained using each of the fuels, relative to mk 1, at the operating point 3 bar IMEP and SOI at 40 CAD BTDC, are presented in Figure 2. The largest differences between the fuels were found to be in the NO<sub>x</sub> emissions. This can be explained by the method used to control the HCCI combustion, since the crank angle position for CA50 was kept constant at TDC by adjusting the EGR level. Since the EGR acts as a diluent in the unburned gas mixture, high levels of EGR reduce the peak burned gas temperature, and thus reduce NO<sub>x</sub> formation [9].

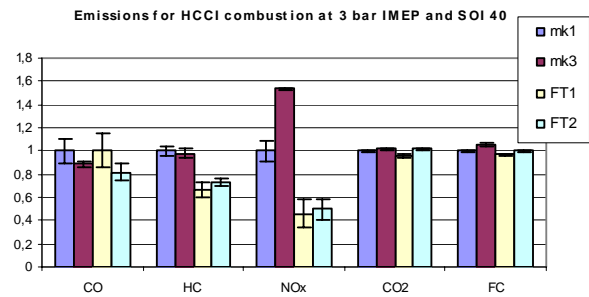


Fig.2. Emissions and fuel consumption obtained using each of the tested fuels, in relation to mk 1, with the engine operating at 3 bar IMEP and SOI at 40 CAD BTDC.

Plots of the NO<sub>x</sub> emissions versus EGR rate are shown in Figure 3. The positions of the measuring points for SOI at 40 and 60 CAD BTDC are indicated, and it can be seen that as the rate of EGR increases NO<sub>x</sub> emissions decrease. The reason why lower levels of EGR are required for mk 3 (and, to a lesser degree for mk 1) to keep CA50 at TDC than for the Fischer-Tropsch fuels is because they have lower cetane numbers. Fuels with high cetane numbers ignite more readily than fuels with lower cetane numbers, and hence require more EGR to maintain the same CA50.

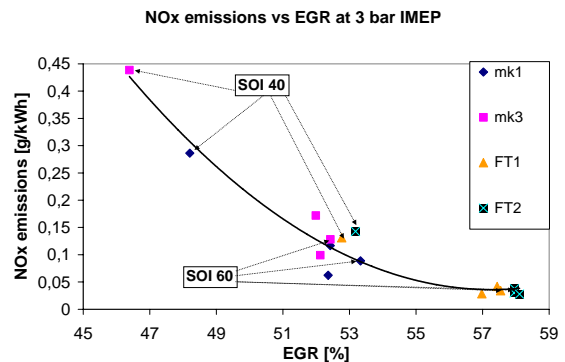


Fig.3. NO<sub>x</sub> emissions versus EGR rate.

As shown in Figure 2, there were also notable differences in HC emissions between the fuels, suggesting that there were also significant differences in their combustion efficiency. This was confirmed by calculating combustion efficiencies for each of the fuels based on the HC and CO emission data and the respective air:fuel ratios, using equation 1 [9]:

$$\eta_c = 1 - \frac{\sum_i x_i \cdot Q_{HV_i}}{(\dot{m}_f / (\dot{m}_f + \dot{m}_a)) \cdot Q_{HV_f}} \quad (1)$$

The calculated efficiencies are shown in Figure 4, where it can be seen that the Fischer-Tropsch fuels have slightly higher combustion efficiencies than both Mk 1 and Mk 3.

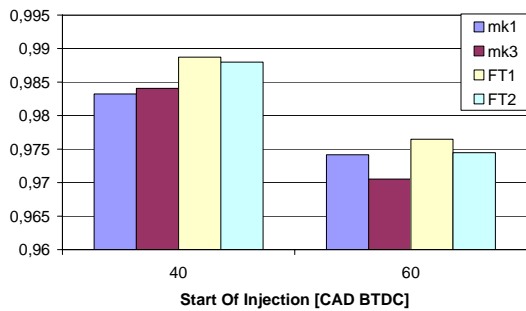


Fig.4. Combustion efficiencies calculated from exhaust emissions for each of the test fuels

The fuel consumption is similar for each of the fuels, but small differences can be seen that are mainly attributable to differences in their heating values, see Figure 2 and Table 3.

Similar trends for emissions were observed with the SOI at 60 CAD BTDC, as can be seen in Figure 5. The combustion efficiency is lower with earlier injection because crevices make a greater contribution to hydrocarbon emissions.

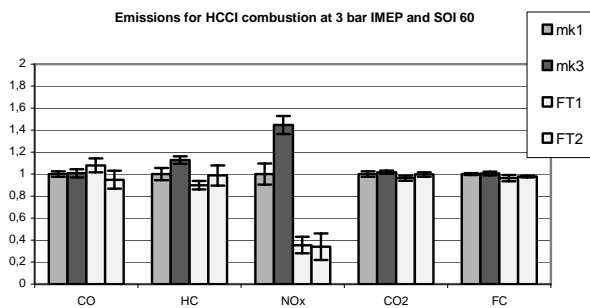


Fig.5. Emissions and fuel consumption for the tested fuels, in

relation to mk 1, with the engine operating at 3 bar IMEP and SOI at 40 CAD BTDC.

The main reason for adopting HCCI combustion is that it generates lower soot emissions, usually close to zero, than conventional Diesel combustion [10]. Figure 6 presents the soot emissions obtained at the 3 bar IMEP operating points with each of the tested SOI. The larger amounts of soot found with the SOI at 40 CAD BTDC is due to the late injection (in HCCI combustion mode) resulting in a less homogeneous charge compared to earlier injection timings. Fischer-Tropsch fuels are known to generate significantly lower soot emissions when used in conventional Diesel combustion, due to their low aromatic contents [3]. The increased soot emissions for Fischer-Tropsch fuels found with the 40 CAD BTDC SOI injection timing are probably due to the higher EGR rate and consequently lower temperatures and reduced soot oxidation. In Figure 3 it can be seen that the EGR rate is significantly lower for mk 1 and mk 3 compared to FT1 and FT2 when SOI was at 40 CAD BTDC. However, the measured soot levels were very low; below the accuracy limits of the instrument used to measure them.

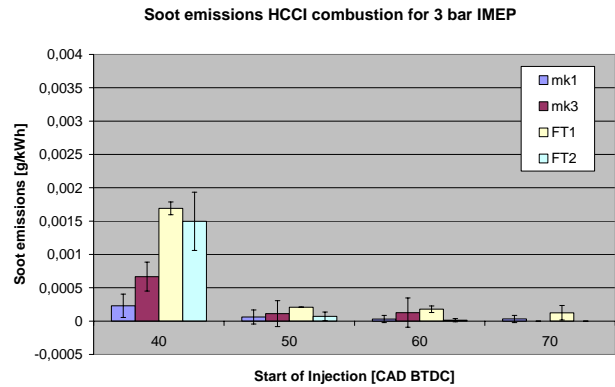


Fig.6. Soot emissions obtained at 3 bar IMEP with each of the tested fuels and start of injection timings

## 5.2 Combustion analysis

The combustion was phased to yield 50 percent of released heat at TDC. Figure 7 shows the accumulated heat release curves and Figure 8 the rate of heat release curves obtained with each of the fuels.

The combustion efficiency is, as expected, lower for HCCI combustion than for Diesel combustion, as can be seen in Figure 4.

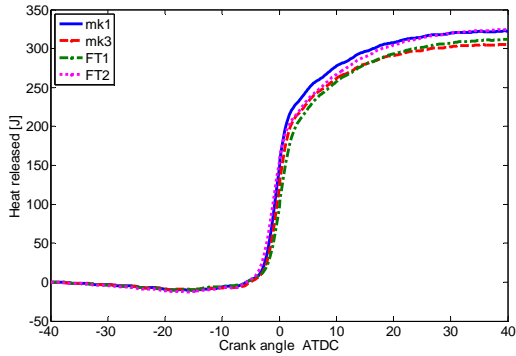


Fig.7. Accumulated heat release curves obtained at 3 bar IMEP and SOI at 40 CAD BTDC for each of the tested fuels.

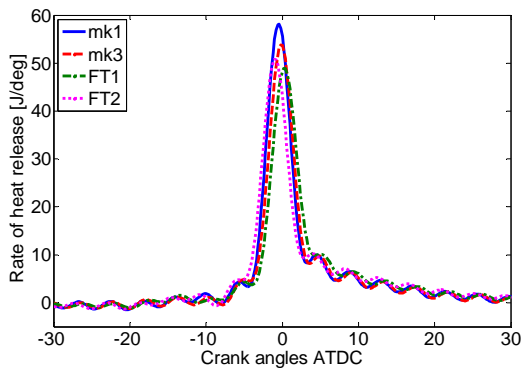


Fig.8. Rate of heat release curves obtained at 3 bar IMEP and SOI at 40 CAD BTDC for each of the tested fuels.

Heat release and rate of heat release curves obtained with SOI at 60 CAD BTDC are shown in Figures 10 and Figure 11, respectively. The main difference between the curves obtained with early (SOI at 60 CAD BTDC) and late (SOI at 40 CAD BTDC) injection is that the former resulted in low temperature reactions, at about -15 deg CA ATDC.

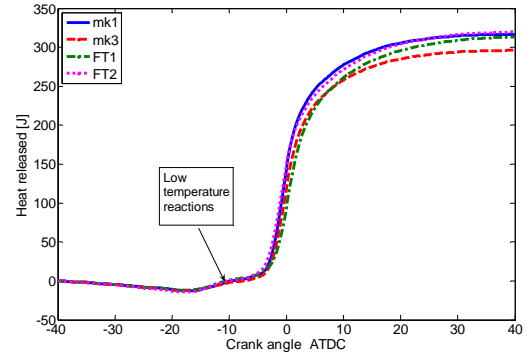


Fig.9. Heat release at 3 bar IMEP and SOI at 60 CAD BTDC.

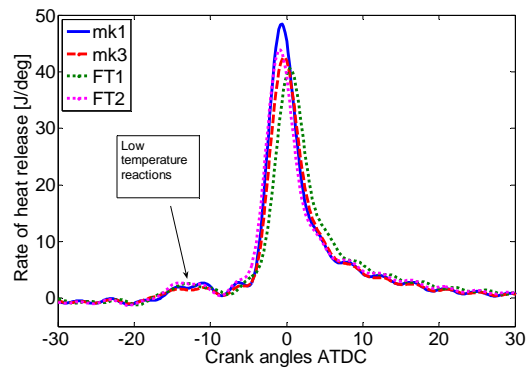


Fig.10. Rate of heat release at 3 bar IMEP and SOI at 60 CAD BTDC.

By definition, low temperature reactions (LTR) can only occur at low temperatures, which early injection yields, while with late injection in-cylinder temperatures are too high for LTR [8].

## 6. Conclusions

The following conclusions can be drawn from these experiments:

1. HCCI combustion with Fischer-Tropsch fuel is possible and results in low exhaust emissions
2. The higher cetane number of the F-T Diesel fuel requires increased levels of EGR to optimize combustion phasing.
3. The increased EGR levels result in lower NO<sub>x</sub> emissions.
4. The high cetane numbers of F-T Diesel fuels reduce the maximum IMEP levels that can be reached in HCCI mode.
5. The soot emissions are nearly zero in HCCI combustion mode and are not significantly affected by the fuel specification.
6. The efficiency, and hence fuel consumption, is similar for F-T Diesel fuel to that of Swedish low sulfur Diesel and slightly lower than that of the European fuel due to its slightly lower combustion efficiency and lower heating value.

## 7. Acknowledgements

The authors would like to thank the Swedish Road Administration and the following companies for their financial and technical support; GM Powertrain, Scania, Statoil, Volvo Powertrain, Volvo Car Corporation and Volvo Technology. The help with the experiments Dr Arjan Helmantel, Mr. Rolf Berg and all the technical staff at the Division of Combustion and Multiphase Flow, Applied Mechanics, Chalmers University are gratefully acknowledged.

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# Paper II



# An Experimental Investigation of Fischer-Tropsch Fuels in a Light-Duty Diesel Engine

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## ABSTRACT

Experiments were performed using a Light-Duty, single-cylinder, research engine in which the emissions, fuel consumption and combustion characteristics of two Fischer-Tropsch (F-T) Diesel fuels derived from natural gas and two conventional Diesel fuels (Swedish low sulfur Diesel and European EN 590 Diesel) were compared.

Due to their low aromatic contents combustion with the F-T Diesel fuels resulted in lower soot emissions than combustion with the conventional Diesel fuels. The hydrocarbon emissions were also significantly lower with F-T fuel combustion. Moreover the F-T fuels tended to yield lower CO emissions than the conventional Diesel fuels.

The low emissions from the F-T Diesel fuels, and the potential for producing such fuels from biomass, are powerful reason for future interest and research in this field.

## INTRODUCTION

The increasing numbers of vehicles operated by internal combustion engines with crude oil derived fuels are not only damaging the environment and human health but also depleting oil reserves. The vehicles are polluting the air with combustion products and the emissions adversely affect human health, causing cardiovascular diseases, cancer, allergies, asthma and lung diseases [1].

The popularity and wide use of fuels derived from fossil oil, such as diesel and gasoline, are based on their high energy density, ease of handling, and well-established manufacturing technology. An additional advantage is their relatively low price, compared to alternative fuels. However, oil reserves are limited and concentrated in a few areas around the world [2]. In the near future the increased demand for these limited resources will lead

to world oil production peaking, and then declining [3]. So there are likely to be oil shortages in the future.

Prompted by increasingly stringent emissions legislation, global and national, various measures have been developed to reduce the pollution vehicles generate, and thus their adverse effects on human health and the environment. However, further legislative restrictions on emissions are anticipated, obliging engine manufactures to develop even less polluting vehicles, while meeting customers' demands for powerful vehicles with good fuel economy.

A possible way to address the problems associated with the limited oil reserves and to meet the even more rigorous emissions limits anticipated in the future is to develop alternative fuels. Alternative feedstocks for liquid fuel production include, biomass, waste, coal and natural gas, and in order to protect the environment, a life cycle analysis is therefore important for each fuel considered.

One possible alternative fuel is Fischer-Tropsch (F-T) fuel derived from natural gas, reserves of which are more widely spread around the world than oil reserves [2]. Another important advantage of F-T fuels is that they can also be produced from coal, biomass and waste. Furthermore, these fuels can be used in contemporary diesel engines without any modifications and can be blended with conventional Diesel fuel with minimal complication. A further advantage of F-T fuels is that their properties can be adjusted to suit Diesel combustion by selecting appropriate catalysts for the Fischer-Tropsch process and by modifying the product upgrade processes. However, several challenges must be surmounted before Fischer-Tropsch fuels can be commercially produced from biomass in Sweden. One of the challenges is to develop economically viable manufacturing processes, yielding F-T fuels with prices comparable to those for Diesel and gasoline fuels derived from crude oil [4].

The most fuel-efficient engine today is the Diesel engine, due to its high compression ratio. However, conventional

Diesel combustion is known to produce large amounts of nitrogen oxides (NO<sub>x</sub>) and particulate matter (PM). These emissions can be reduced by controlling the equivalence ratio and in-cylinder temperature, avoiding high soot and NO<sub>x</sub> formation zones. One way to accomplish this is to control the injection timing and the amount of EGR.

Experiments have shown that replacing Diesel fuel by F-T fuel in conventional Diesel engines can reduce exhaust soot emissions by around 30 percent [5, 6]. The main reason for this is the much lower, close to zero, aromatic content of F-T fuels. The final boiling point has also been shown to be an important parameter for soot formation [7]. Furthermore, use of F-T fuels has been found to slightly reduce NO<sub>x</sub> emissions, due to their high cetane number [9, 10].

Although CO and HC emissions from Diesel combustion are relatively low, compared with combustion in a gasoline engine, these emissions must not be increased when lowering NO<sub>x</sub> and PM emissions. F-T fuels could also be advantageous in this respect, since data in the literature show that F-T fuels generate relatively low CO and HC emissions [11, 12].

The purpose of this study was to investigate the effects of using F-T fuel in a Diesel engine operating in conventional Diesel combustion mode. The injection timing and amount of EGR were varied to study the effects of these variables on engine operational characteristics when using F-T fuels and to investigate if the calibration of unmodified Diesel engines can be further optimized for F-T fuels.

The studies presented here included engine tests with two different F-T fuels produced from natural gas in a pilot power plant at Statoil in Trondheim, Norway. Combustion parameters of these fuels were compared to those of two conventional Diesel fuels (Swedish low sulfur Diesel and European EN 590 Diesel).

## EXPERIMENTS

### ENGINE SETUP

The engine used for the experiments was a single cylinder, light-duty Diesel engine, the specification of which can be found in table 1. In table 2 the injection system is presented.

**Table 1:** Engine specifications

Engine type	Ricardo Hydra single cylinder
Displacement	480 cm <sup>3</sup>
Bore	81 mm
Stroke	92.3 mm
Cylinder head type	Volvo NED 5, 4 valves
Piston type	Toroidal bowl
Compression ratio	18

**Table 2:** Injection system specifications

Injection system	Bosch common rail
Injector type	Valve covered orifice, piezo
No. of nozzle holes	5
Nozzle hole diameter	0.158 mm
Umbrella angle	140°

### MEASURING EQUIPMENT

Emissions, fuel consumption and the in-cylinder pressure were measured during operation of the engine with equipment presented in table 3.

**Table 3:** Measuring equipment

HC meter	J.U.M. VE7 FID
CO meter	Maihak infrared
CO <sub>2</sub> meter	Maihak infrared
CO <sub>2</sub> meter (for EGR)	Maihak infrared
NO <sub>x</sub> meter	Ecophysics CLD 700 EL ht
Smoke meter	AVL 415S, variable sampling volume
Fuel balance	AVL 730
In-cylinder pressure sensor	AVL GU12
Data acquisition system	SenseTool SMETec Combi

### FUELS

The fuels included in the investigation were two Fischer-Tropsch fuels (FT1 & FT2). The reference fuels were Swedish low sulfur Diesel and European EN 590 Diesel. The specifications of the F-T fuels and the reference fuels are presented in table 4.

**Table 4:** Fuel specifications

	MK 1 (Swedish low sulfur Diesel)	MK 3 (European EN 590 Diesel)	FT1	FT2
Density [kg/m <sup>3</sup> ]	815	837	769	779
Sulfur [mg/m <sup>3</sup> ]	1.4	6.5	<1	<1
Viscosity at 40°C [mm <sup>2</sup> /s]	1.9	2.2	2.8	2.5
Cetane number	55	53	64	69
Aromatics [%]	5	24	< 0.1	< 0.1
Initial Boiling Point [°C]	184	184	177	183
Final Boiling Point [°C]	291	326	297	341
Heating value [MJ/kg]	43.5	42.8	43.7	43.6
H/C ratio	2.02	1.87	2.15	2.14

The aromatic contents are lower and thus the H/C ratios are higher for the F-T fuels than for MK1 and MK3. The density of the fuel also depends on the H/C ratio, and it can be seen that the both F-T fuels have densities lower than 800kg/m<sup>3</sup>.

The FT1 and FT2 fuels have higher cetane numbers than the MK1 and MK3 fuels, and hence are likely to have shorter ignition delays.

The FT2 fuel has a higher final boiling point than the FT1 fuel.

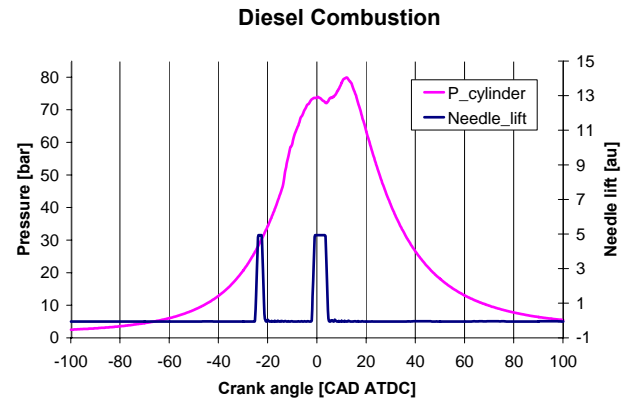
## EXPERIMENTS

The Engine was operated in conventional Diesel combustion mode with standard operational settings (calibration) for a Volvo Cars production engine, except for injection timing and EGR levels, which were varied around the standard settings. The test cases were representative for the European Driving Cycle, EUDC. The parameters were investigated at two loads (3.6 and 6.6 bar IMEP) at a constant engine speed of 2000 rpm.

A pilot injection, in which 9% of the total injected fuel mass was injected, was used in all of the experiments. In Figure 1, the injection timing and pressure trace obtained for one operational case are presented.

The amounts of EGR applied were 20, 25 and 30% in the low load cases and 10, 15 and 20% in the medium load cases. The amount of EGR was calculated with equation (1) [13].

$$EGR[\%] = 100 \cdot \frac{CO_{2,intake}}{CO_{2,exhaust}} \quad (1)$$



**Figure 1:** Pressure trace and injection timing for the operational point 6.6 bar IMEP, pilot inj. at -26 and main inj. at -3 CAD ATDC.

The starts of the pilot and main injection were set to -20 or, -15 and -4 or, 1 CAD ATDC respectively, for the low load experiments and to -26 or, -21 and -3 or, 2 CAD ATDC, respectively for the medium load experiments. The standard settings was, for the low load, pilot injection at -20 and main injection -4 CAD ATDC and EGR 24 % and for the high load pilot injection at -26 and main injection -3 CAD ATDC and EGR 16 %.

Table 5 and 6 presents the operating points with EGR levels and injection timing, for low load and medium load respectively. In total, 24 measuring points were investigated and experiments were repeated three times in order to estimate standard deviations.

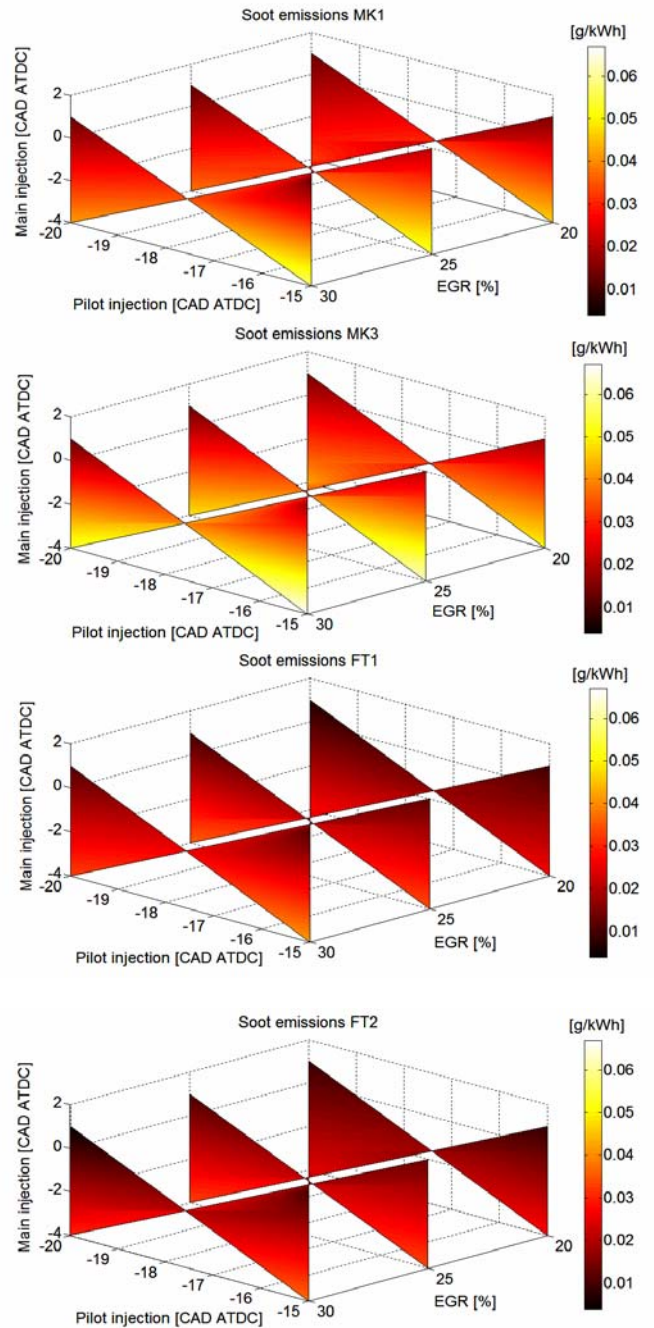
Unfortunately, however, the available quantity of FT2 was only sufficient to cover the low load test cases.

**Table 5:** Operating points at low load.

Operating point	Pilot injection [CAD ATDC]	Main injection [CAD ATDC]	EGR [%]
1	-20	-4	20
2	-20	1	20
3	-15	-4	20
4	-15	1	20
5	-20	-4	24
6	-20	1	24
7	-15	-4	24
8	-15	1	24
9	-20	-4	30
10	-20	1	30
11	-15	-4	30
12	-15	1	30

**Table 6:** Operating points at medium load.

Operating point	Pilot injection [CAD ATDC]	Main injection [CAD ATDC]	EGR [%]
13	-26	-3	10
14	-26	2	10
15	-21	-3	10
16	-21	2	10
17	-26	-3	16
18	-26	2	16
19	-21	-3	16
20	-21	2	16
21	-26	-3	20
22	-26	2	20
23	-21	-3	20
24	-21	2	20



**Figure 2:** Soot emissions at 3.6 bar IMEP.

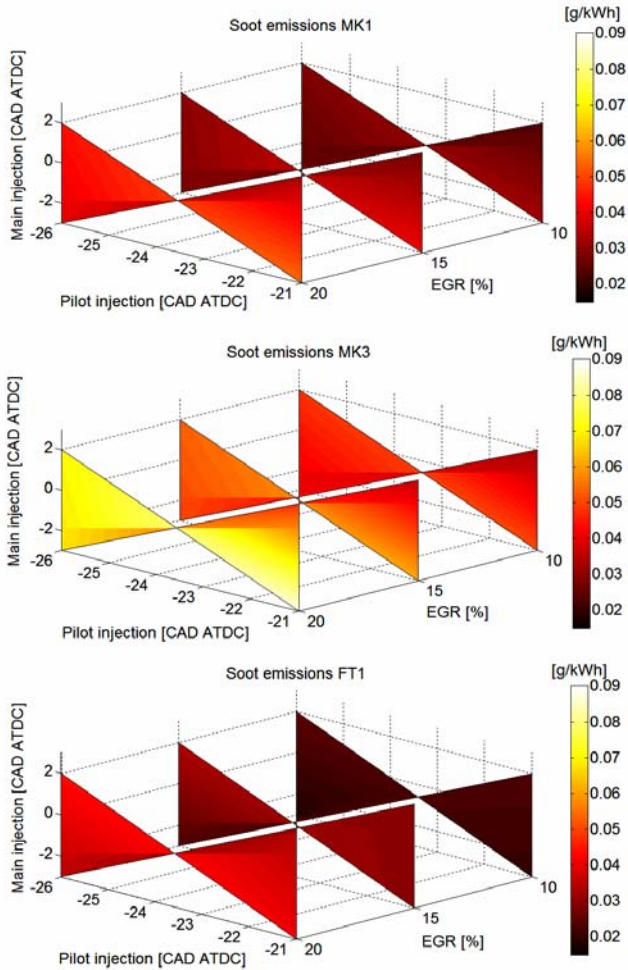
## RESULTS AND DISCUSSIONS

The emissions and fuel consumptions are presented in figures for every fuel considered. The corners in the plots represent the twelve operating points. And linear relations between the measuring points is estimated in order to obtain a comprehension of the trends.

### SOOT EMISSIONS

The soot emissions obtained with each of the tested fuels in the various operational cases are presented in Figures 2 and 3.

The results clearly indicate that F-T fuels produce lower soot emissions than the reference fuels, MK1 and MK3. Moreover, MK3 produces larger amounts of soot emissions than MK1. The soot emissions are strongly dependent on the amount of aromatics in the fuel and accordingly table 4, MK3 has the highest content of aromatics and the F-T fuels the lowest.

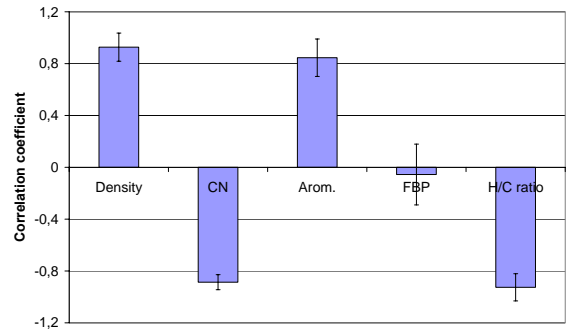


**Figure 3:** Soot emissions at 6.6 bar IMEP

An analysis to assess the significance of some of the fuel parameters was performed and the results are presented in Figure 4, which shows the linear relationships between soot emissions and the fuels' density, cetane number, aromatic content, final boiling point and H/C ratio. The error bars represent the standard deviations of the relationships, based on the twelve measuring points.

It can be seen that the final boiling points of the tested fuels do not have a major impact on the soot emissions. This is inconsistent with results presented by Kono N. et al [14], who found that as the final boiling point was decreased by 10°C the PM emissions declined by 3-7%. However, the aromatic content of their fuel was higher than that of the fuels tested in this study.

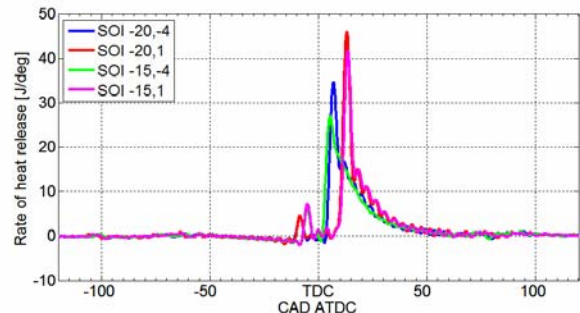
The density, cetane number and aromatic content of the fuels seemed to be the major parameters affecting soot emission formation in these tests. These three parameters are also inter-dependent since a fuel with a low H/C ratio will have high aromatic contents, low cetane number and high density.



**Figure 4:** Correlations between soot emissions and: density, cetane number, aromatic content, final boiling point, and H/C ratio.

Analysis of the impact of injection timing on the soot emissions, showed that retarding the pilot injection increases soot emissions slightly for both engine loads, see Figures 2 and 3. This is because retarding the pilot injection shortens the ignition delay and reduces the premixed peak in heat release diagram, see figure 5. This is the result of less time for mixing but it also leads to a decrease in maximal global in-cylinder temperature and thereby less soot oxidation and the soot emissions are therefore slightly increased. In contrast, retarding the main injection results in lower soot emissions because as can be seen in Figure 5, it increases the ignition delay and the time available for fuel-air mixing. More of the fuel will thus be burned in partly premixed conditions, resulting in less soot formation. This is generally applicable to all of the fuels at both of the tested loads. This is because retarding the pilot injection shortens the ignition delay and reduces the premixed peak in heat release diagram, see figure 5.

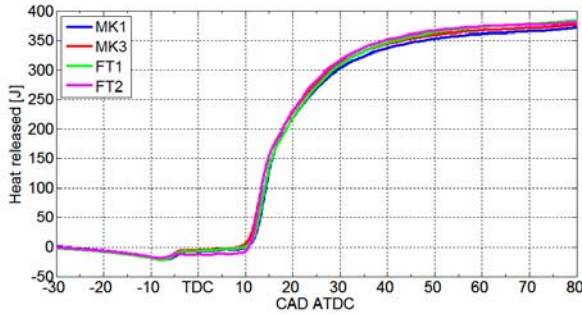
If EGR is increased, the soot emissions are also increased which can be explained by the associated reductions in in-cylinder temperatures and oxygen contents and, consequently, lower rates of soot oxidation.



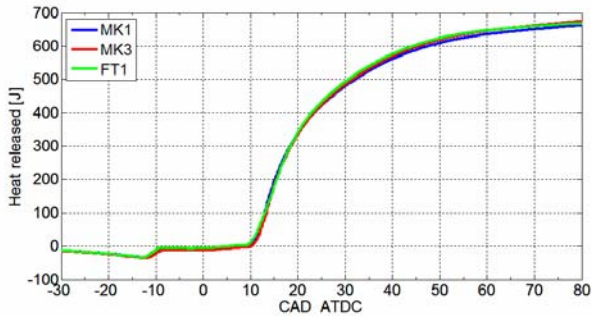
**Figure 5:** Rate of heat-release curves obtained with MK1 at 3.6 bar IMEP, with each of the tested injection timings (pilot and main CAD ATDC) and 20% EGR.

In Figures 6 and 7 heat release curves are shown for the operational points 3.6 bar IMEP, pilot injection at -15 CAD ATDC and main injection at -4 CAD ATDC and 20% EGR and 6.6 bar IMEP, pilot injection at -21 CAD ATDC, main injection at -3 CAD ATDC with 10% EGR. It

can be seen that the curves obtained with the different fuels are very similar.



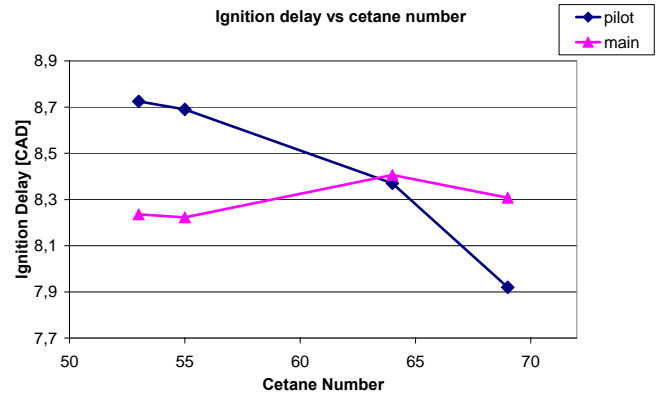
**Figure 6:** Heat release curves for the operational point 3.6 bar IMEP, pilot inj. at -15 and main inj. at 1 CAD ATDC with 20% EGR.



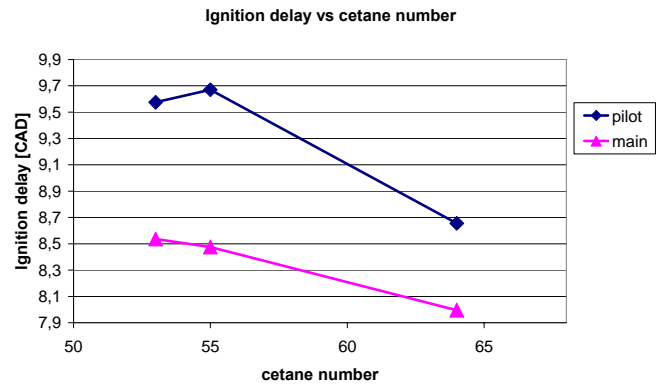
**Figure 7:** Heat release curves for the operational point 6.6 bar IMEP, pilot inj. at -21 and main inj. at 2 CAD ATDC with 10% EGR.

The ignition delays for the pilot and main injections were calculated for each of the fuels and the results are presented in Figures 8 and 9. The calculated operational points were at 3.6 bar IMEP, pilot injection at -15 CAD ATDC, main injection at -4 CAD ATDC with 20% EGR and at 6.6 bar IMEP, pilot injection at -21 CAD ATDC, main injection at -3 CAD ATDC with 10% EGR.

In general the ignition delay decreases as the cetane number increases, but the differences involved are small. The trends are illustrated in Figures 8 and 9 and it can be seen that at low load the trend is accurate for the pilot injections. However, the ignition delay for the main injection is not affected by the cetane number, due to the influence of the pilot injection. Regarding the medium load the trend can be seen for both the main and the pilot injection. For the medium load the pilot does not influence the ignition delay for the main injection.



**Figure 8:** Ignition delays for main and pilot injections with each of the tested fuels at low load.

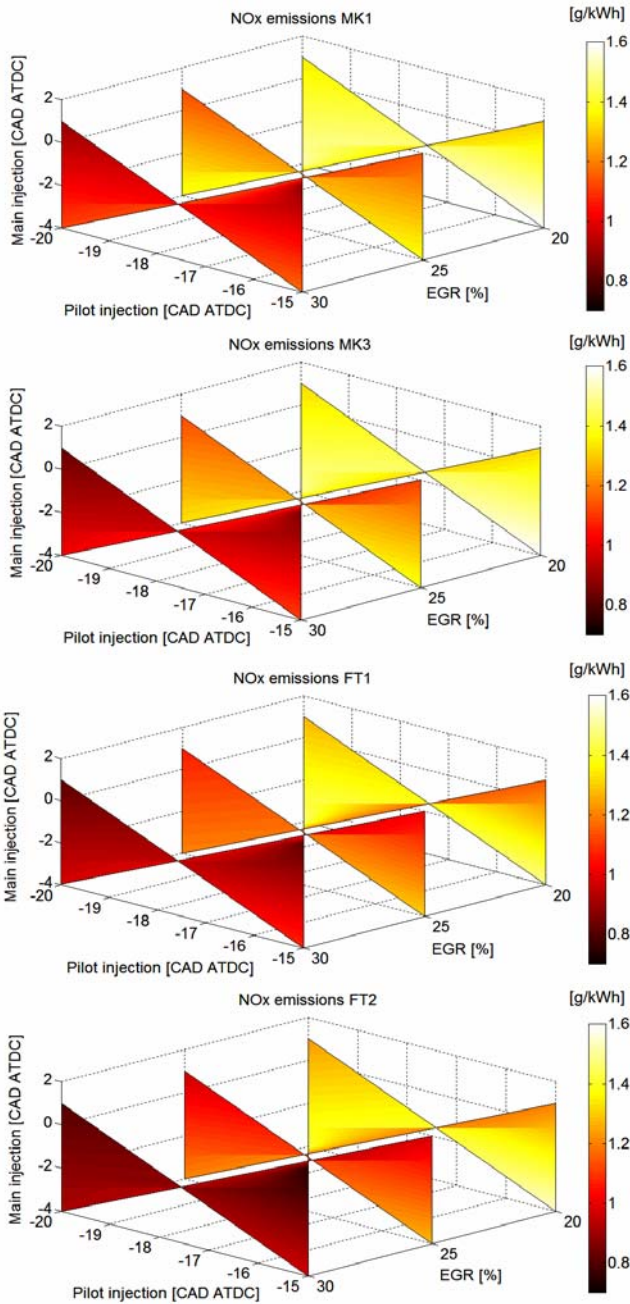


**Figure 9:** Ignition delays for main and pilot injections with each of the tested fuels at medium load.

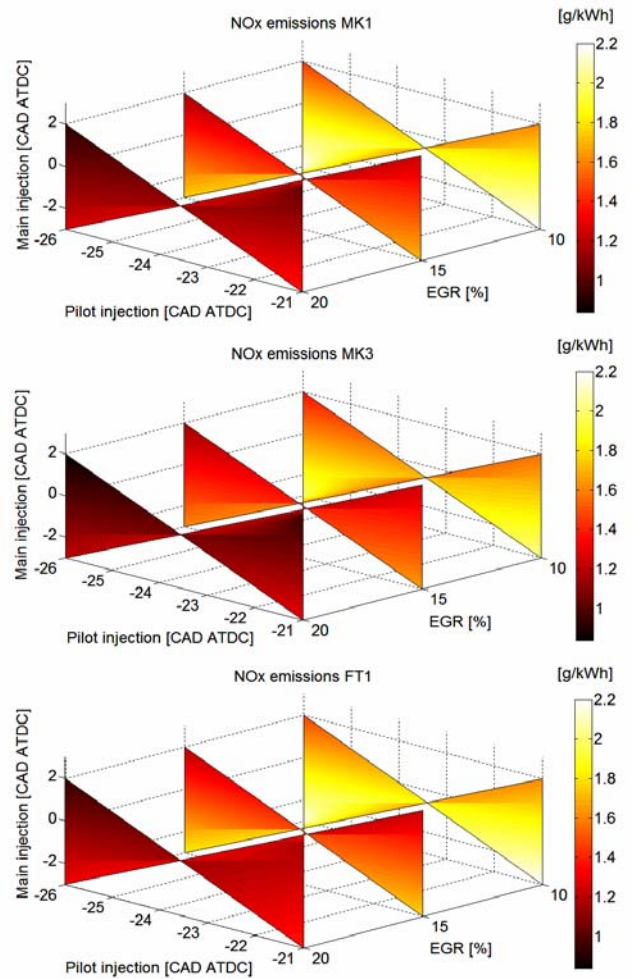
## NO<sub>x</sub> EMISSIONS

NO<sub>x</sub> formation depends mainly on the local temperature, equivalence ratio, and available time.

In Figures 10 and 11 the measured NO<sub>x</sub> emissions are shown. Only small difference between the fuels were found in this respect, although tendencies were observed for the F-T fuels to yield lower NO<sub>x</sub> emissions at light load, and higher NO<sub>x</sub> emissions at medium load, than the references fuels. Alleman et al [12] found that their high H/C ratio and the low aromatic contents also reduced the NO<sub>x</sub> emissions yielded by F-T fuels. A probable reason for the different findings in this study, at medium load, is that the shorter ignition delay obtained with the F-T fuel led to higher cylinder pressures (temperatures) and thus higher NO<sub>x</sub> formation.

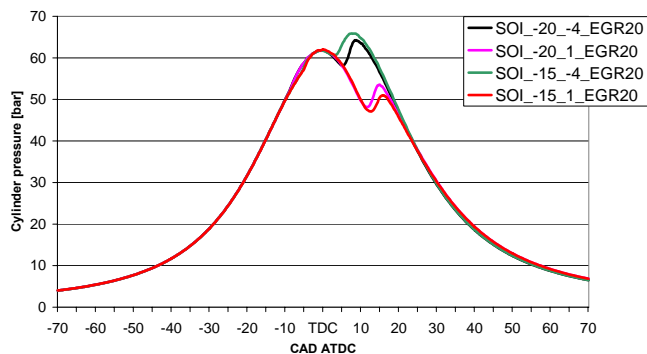


**Figure 10:** NO<sub>x</sub> emissions at 3.6 bar IMEP

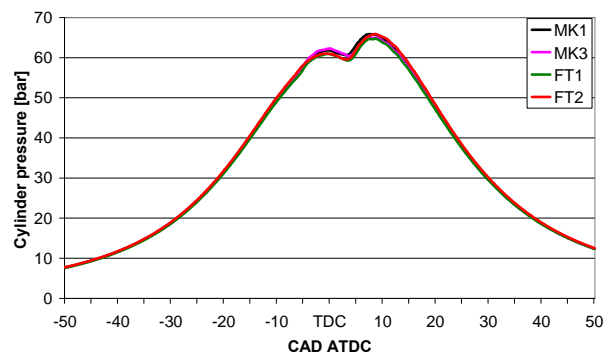


**Figure 11:** NO<sub>x</sub> emissions at 6.6 bar IMEP

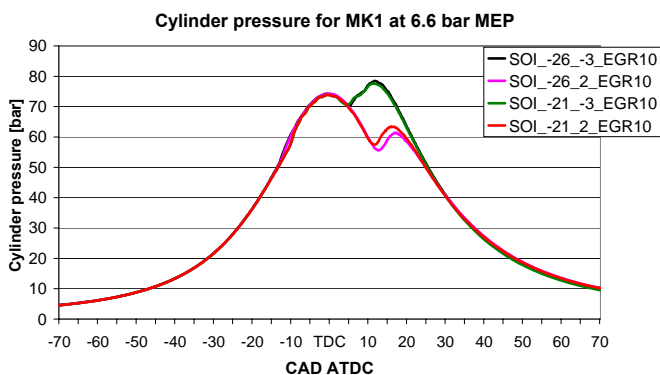
For the NO<sub>x</sub> emissions, retarding the main injection results in lower emissions while the timing of the pilot injection seems to have a minor influence, see Figures 10 and 11. The reasons for this difference can be seen in Figures 12 and 13, which show that a later main injection leads to lower cylinder pressure and thus lower in-cylinder temperatures at both engine loads. However, retarding the pilot injection seems to have different effects at the two loads. For the low load case, Figure 12, retarding it slightly increases the cylinder pressure for an early main injection while for a late main injection it has the opposite effect, decreasing the pressure. For the medium load case, Figure 13, the timing of the pilot injection does not change the cylinder pressure for an early main injection. For a late main injection retarding the pilot injection increases the cylinder pressure and NO<sub>x</sub> emissions.



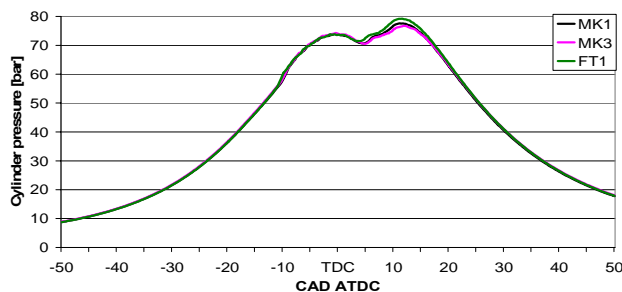
**Figure 12:** Cylinder pressure traces obtained with MK1 at 3.6 bar IMEP and each of the tested injection timings (pilot and main, CAD ATDC) with 20% EGR.



**Figure 14:** Cylinder pressure traces obtained with each of the tested fuels at the operating point 3.6 bar IMEP with pilot injection at -15 CAD ATDC and main injection at -4 CAD ATDC and EGR 20%.



**Figure 13:** Cylinder pressure traces obtained with MK1 at 6.6 bar IMEP and each of the tested injection timings (pilot and main CAD ATDC) with 10% EGR.



**Figure 15:** Cylinder pressure traces obtained with each of the tested fuels at the operating point 6.6 bar IMEP with pilot injection at -21 CAD ATDC and main injection at -3 CAD ATDC and 10% EGR.

Figure 14 shows the pressure traces obtained at the low load operational point with a pilot injection at -15 CAD ATDC and main injection at -4 CAD ATDC. The cylinder pressures obtained with the different fuels do not show any significant differences. This also holds for late main injection. For the medium load case, differences can be seen between the fuels at the operational point with a pilot injection at -21 CAD ATDC and main injection at -3 CAD ATDC, Figure 15. The F-T fuels yield the highest cylinder pressures, followed by MK1 then MK3 (scales with ignition delay.) This confirms the results shown in Figure 11, which shows that FT1 yield slightly higher  $\text{NO}_x$  emissions than MK1, which in turn yields slightly higher  $\text{NO}_x$  emissions than MK3.

The data presented in Figures 10 and 11 clearly show that increasing the EGR reduces  $\text{NO}_x$  emissions.

## HC EMISSIONS

Hydrocarbon emissions are mainly caused by over-mixing (i.e. mixing fuel with air to ratios that are leaner than the flammability limit) during ignition delay and by under-mixing (insufficient mixing of the fuel and air in the nozzle orifices) during expansion. Spray-wall interactions may also cause HC-emissions.

The HC emissions are presented in Figures 16 and 17. It can be seen that the F-T fuels yield lower hydrocarbon emissions than the conventional fuels, probably because they have higher cetane numbers, which generally reduces ignition delay, over-leaning and, thus, hydrocarbon emissions.

The lower density of the F-T fuels leads to reductions in fuel spray tip penetration, equation (2) [15, 16] and less interactions with the combustion chamber walls and thereby lower HC emissions.

$$x_{tip} = \sqrt{\frac{2 \cdot 0.4 \cdot 3 \cdot d \cdot \left(\frac{\rho_f}{\rho_A}\right)^{0.5} \cdot U \cdot t}{16 \cdot \pi^{0.5} \cdot C_t}} \quad (2)$$

Retarding the main injection increases the HC emissions by increasing the ignition delay in both load cases while retarding the pilot injection in the low load case gives slightly lower HC emissions due to the slight reduction in ignition delay it leads to, discussed above. For the high load case retarding the pilot injection leads to a slight decrease in HC emissions except for MK3 where a slight increase can be found.

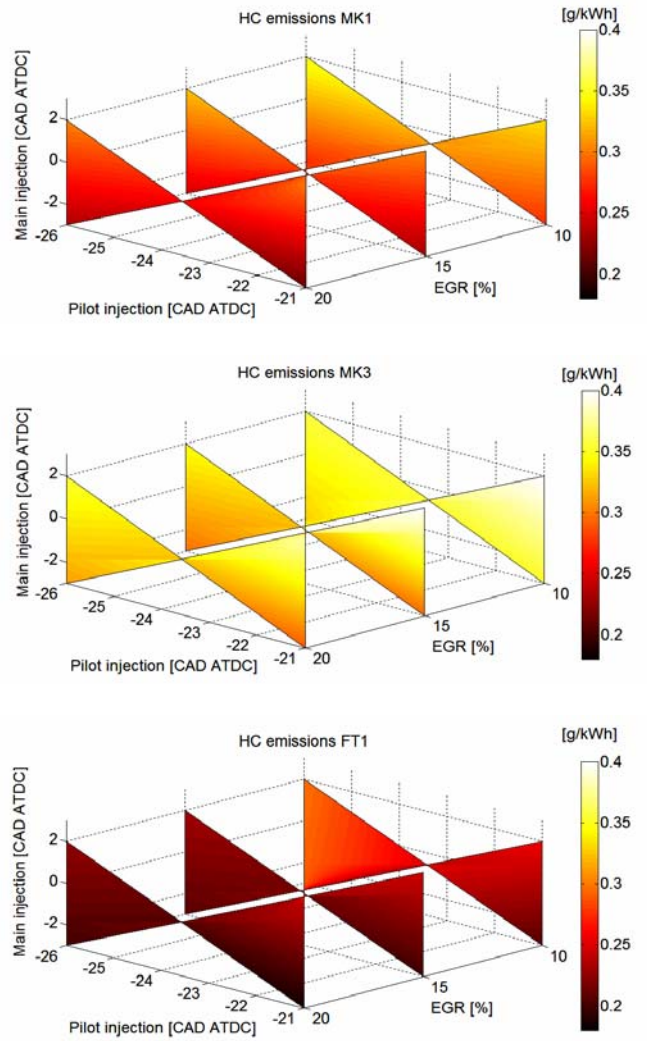


Figure 17: HC emissions at 6.6 bar IMEP.

In low load conditions increasing EGR increases HC emissions, while in the higher load conditions it reduces them.

The ignition delays observed with each of the tested amounts of EGR at the operating points with late pilot injections and early main injections are presented in Figure 18. The results show that with low EGR the ignition delay is shortest for the F-T fuel. When the amount of EGR increases, there is no apparent correlation between the fuels' cetane numbers and the ignition delay.

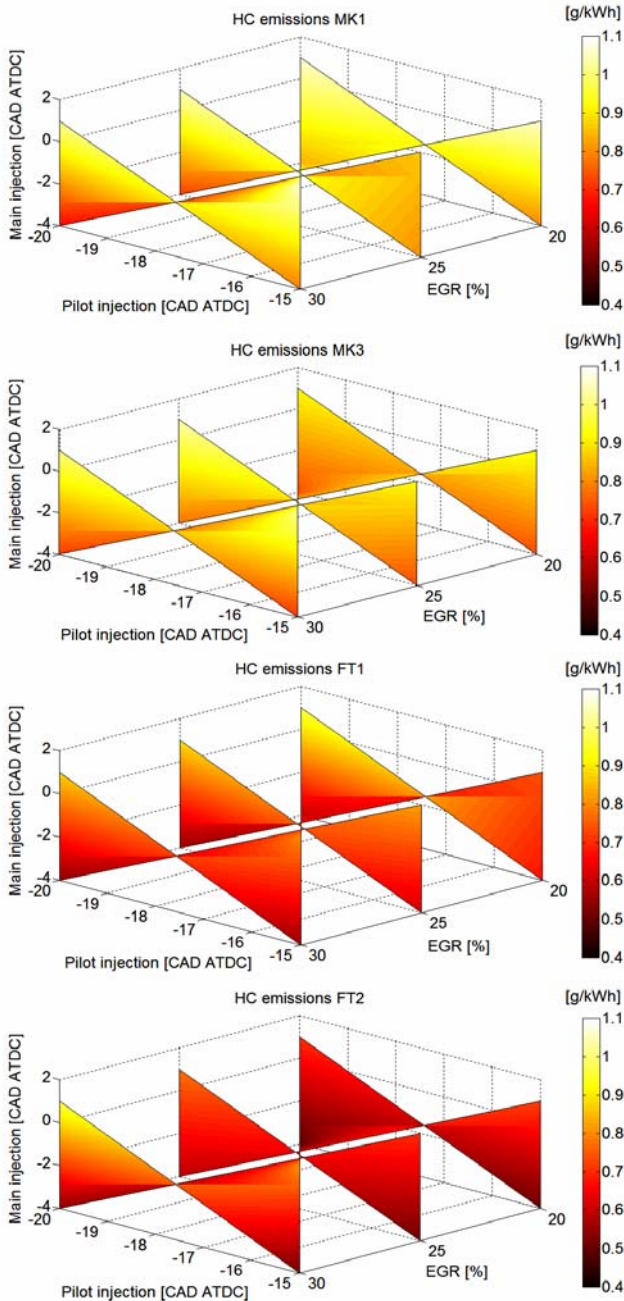
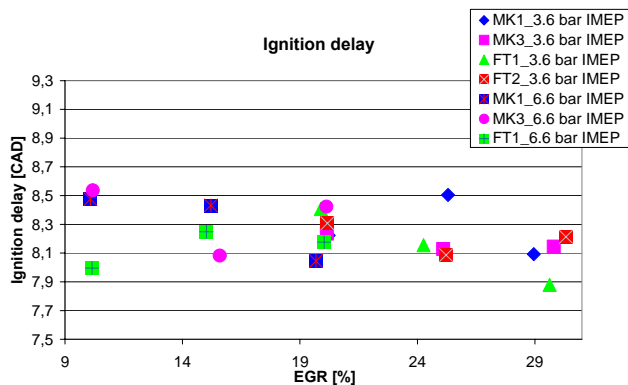
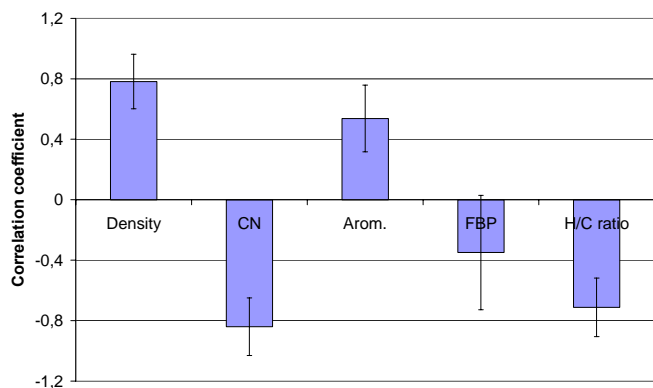


Figure 16: HC emissions at 3.6 bar IMEP



**Figure 18:** Ignition delays for each of the tested amounts of EGR at the operating points with pilot injection at -15 CAD ATDC and main injection at -4 CAD ATDC for low load and -21 CAD ATDC and main injection at -3 CAD ATDC for medium load.

To assess the impact of the fuel parameters on the HC emissions another correlation analysis was performed, the results of which are presented in Figure 19. As can be seen, the fuel parameters that most strongly affect the formation of HC emissions are the density, the cetane number and the H/C ratio, while the aromatic contents have less effect than on soot emissions.

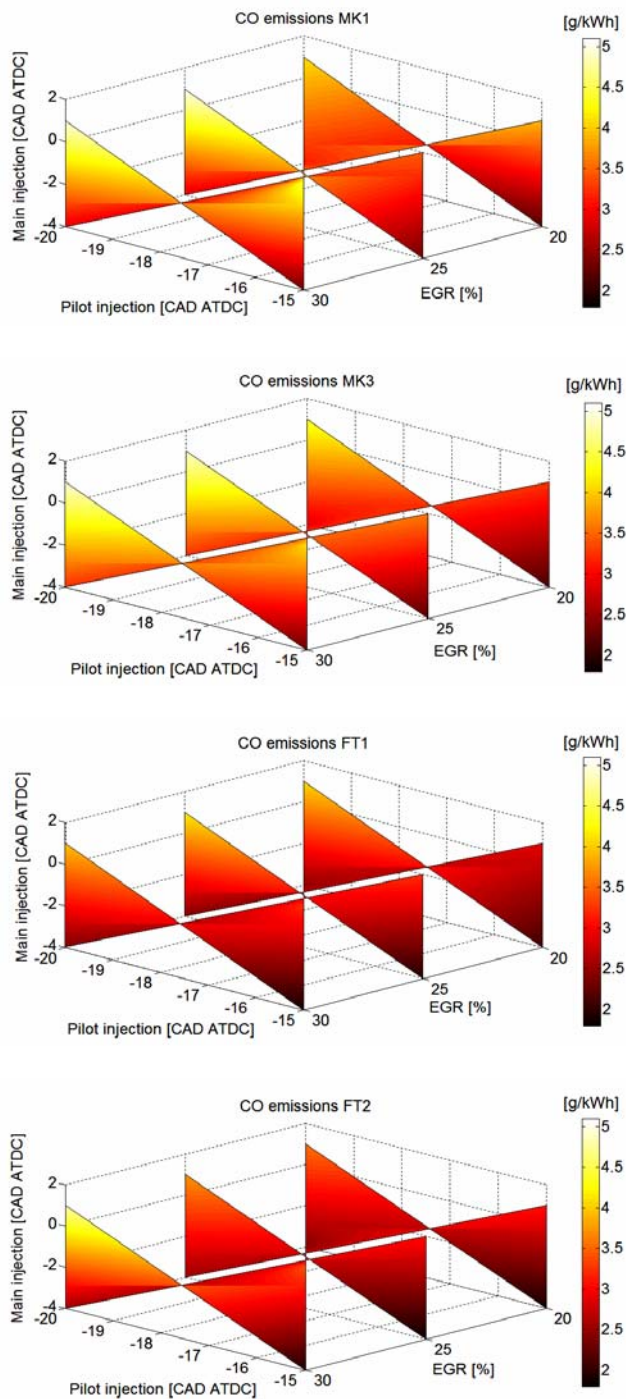


**Figure 19:** Correlation between HC emissions and: density, cetane number, aromatic content, final boiling point, and H/C ratio of the fuels.

## CO EMISSIONS

The CO emissions increase when the main injection is late, as shown in Figures 20 and 21 for both of the load cases. CO formation depends on the local equivalence ratio, pressure, local temperature in the cylinder and the time available to reach equilibrium values. Spray-wall interactions will also influence CO emissions.

The fuel effects on CO emissions can be seen in Figures 18 and 19. The MK3 fuel yields the highest CO emissions and FT1 the lowest. The likeliest explanation for this is the differences in their density, since FT1 has the lowest and MK3 the highest density. Low density leads to shorter fuel spray penetration and less fuel close to walls [15, 16].

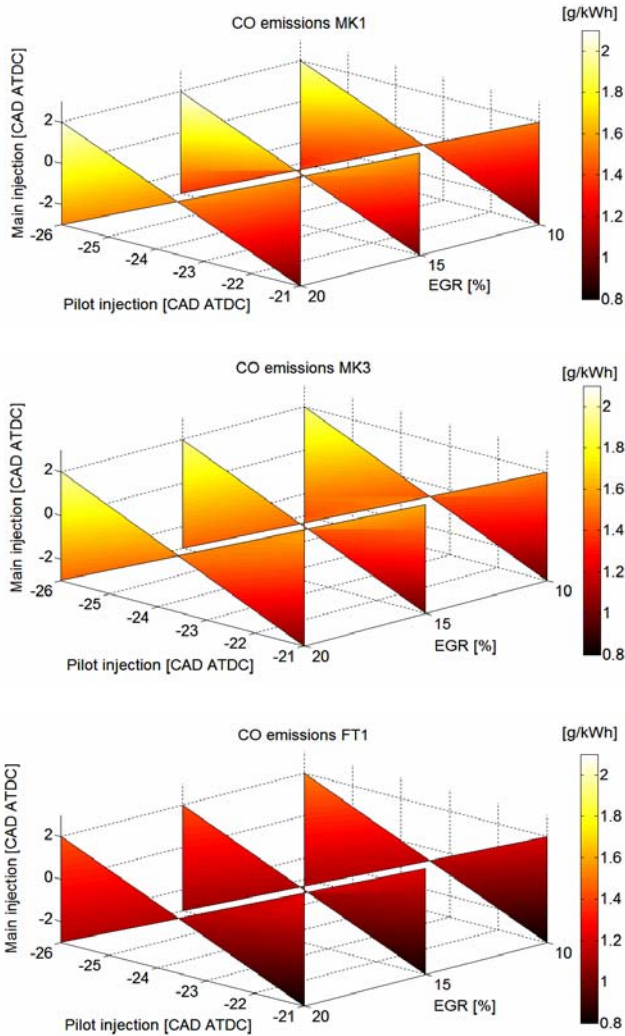


**Figure 20:** CO emissions at 3.6 bar IMEP

When the main injection is late both the cylinder pressure, (cf. Figure 10) and the global temperature are lower. The ignition delay is also increased and in combination with the late injection less time is available for combustion products-air mixing before the CO oxidation freezes at 1500 K [17]. The oxidation of CO to CO<sub>2</sub> is temperature-dependent, and when the temperature is lower the oxidation rate slows down, therefore retarding the main injection leads to higher CO emissions for both test cases. A late pilot injection, on the other hand, reduces the CO emissions because of

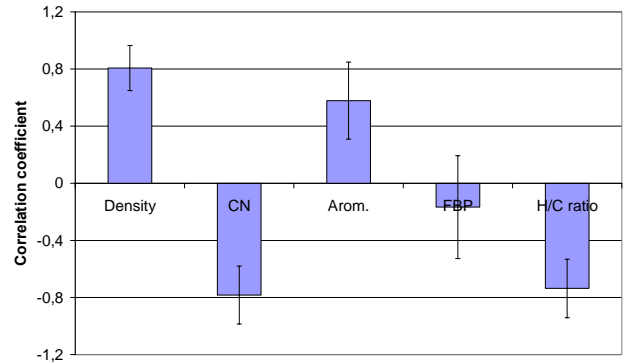
the associated increases in cylinder pressure and global temperature, fig 12.

Increasing EGR levels were found to increase CO emissions. This is not surprising since EGR is used to reduce the combustion temperature.



**Figure 21:** CO emissions at 6.6 bar IMEP

Correlations between CO emissions and the density, cetane number, aromatics, final boiling point and H/C ratio of the fuels were also calculated, see Figure 22. The variables with the strongest effects on the CO emissions are the density and the cetane number.

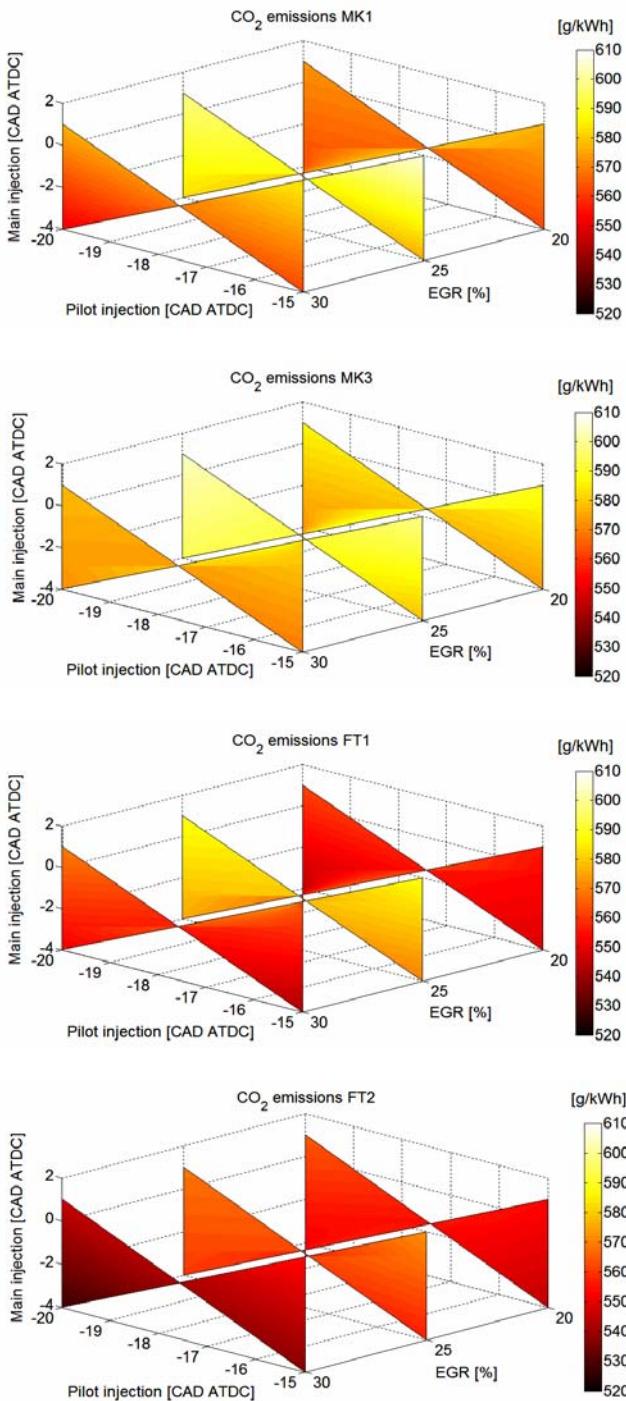


**Figure 22:** Correlations between the soot emissions and the density, cetane number, aromatic content, final boiling point, and H/C ratio of the fuels.

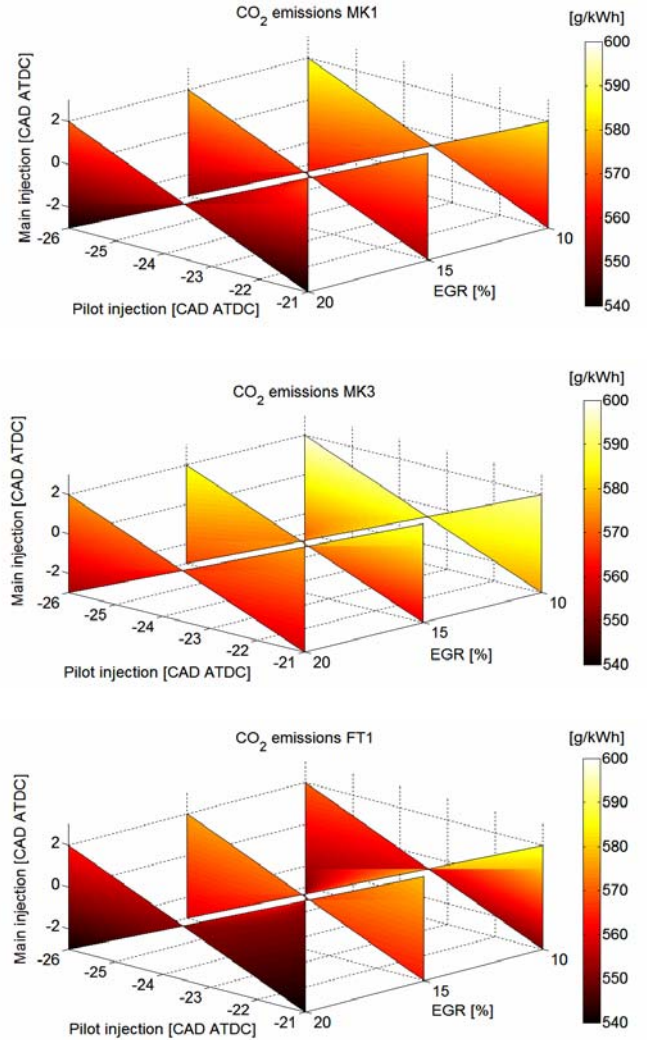
### CO<sub>2</sub> EMISSIONS

The CO<sub>2</sub> emissions presented in Figures 23 and 24. For the medium load case CO<sub>2</sub> emissions decline with increasing EGR. However, for the low load case the intermediate level of EGR (25%) yields the highest CO<sub>2</sub> emissions, probably reflecting a balance between the EGR-enhancing effects on CO formation and CO oxidation, see Figures 20 and 21.

The data also show that there are differences between the CO<sub>2</sub>-emissions characteristics of the fuels. The FT1 fuel yields the lowest level of CO<sub>2</sub> emissions and MK3 the highest. It can also be seen that retarding the main injection results in increased CO<sub>2</sub> emissions. The reason for this depends on the reduced thermodynamic efficiency due to the late phasing of the heat release.



**Figure 23:** CO<sub>2</sub> emissions at 3.6 bar IMEP



**Figure 24:** CO<sub>2</sub> emissions at 6.6 bar IMEP

To assess the CO<sub>2</sub>-forming potential of the fuels under ideal conditions, their chemical composition was estimated and the amounts of CO<sub>2</sub> theoretically produced per unit energy (lower heating value) supplied by each fuel was determined.

To calculate the chemical composition, C<sub>a</sub>H<sub>b</sub>, of the fuels, equation (3) is used [18].

$$C_aH_b + \left(a + \frac{b}{4}\right)(O_2 + 3.773N_2) = a \cdot CO_2 + \frac{b}{2}H_2O + 3.773\left(a + \frac{b}{4}\right) \cdot N_2 \quad (3)$$

The results of the calculations presented in table 7, show that MK3 produces the most CO<sub>2</sub> under ideal conditions, explaining why MK3 generates higher CO<sub>2</sub> emissions than the other fuels.

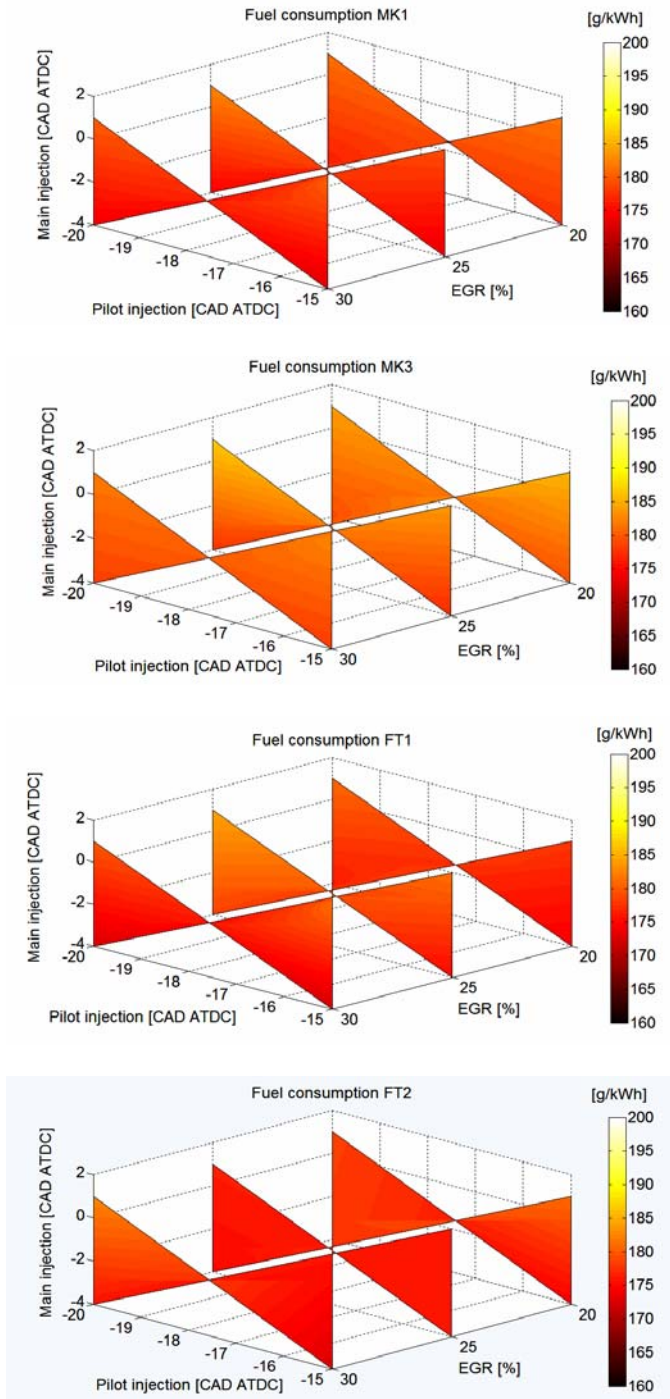
**Table 7:** calculated amounts of CO<sub>2</sub> produced from each of the fuels under ideal conditions.

	MK1	MK3	FT1	FT2
H/C ratio	2.02	1.87	2.15	2.14
Heating value [MJ/kg]	43.5	42.8	43.7	43.6
Produced CO <sub>2</sub> / energy [g CO <sub>2</sub> /MJ]	<b>0.03726</b>	<b>0.03828</b>	<b>0.03675</b>	<b>0.03684</b>

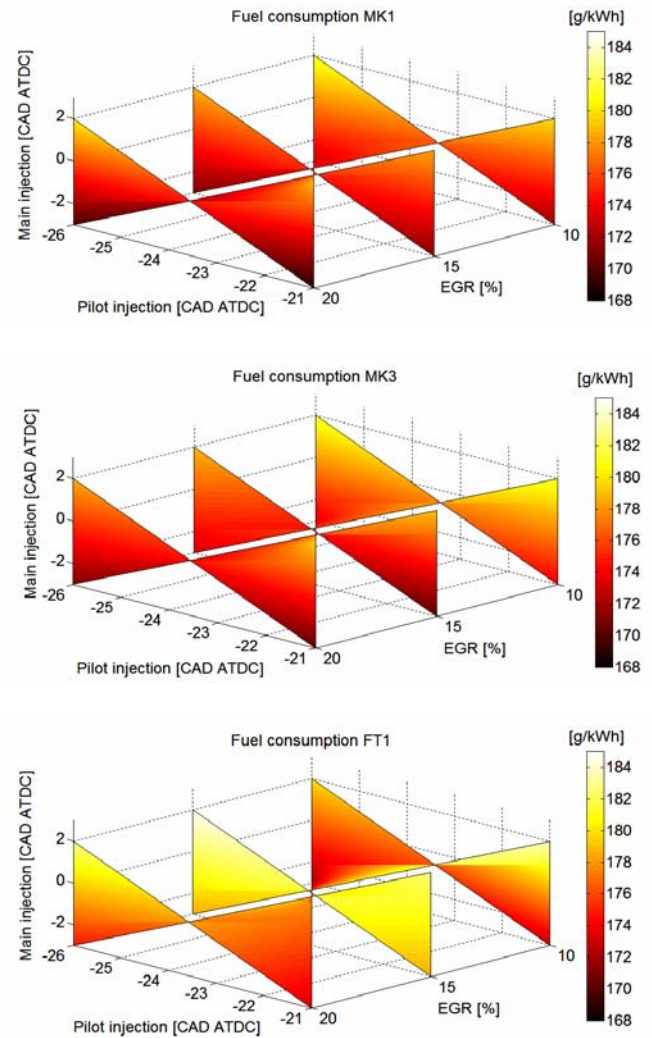
If the CO<sub>2</sub> emissions would be compensated for the CO<sub>2</sub> forming potential (compared to MK1) the CO<sub>2</sub> emissions from the different fuels would be similar.

## FUEL CONSUMPTION

Figures 25 and 26 present the specific fuel consumption for each of the fuels, which shows the differences between the fuels are minor in this respect.



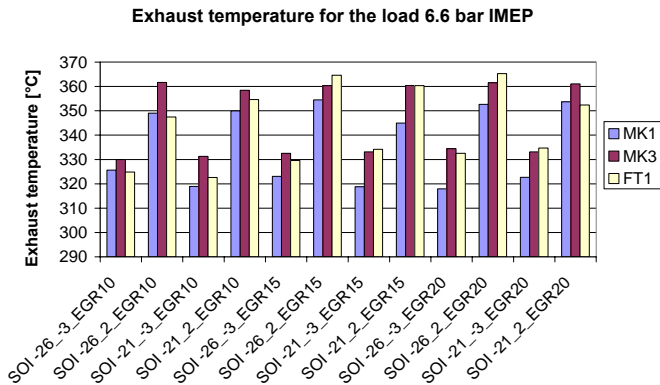
**Figure 25:** Fuel consumption at 3.6 bar IMEP



**Figure 26:** Fuel consumption at 6.6 bar IMEP

Late main injections increase fuel consumption for each tested combination of fuel and load. This is because they cause the heat release to peak later, and more fuel has to be injected in order to achieve the same load due to loss in expansion.

In Figure 27 the exhaust temperature is presented for the medium load case. As it can be seen, the exhaust temperature is higher for the case with late main injection and this also explains the higher fuel consumption for late main injection.



**Figure 27:** Exhaust temperature for the medium load.

## CONCLUSIONS

Apart from the fact that F-T fuels can be produced from variety of feedstocks, including biomass, waste, natural gas and coal, the results of this study show that F-T Diesel is a very good alternative fuel with several advantages:

- Due to its negligible aromatic contents, the soot emissions are low with FT fuel.
- Combustion with FT fuels results in reduced HC and CO emissions due to higher cetane numbers and lower densities.
- The NO<sub>x</sub> emissions found in this study are lower for F-T fuel compared to conventional Diesel fuels.
- The specific fuels consumption is similar for the F-T fuels to that of conventional Diesel fuels.

Moreover, in addition, to the results above:

- F-T fuel can be blended with fossil fuel.
- No engine modifications are required to operate the engine with F-T fuels.

The challenge to develop economical feasible manufacturing process for F-T fuels with low environmental impact.

## ACKNOWLEDGMENTS

The authors would like to thank the Swedish Road Administration, EMFO program, and the following companies for their financial and technical support: GM Powertrain, Scania, Statoil, Volvo Powertrain, Volvo Car Corporation and Volvo Technology. The help with performing the experiments, Dr Arjan Helmantel, Mr. Rolf Berg and all the technical staff at the Department of Applied Mechanics, Chalmers University is gratefully acknowledged.

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## ABBREVIATIONS

<b>ATDC</b>	after top dead centre
<b>CN</b>	cetane number
<b>C<sub>t</sub></b>	0.161 (constant)
<b>ρ<sub>A</sub></b>	air or ambient gas density
<b>ρ<sub>F</sub></b>	fuel density
<b>SOI</b>	start of injection
<b>t</b>	time
<b>U</b>	injection velocity
<b>x<sub>tip</sub></b>	spray tip distance

