

Effects of Different Type of Gasoline Fuels on Heavy Duty Partially Premixed Combustion

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ABSTRACT

The effects of fuel properties on the performance and emissions of an engine running in partially premixed combustion mode were investigated using nine test fuels developed in the gasoline boiling point range. The fuels covered a broad range of ignition quality and fuel chemistry.

The fuels were characterized by performing a load sweep between 1 and 12 bar gross IMEP at 1000 and 1300 rpm. A heavy duty single cylinder engine from Scania was used for the experiments; the piston was not modified thus resulting in the standard compression ratio of 18:1.

In order to properly run gasoline type of fuels in partially premixed combustion mode, an advanced combustion concept was developed. The concept involved using a lot of EGR, very high boost and an advanced injection strategy previously developed by the authors.

By applying this concept all the fuels showed gross indicated efficiencies higher than 50% with a peak of 57% at 8 bar IMEP. NO_x were mostly below 0.40 g/kWh only in few operative points 0.50 g/kWh was reached. At high load the soot levels were mostly a function of the octane number; with RON higher than 95 it was possible to be below 0.5 FSN while for the more reactive fuels a peak value of 3 FSN was reached at 13 bar IMEP.

The pressure rise rate reached a peak of 19 bar/CAD with fuels which had a RON above 95, when the octane

number decreased below 90 the pressure rise rate was always below 14 bar/CAD.

INTRODUCTION

First introduced by Onishi in 1979 for 2-stroke cycle engines [1], Homogenous Charge Compression Ignition, HCCI, seemed to be an excellent solution to simultaneously reduce fuel consumption, soot and NO_x. Due to very stringent emission regulations and need to reduce CO₂ emissions, HCCI was widely and deeply investigated for 4-stroke engine cycles between the end of the Nineties and first half of the next decade. More than a decade of research underlined many limitations of HCCI. The control of the combustion is one of the most critical aspects. Since HCCI relies on the autoignition of the fuel-air mixture, factors affecting the start of combustion are numerous and require significant technological efforts to a point that the gain is negligible as compared to the effort. In addition HCCI combustion has poor power density; high load operations are hard to achieve without resulting in very high acoustic noise which can not be tolerated by vehicle occupants and pedestrians.

To overcome the controllability problems and increase the power density Partially Premixed Combustion, PPC, was introduced [2]. The target of PPC is to achieve a separation between the end of the injection and the start of the combustion. By doing so the process is controlled by the injection event while emissions are reduced because of the low stratified mixture when combustion takes place. There are many ways to accomplish the

separation between end of injection and start of combustion: inject fuel very early in the cycle, decrease the compression ratio, use a large amount of EGR and decrease the cetane number of the fuels [3-10]. Toyota successfully implemented Diesel PPC to production but unfortunately the operating window of this strategy was limited at half load and roughly 2750 rpm [10]. In order to separate the end of injection with the start of combustion even at high loads, in 2006, Kalghatgi proposed to inject Gasoline in a compression ignition engine. The results were remarkable because he was able to run the engine at 13 bar IMEP and with 25% of EGR achieving 0.4 g/kWh of NO_x, 0.9% AVL smoke opacity and 174 g/kWh as fuel consumption [11, 12]. To accomplish roughly the same goal with Diesel, Andersson showed that 80% of EGR was needed and unfortunately the fuel consumption was 285 g/kWh [9].

Following the path introduced by Kalghatgi, the authors started to investigate the viability of using high Octane Number fuels in PPC. The first paper [13] underlined that a full separation between the end of injection and the start of combustion is optimal to simultaneously reduce NO_x and soot but unfortunately fuel consumption increases due to the high pressure oscillation created after combustion [14]. In a second paper [15] an advanced combustion concept was introduced in order to run high load PPC and simultaneously achieve low emissions and high efficiency. At high load the concept consisted of injecting a considerable amount of fuel in the first injection very early in the compression stroke, add lots of EGR in order to avoid reactions during the compression stroke and then trigger the combustion through the stratification created by the second injection around TDC. The injection strategy developed in [15] was applied to Ethanol at high load [16]; high efficiency and NO_x and soot emissions simultaneously below EURO VI and US10 were achieved.

The aim of this paper is to understand how fuels in the Gasoline boiling range, but with RON spanning from 99 to 69, behave with this advanced combustion concept. A load sweep was performed between 1 and 12 bar gross IMEP at 1000 and 1300 rpm, 50% of EGR was used and the fuel amount in the pilot was a function of the reactivity of the fuel under examination.

EXPERIMENTAL SET-UP

Engine- The engine used during the experiments was a heavy duty single cylinder compression ignition engine, Scania D12. The cylinder head was flat and the piston used was a shallow bowl type. The geometrical properties can be found in Table 1. The engine was boosted by using compressed air from an external air line; the back pressure was increased to be within 0.2 and 0.4 bar higher than the inlet one thus allowing the EGR flow. The inlet pressure was adjusted by using a waste gate valve. A heater (from Leister) placed before the inlet manifold, was used to heat up the air at the desired inlet temperature.

Table 1: Geometric properties of the Scania D12.

Displaced volume	1966 cc
Stroke	154 mm
Bore	127.5 mm
Connecting Rod	255 mm
Geometric Compression Ratio	18:1
Swirl Ratio	2.9

EGR- The EGR used in this paper is defined as the ratio of carbon dioxide in the intake and exhaust. The exhaust gasses were cooled down before being introduced in the intake.

Injection system- The Scania D12 was equipped with a first prototype common rail system from Bosch. The commercial nozzle was replaced with one that had an umbrella angle of 120°. The nozzle had 8 orifices, their diameter was 0.18 mm. The rail pressure was held constant at 1800 bar. The fuel flow was measured by using a gravity scale with two digits precision from Sartorius and each operative point was sampled for at least two minutes.

Emission Measurements Systems- The emissions were measured with the following devices. By using a Cusson gas analysis system, CO and CO₂ were measured by non-dispersive infrared analyzer, O₂ with a paramagnetic analyzer and total HC with a heated flame ionization detector from JUM. A chemiluminescent analyzer was used for NO_x (ECO Physics CLD 700 EL unit) and the smoke was measured with an AVL 415 opacimeter. Each analyzer was calibrated with an appropriate calibration gas before every set of measurements. A schematic of the experimental apparatus used in these experiments is shown in Figure 1.

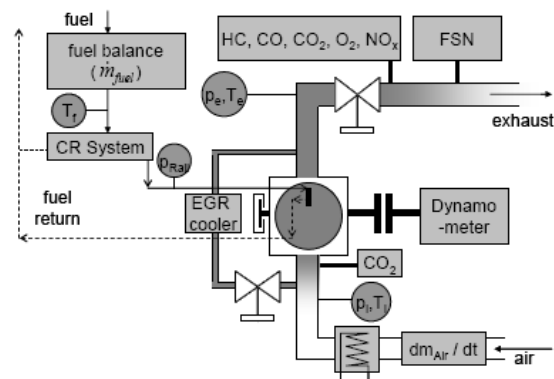


Figure 1: Scania D12 schematic experimental apparatus.

FUELS

Nine fuels, supplied by Chevron, were tested. All the fuels were in the gasoline boiling point range; their main properties can be found in Table 2. In order to facilitate

the comprehension of the results the fuels were divided in three categories according to their Octane Number. The RON and MON of fuels FR47331CVX and FR47333CVX were unknown, it was decided to place them in Group 2 and 3 respectively because of similar ignitibility behaviors.

Table 2: Main properties of the nine Chevron fuels under examination.

		RON	MON	C	H/C	O/C	LHV [MJ/kg]	A/F _{stoich}
Group1	FR47335CVX	99.0	96.9	7.04	2.28	0.00	44.30	15.10
	FR47332CVX	97.7	87.5	6.61	2.06	0.07	39.70	13.44
	FR47337CVX	96.5	86.1	7.53	1.53	0.00	42.10	14.03
Group2	FR47338CVX	88.6	79.5	7.21	1.88	0.00	43.50	14.53
	FR47330CVX	87.1	80.5	7.20	1.92	0.00	43.50	14.60
	FR47331CVX	N/A	N/A	6.90	1.99	0.03	41.60	14.02
Group3	FR47336CVX	70.3	65.9	7.10	2.08	0.00	43.80	14.83
	FR47334CVX	69.4	66.1	7.11	1.98	0.00	43.80	14.68
	FR47333CVX	N/A	N/A	7.16	1.97	0.00	43.70	14.65

ADVANCED INJECTION STRATEGY

The injection strategy developed in [15] in order to run high Octane Number fuels in a compression ignition engine and simultaneously achieve high efficiency and low emissions consisted of two injections. The first one is placed very early in the compression stroke and it is used to create a homogeneous mixture while the second is injected around TDC and its function is to trigger the combustion event. The fuel amount in the first injection is independent on the load and it is a function only of compression ratio, reactivity of the fuel and EGR level; this injection is always placed at -60 TDC. The load and combustion phasing are controlled through the second injection. It is important to underline that the first injection must not to react during compression stroke thus a certain amount of EGR has to be added to avoid pre-ignition. All the fuel belonging to Group 3 and FR47331CVX did not need the first injection because of their high reactivity. For the remaining five, the fuel amount injected in the first injection was 7.70 bar Fuel MEP¹. The percentage of fuel amount in the first injection as a function of the load is presented in Figure 2.

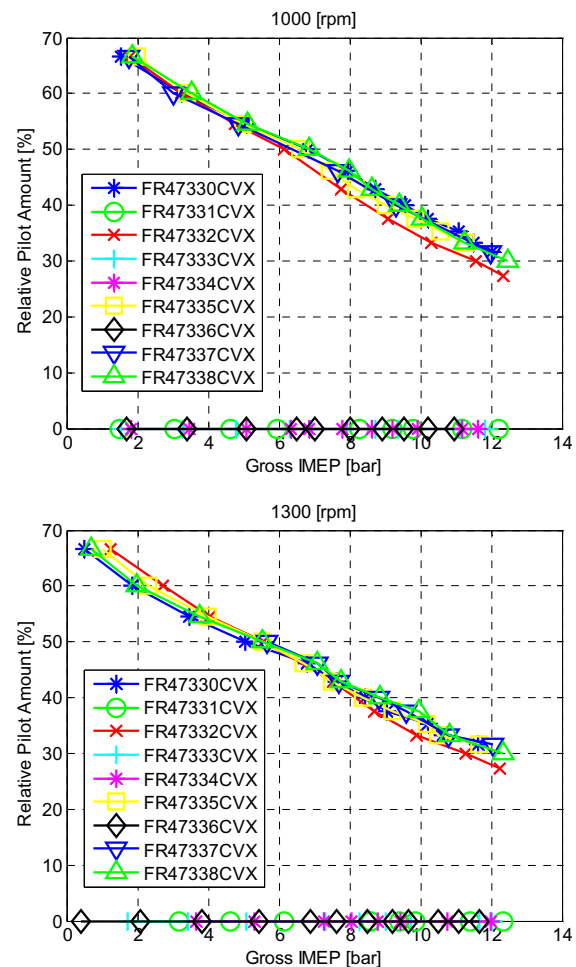


Figure 2: Fuel percentage in the pilot injection as a function of load, speed and fuel type.

¹ Fuel MEP is defined as the fuel energy per injection, joules, normalized with the displacement volume.

EXPERIMENTS AND INLET CONDITIONS

The nine fuels were tested on a load sweep between 1 and 12 bar gross IMEP at 1000 and 1300 [rpm]. The inlet pressure was adjusted according to the load, see Figure 3, in order to have values that in real life applications can be achieved by using a conventional

turbo system. As the load decreased the inlet temperature was raised in order to keep stable combustion, see Figure 4. The EGR levels shown in Figure 5 were between 40 and 50%; this amount was chosen in order to keep NO_x below 0.40 g/kWh at the highest load. Between 6 and 12 bar gross IMEP the engine was running at roughly λ 1.50, Figure 6, in order to have enough air to avoid excess of smoke.

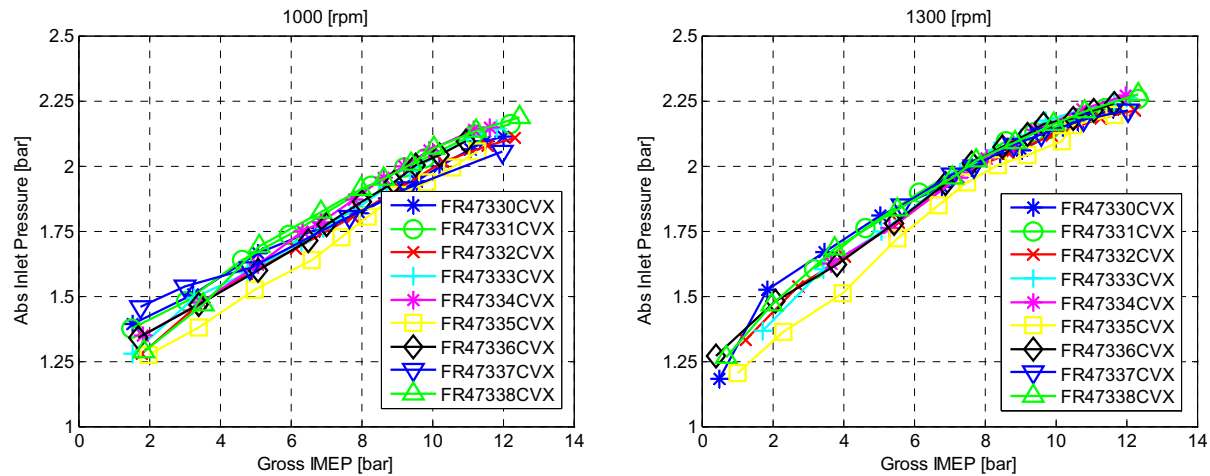


Figure 3: Absolute inlet pressure as a function of load, speed and fuel type.

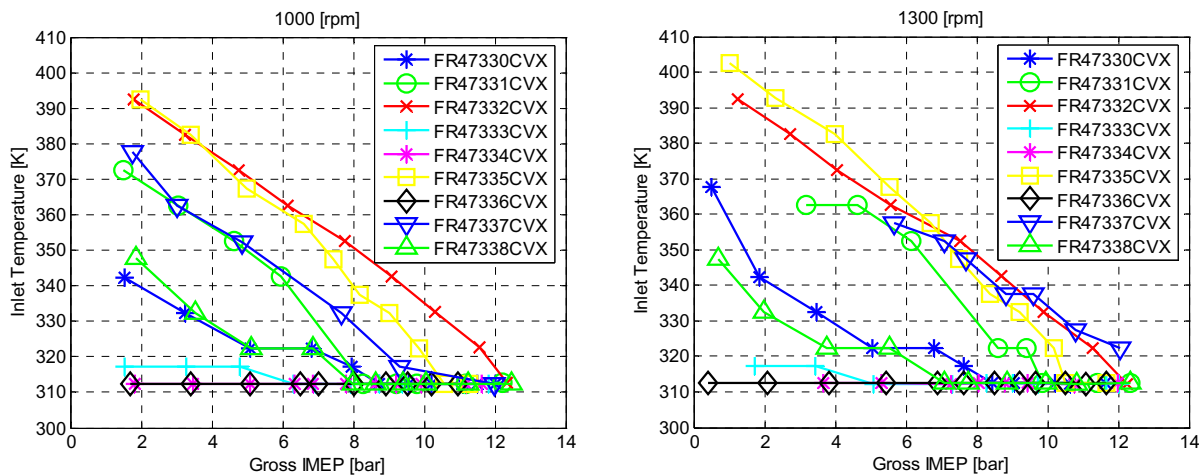


Figure 4: Inlet temperature as a function of load, speed and fuel type.

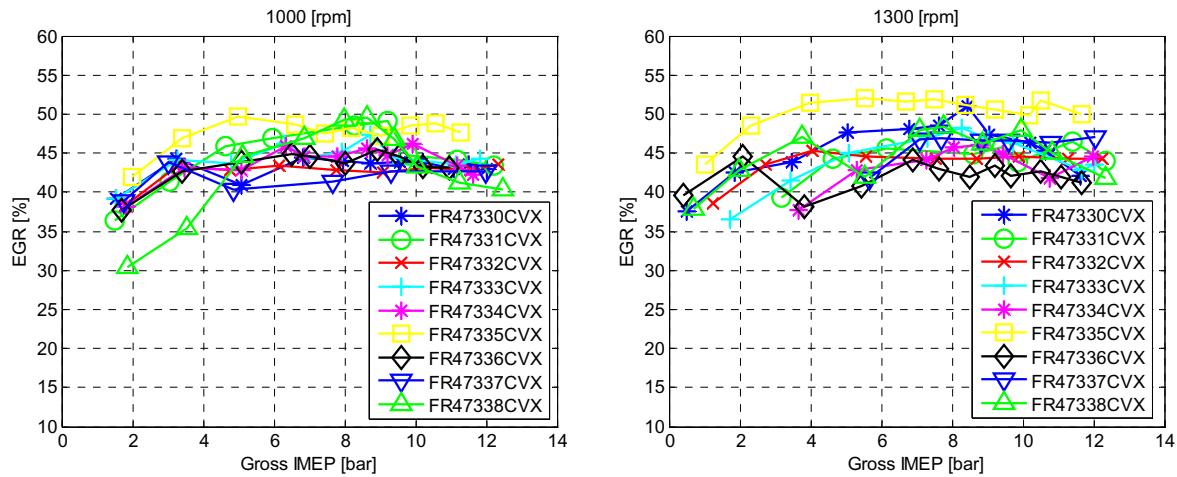


Figure 5: Exhaust gas recirculated, EGR, as a function of load, speed and fuel type.

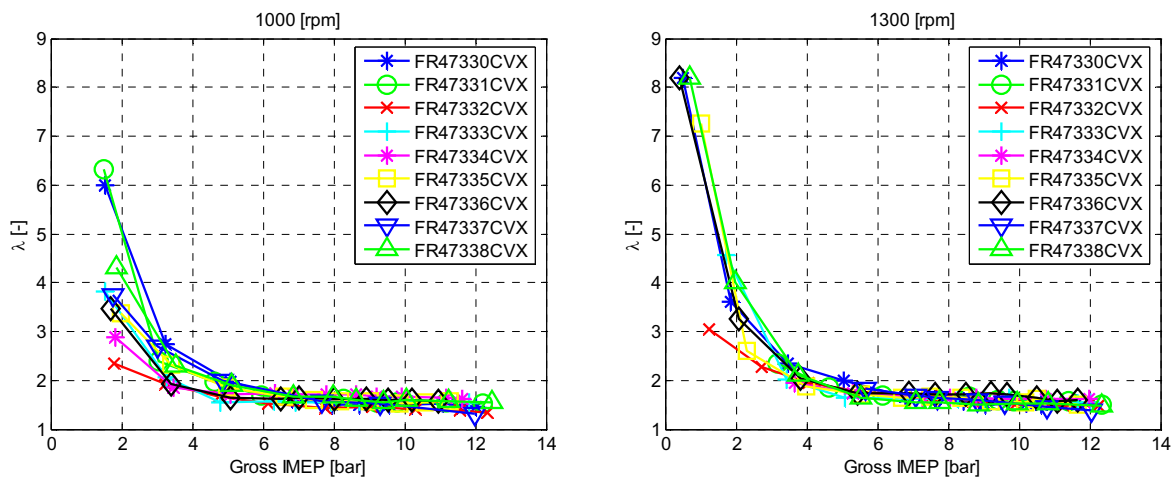


Figure 6: Relative excess of air, λ , as a function of load, speed and fuel type.

RESULTS AND DISCUSSION

Efficiencies

The gross indicated efficiencies plotted as a function of load and speed are shown from Figure 7 to Figure 9 respectively for the fuels from Group 1 to 3. As a result of lower heat transfer, by increasing the engine speed the efficiency increases for all the 9 fuels. On the other hand for a given speed the gross indicated efficiency increases by decreasing the fuel Octane Number. Figure 10 shows that, at high load, the mixing period² decreases together with the fuel Octane Number. This suggests that when combustion takes place the hot areas of the combustion are closer to the wall with fuels belonging to Group 1 while the hot temperature reactions are more confined in the piston bowl area with fuel belonging to Group 3; Group 2 is in an intermediate position. If the hot areas of the combustion are closer to

the wall the heat transfer is enhanced thus resulting in slightly lower gross indicated efficiency when the Octane Number increases from 69 to 99.

With load higher than 4 bar gross IMEP all the nine fuels are showing efficiency much higher than 50% with a peak of 57% at 8 bar IMEP at 1300 [rpm] with fuels FR47338CVX and FR47333CVX. Despite the fact that more than 50% of EGR was used, the combustion efficiency³ shows values higher than 95% after 4 bar gross IMEP and it steadily increases to more than 99% at higher load. At very low load some fuels exhibited combustion efficiency below 80 %, e.g. FR47330CVX, this was properly due to insufficient inlet temperature to keep high enough temperature during combustion to promotes reaction from HC to CO and from CO to CO₂; see Figure 12. High values of gross indicated efficiency were achieved thanks to the completeness of combustion, high combustion efficiency, and high thermal efficiency; see Figure 11. With loads higher than

² The crank angles between the end of last injection and 10% of total heat release.

³ Calculated from the exhaust gas analysis [17].

4 bar IMEP, the thermal efficiencies were in the range of 52 – 57%. High thermal efficiency can be achieved by simultaneously minimizing the heat transfer and the exhaust loss.

By applying the first law of thermodynamics from the compression to the expansion stroke, the energy balance can be written as:

$$Fuel_Energy = W_{i_g} + HT + Ex + Comb + Other \quad (1)$$

Where *Fuel_Energy* is the fuel mass per cycle times the heating value, *W_{i_g}* is the gross indicated work, *HT* is the heat transfer, *Ex* the exhaust energy loss, *Comb* is the combustion inefficiency (CO, HC and H₂), *Other* are the blow-by and crevice losses (negligible). By dividing the combustion loss, heat transfer and the exhaust loss with the fuel energy, the relative losses can be computed as a function of load and speed; see Figure 13, Figure 14 and Figure 15. With loads higher than 4 bar gross IMEP the combustion losses are negligible (below 5%), the heat transfer is between 25 and 30% and the exhaust losses are not higher than 20%. Low heat transfer and low exhaust loss can be simultaneously achieved by:

- High boost.
- Properly selected combustion phasing and combustion duration.

By boosting the engine the internal energy of the combustion volume increases (more mass), the heat transfer is mainly a function of the temperature gradient between the bulk volume and wall (it is independent of the mass inside the combustion chamber thus it is independent on the boost level) and the exhaust losses are roughly constant (the increase in exhaust mass balances with the decrement in exhaust temperature due to the higher mixture dilution⁴). For a given fuel energy per cycle, the indicated gross work can be expressed as a function of the above mentioned parameters as follows:

$$W_{i_g} \approx Internal_Energy - HT - Exhaust - Other \quad (2)$$

thus $P_{in} \uparrow \Rightarrow Internal_Energy \uparrow \Rightarrow W_{i_g} \uparrow$

According to previous research [17, 18] depending on the load, if the phasing of the combustion, CA50, is placed between 5 and 15 CAD after TDC the heat transfer can be minimized (lower combustion temperature as compared to the ideal Otto cycle). On the other hand if the combustion duration is in the range of 10 – 30 CAD the expansion stroke is maximized and heat transfer minimized; if this duration is longer the effective expansion ratio decreases while if too short the

heat transfer increases. For these experiments the combustion duration and combustion phasing as a function of load and speed are shown in Figure 16 and Figure 17 respectively.

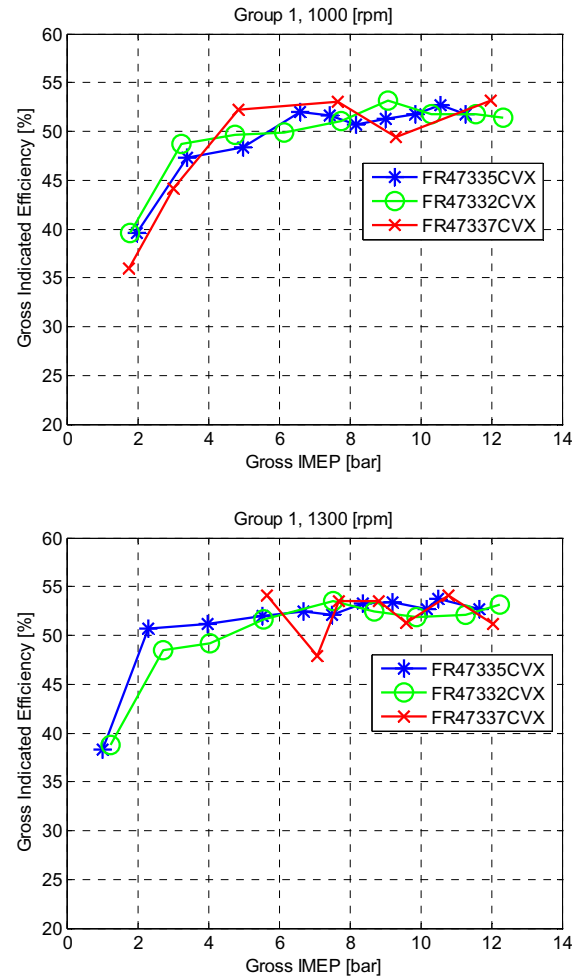


Figure 7: Gross indicated efficiency as a function of IMEP at 1000 and 1300 [rpm] for fuel belonging to Group 1.

$$Exhaust_Entropy = (m_{air_in} + m_{fuel}) \cdot c_{p_exhaust} \cdot (T_{exhaust} - T_{reference})$$