PERFORMANCE AND EMISSIONS CHARACTERISTICS OF ALCOHOL / DIESEL DUAL FUEL ENGINE

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Abstract. *The aim of this paper is to investigate the performance and emissions characteristics of a turbocharged and aftercooled diesel engine operated with hydrated alcohol, as primary fuel, and diesel fuel, as ignition source. In such dual fuel operation, much of the energy released comes from the combustion of the renewable fuel while only a small amount of diesel fuel provides ignition through timed cylinder injection. The effects of diesel-ethanol substitution ratios, engine speed, and load on the equivalent brake thermal efficiency and emissions of nitrogen oxides, smoke and hydrocarbons are examined for alchol-diesel fuel operation and, afterward, compared with the original diesel operation. The results show that, over a wide range of operating conditions (engine speed, load), the dual fuel engine clearly shows the benefits of reduced* NO_X *and smoke emissions. However, under low loads, the results indicate high HC emissions and a lower thermal efficiency when compared to those of the corresponding diesel engine.*

Keywords*: Diesel, dual fuel engine, emissions, ethanol.*

1. INTRODUCTION

Environmental awareness, stimulated fuel diversification and the high price of conventional fuels have led to the creation of incentives to promote and further evaluate alternative fuel sources for conventional internal combustion engines. One example in compression ignition engines is the use of a renewable fuel as partial supplement to diesel fuel. Such operation, known as “dual fuel” or “fumigation”, represents an attractive and flexible means for utilizing a range of renewable fuels, including ethanol. The dual concept involves the utilization of secondary fuel by first mixing it with the air intake; ignition of this premixed lean charge is then accomplished by injecting a small quantity of diesel fuel, the “pilot”, near top dead centre of the compression stroke. This pilot fuel can readily auto-ignite to provide an ignition source for subsequent flame propagation within the surrounding gaseous fuel–air mixture (Badr et al., 1999; Kusaka, 2000).

Studies on the use of ethanol in Diesel engines have been continuing since the 1970s. The initial investigations were focused on reduction of the smoke and particle levels in the exhaust (Can, 2004).

Broukhiyan and Lestz (1981) applied ethanol fumigation (addition of alcohols to the intake air charge) to a light-duty automotive Diesel engine, up to 50% of the total fuel energy. Their research was undertaken to study the effect of ethanol fumigation on the performance (efficiency), combustion knock characteristics and exhaust emissions. For all conditions except the 1/4 rack setting (light load) condition, modest thermal efficiency gains were observed upon fumigation. The occurrence of severe knock limited the maximum amount of ethanol that could be fumigated. Brake NO_X and particulate concentrations were found to decrease for all conditions tested.

Hayes et. al. (1988) tested 100, 125, 150, 175 and 200 proof ethanol as fumigants in a 6-cylinder turbocharger diesel engine at 2400 rpm. Ethanol was injected directly into the intake ports. They found that the lower proofs reduced the maximum rate of pressure rise. Any proof at lower load reduced NO levels. HC emissions increased greatly as the ethanol substitution was increased without depending on the ethanol proof.

Abu-Qudais et al. (2000) have found that the fumigation and alcohol-diesel blends (mixture of the fuels prior to injection) methods of have similar behavior in affecting the performance and emissions, but the improvement in using the fumigation method was better. This work was performed on single cylinder, direct injection, aspirated research engine.

The objective of the present work is to investigate the performance and emissions characteristics of a turbocharged and aftercooled diesel engine operated with hydrated alcohol (93.3 ^{o}INPM) and diesel pilot ignition. With this in mind, experimental tests of the dual fuel operation (over a wide range of engine speeds, loads and substitution ratios of ethanol by diesel fuel) were conducted at the Vehicles Laboratory of Pontifical Catholic University of Rio de Janeiro.

2. PROPERTIES OF FUEL ALCOHOL

The fuel alcohol is a renewable and clean product that contributes to reducing the glasshouse effect and substantially diminished the air pollution, minimizing its impacts on public health. In Brazil, the intense use of alcohol restricts the emission of pollutants of the growing fleet o vehicles, mainly carbon monoxide, sulfur oxides, lead compounds and toxic organic compounds such as benzene. Fuel hydrated ethyl alcohol (AEHC), used in alcohol-run cars (Otto cycle engine), as the actual name says, is hydrated, that is, it has water. This water content is on average 7% (ANP Rule 36/2005 fixed the alcohol content in the 92.6-93.8º INPM band. In this work, the alcohol content was 93.3 ^{o}INPM.

Alcohol fuels have a very low octane number and need a positive ignition source or additives (ignition improvers) in the fuel to raise the cetane number to be used in diesel engines. Alcohol fuels have some emissions advantages regarding NOX and particulate emissions, but some emission components can also be higher than for diesel fuel (Majewsky e Khair, 2006). However, alcohols have very poor lubricating characteristics and the heating value of alcohol is less than that of diesel fuel; therefore, more alcohol than diesel fuel is required by mass and volume (Sahin, 2005).

3. EXPERIMENTAL APPARATUS AND TEST CONDITIONS

3.1. Experimental Facilities

The engine used in the present study is an MWM model 4.10 TCA, four cylinder, direct injection, turbocharged, aftercooled diesel. This commercial engine, used in a variety of vehicle applications, has been properly modified to operate under dual fuel conditions. The technical specifications of the engine are given in Tab. 1.

Table 1. General Specification of the MWM 4.10 TCA Diesel Engine Test Cases Examined.

Item	Specification
Number of cylinders and arrangement	4 in-line
Bore	103 mm
Stroke	129 mm
Displacement	4.3 L
Compression Ratio	17:1
Valves / Cylinders	2
Speed range	840 – 2600 rpm
Rated power and Speed	107 kW (145 cv) at 2600 rpm
Peak Torque and Speed	500 Nm at 1600 rpm

The engine was tested on an AVL model Alpha 240 electric dynamometer, rated for testing engines up to up to 600 Nm and 8000 rpm. The strength of the electromagnetic field coupling the rotating and stationary parts of the dynamometer was adjusted in order to increase or decrease the resistance offered to the engine rotation. Likewise, the diesel engine injection pump was driven by a LENZE linear actuator.

For measurement of the air flow rate, two calibrated nozzles were used. The nozzles were installed in a compensation tank, which was installed upstream of the turbocharger inlet to damp out the air pulsation generated by the engine. The pressure drop across the nozzles was transmitted to the acquisition system using a differential pressure transducer.

A high precision AVL model 733S electronic flow meter was used to measure diesel fuel mass flow rate. The AEHC flow measurement was via a coriolis-type flow meter (Micro Motion CMF 010).

NO_X and HC were measured by sampling the exhaust gases and analyzing them with a TESTO 350 XL emissions analyzer. Before of the tests, all the gas sensors of the analyzer were calibrated with standard gases. Smoke samples were analyzed using a variable sampling smoke meter AVL 415S, which is a compact unit for automatic measurement of the soot content in the exhaust gas of particulate emitting internal combustion engines.

The output signals of all the above mentioned instruments, as well as diesel pump throttle, torque, engine speed, temperatures (air, diesel fuel, AEHC, lubricating oil, coolant and exhaust gas), relative humidity (to correct for air consumption) and pressures (air and AEHC) were fed to a Start-AVL data acquisition system .

Pictures of experimental facilities are shows in Fig. 1.

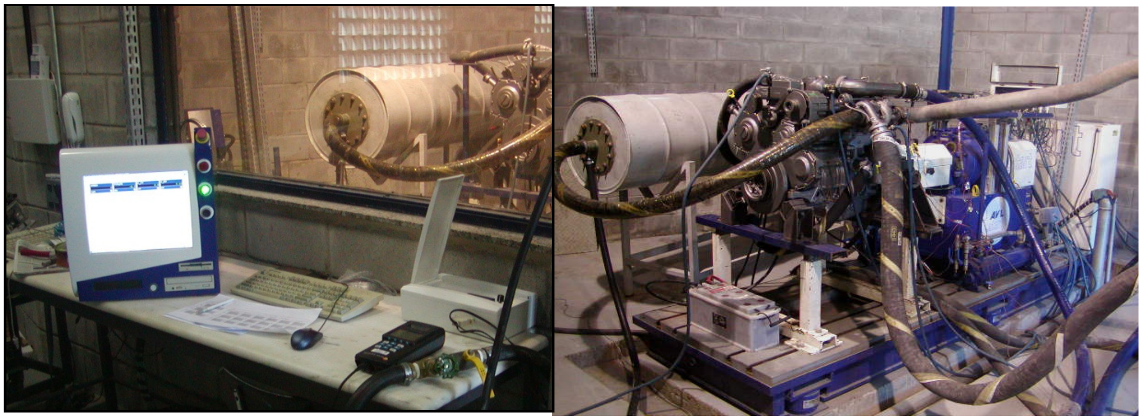

Figure 1. Photographic views of Diesel engine on test bench.

3.2. Test Conditions Examined

To determine their effects on the performance and emission characteristics during the dual fuel operation, the speed, load and diesel-fuel/ethanol substitution ratio were varied. Measurements were taken in the following ranges: load (power) from 11.3 to 113.7 cv, speed from 840 to 2600 rpm, substitution ratio from 0 to 69 percent (depending on operating conditions). First of all, the engine was run at a given load and speed under original diesel operation. Then, under the same engine operation conditions (engine speed, load), hydrated alcohol was injected with a simultaneous reduction of diesel fuel. Dynamometer control decreased automatically with the diesel fuel mass flow rate in order to maintain constant operating conditions. Thus, the AEHC fuel was increased until reach an unacceptable operation condition (combustion failure under low loads or presence of audible knock under higher loads). This procedure was employed to assess dual fuel operation with maximum substitution. Additionally, for each examined condition, performance and emissions of the dual fuel combustion were also evaluated using intermediate values for the substitution ratio.

In order to calculate the percentage of diesel fuel to ethanol substitution, the Eq. (1) was used:

$$SR_{D/E} = [1 - (\dot{m}_{DF}/\dot{m}_{OD})] \tag{1}$$

where $\dot{m}_{DF}$ and $\dot{m}_{OD}$ represent the diesel fuel mass flow rate in dual fuel and original diesel operation, respectively. Original diesel operation is represented by $SR_{D/E} = 0$ and dual fuel operation by $SR_{D/E} > 0$.

Likewise, under dual fuel operation, the thermal efficiency (η_T) was estimated from power output (P) per energy of both fuels.

$$\eta_T = P/[(\dot{m}_D \times LHV_D) + (\dot{m}_E \times LHV_E)] \tag{2}$$

where LHV_D and LHV_E are the lower heating values of both fuels, diesel and ethanol.

4. RESULTS AND DISCUSSION

For both engine operations, i.e. using original diesel and using dual fuel, three measurements were taken to average the data for each operating condition (load, engine speed). The duration for each measurement was of one minute, at a sampling rate of 100 Hz per channel. The repeatability of all results and the experimental error was evaluated according to Holman (1989). The maximum uncertainty in any quantity was in the 3.2–5.5% range.

Since the trends are observed for a broad range of loads and engine speeds, results are provided for just ten loads and one engine speed (1800 rpm).

4.1. Effect of Dual Fuel Operation on the Brake Thermal Efficiency

Figure 2 provides the variation of the brake thermal efficiency (BTE) as a function of percentage substitution ratio for various loads at 1800 rpm engine speed.

As shown, for low loads, BTE for dual fuel operation is noticeably lower compared to original diesel operation even though dual fuel performance is poorer at lower loads and when a higher proportion of AEHC is used. Thus, for example, for the 11,3 cv load, corresponding to 10% of full load at 1800 rpm, BTE decreased from 19.5% (verified under original diesel operation) to 12.5% (under dual fuel operation with a 56% substitution ratio). This difference represents a BTE decrease of 36%. However, as engine load increased, this tendency diminished, due to more complete combustion of the ethanol. At full loads, BTE under dual fuel operation is better that the original diesel operation. For example, at 1800 rpm and 113.7 cv, BTE incremented of 38.9% (original diesel operation) to 41.1% (dual fuel operation with 46% substitution ratio).

On the other hand, increasing the engine load, tends to drop substitution ratio of diesel by AEHC.

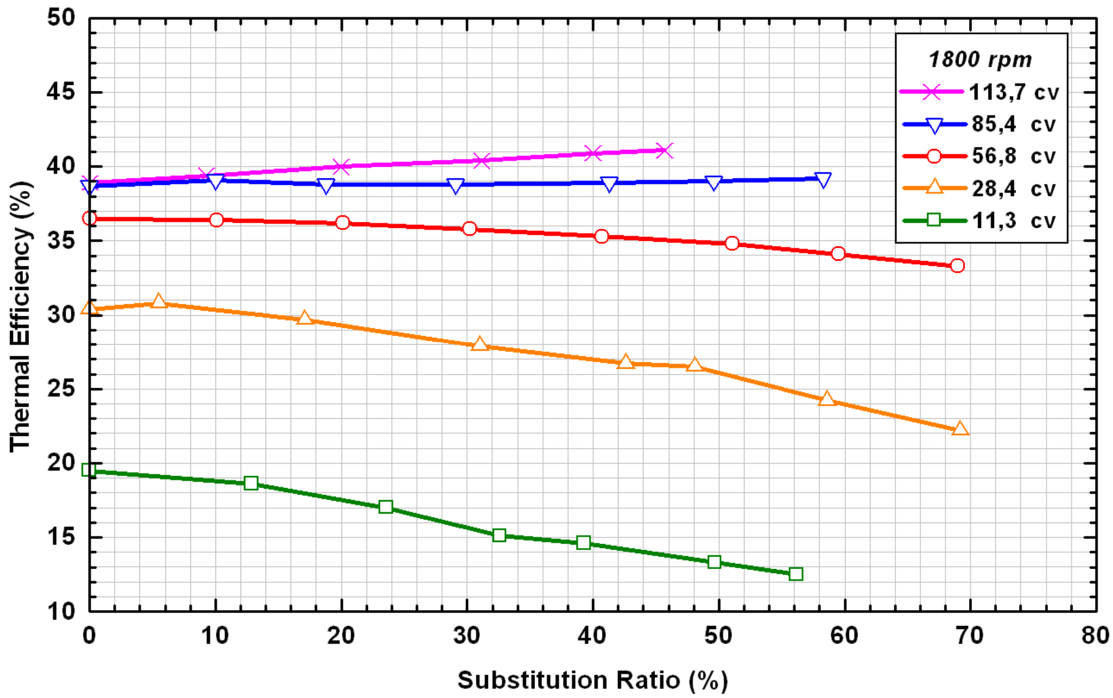

Figure 2. Brake thermal efficiency for different substitution ratios and engine loads. Engine speed: 1800 rpm.

4.2. Effect of Dual Fuel Operation on Nitrogen Oxide Emissions

The results presented in Fig. 3 show the variation of nitrogen oxide emissions for different substitution ratios and engine loads, at speed of 1800 rpm.

Nitrogen oxides, as defined by emissions regulations, include nitric oxide (NO) and nitrogen dioxide (NO_2). In newer technologies of turbocharged diesel engines, the proportion of NO_2 in total NO_X can be high as 15 percent or more. NO_X concentrations in diesel exhaust are typically between 50 and 1000 ppm (Majewsky and Khair, 2006).

As Fig. 3 shows, NO_X emissions are strongly dependent on engine load and the presence of AEHC in the charge mixture. Over a wide range of operating conditions (engine speed, load), in comparison with neat diesel fuel, NO_X emissions were reduced when AEHC and diesel fuels were used in combination. One of the marked features of dual fuel operation is that the ignition delay of the pilot fuel is considerably longer than for the original diesel (Karim, 1980). The charge cooling (by fumigation of AEHC) would increases ignition delay and reduce the maximum combustion temperature. Consequently, provide lower NO_X emissions. Another possible explanation is the low concentration of oxygen in the cylinder charge due to the presence of hydrated alcohol, which replaces an equal amount of air.

However, at higher loads and with the increase of substitution ratio, the NO_X concentrations showed a tendency to increase. Under such conditions, as more fuel is burned more energy is released and combustion temperature rises, thereby producing more NO_X. The turbocharging increases the mass of air inducted into the engine and allows a proportional increase in the injected fuel quantity, which leads to higher engine output. Furthermore, air velocity through the intake port is higher in turbocharged engines relative to naturally aspirated engines. It follows that air motion through the intake port of either a swirl-supported or quiescent combustion system is more likely to enhance mixing and mixture preparation. In general, better mixing leads to higher reaction and oxidation rates and also results in higher combustion temperatures, which in turn causes a reduction in the concentration of HC and CO emissions and, consequently, an increment of NO_X emissions (Majewsky and Khair, 2006).

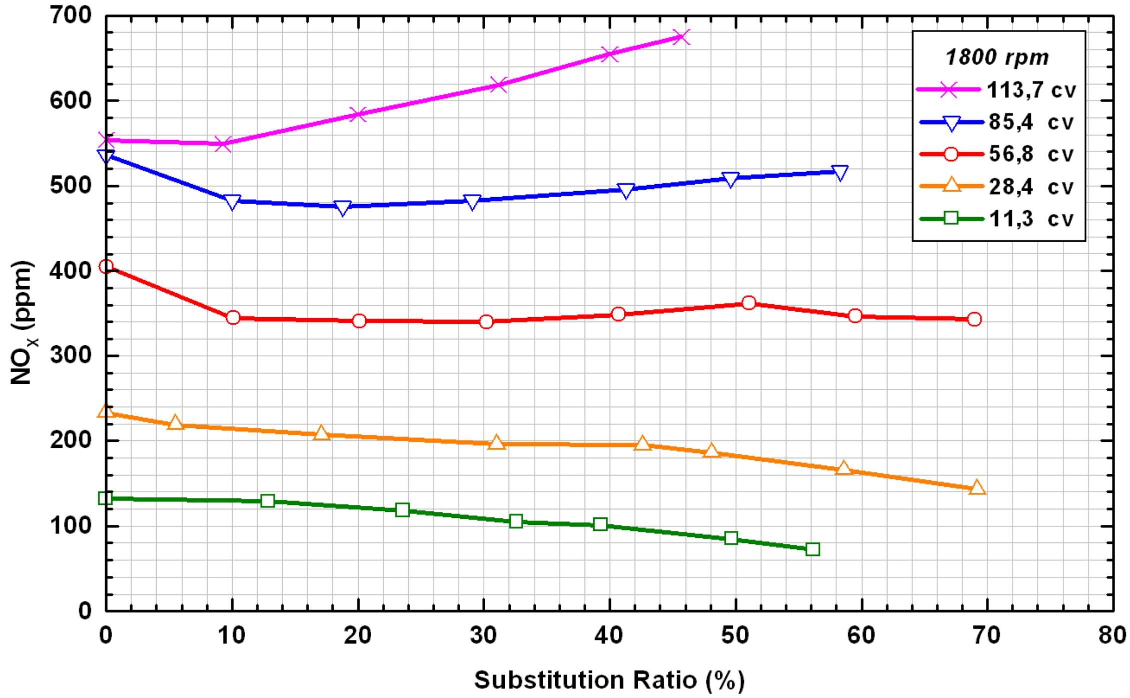

Figure 3. Nitrogen oxide emissions for different substitution ratios and engine loads. Engine speed: 1800 rpm.

4.3. Effect of Dual Fuel Operation on Smoke

The smoke density for dual fuel and diesel operations are shown in Fig. 4. From this figure, it can be seen that there is a drastic reduction in smoke, as more amount of ethanol were used.

It is known that the diesel fuel has a high tendency to soot formation due to its low H/C ratio and the nature of its combustion process (Heywood, 1988). Using ethanol, either as fumigant in a diesel engine (dual fuel operation), increases the hydrogen content in the cylinder interior and eventually reduces the engine smoke under normal diesel engine operating conditions.

Therefore, the charge cooling increases ignition delay and, thus, enhances the mixing of diesel fuel with the AEHC-air mixture which, in turn, makes for better air utilization and less smoke.

4.4. Effect of Dual Fuel Operation on Hydrocarbon Emissions

The emissions of hydrocarbons (HC) as a function of substitution ratio are given in Fig. 5 for various engine loads at 1800 rpm engine speed.

Hydrocarbons or, more appropriately, organic emissions are the consequence of incomplete combustion of hydrocarbon fuel (Heywood, 1988). Gaseous hydrocarbon levels in the exhaust of a diesel engine under normal conditions typically range from approximately 20 to 300 ppm (Majewsky and Khair, 2006).

As indicated in Fig. 5, HC emissions under dual fuel operation are considerably higher compared to diesel operation. These results are consistent and explain the BTE observed in Fig. 2. HC emissions rise rapidly as the substitution ratio of diesel fuel to ethanol is increased. Under these conditions, the AEHC–air mixture is very lean, which complicates flame propagation throughout the whole combustion chamber from the pilot ignition.

The HC emissions tend to increase because of the quench layer of unburned fumigated ethanol present during fumigation. There is no quench layer with diesel fuel injection alone because the combustion is droplet-diffusion-controlled and completely surrounded by air. Also, the high latent heat of vaporization can produce slow vaporization and mixing of fuel and air. These factors result in high HC levels (Abu Qudais, 2000).

Furthermore, for low load involving very lean AEHC–air mixtures, the employment of a large pilot fuel quantity contributes to combustion of the AEHC without hunting. The reduction of diesel injection mass flow rate, due to the AEHC addition, may change drastically the liquid fuel atomization. The tendency is a poorer atomization when the mass flow rate reduces and this has strong effects on hydrocarbons emissions. However, the atomization is most affected when the amount of fuel injected per cycle is reduced below 5–10% of the maximum design level (Abd Alla et al., 2000).

On the other hand, for a constant substitution ratio, with an increase in engine load there is a corresponding decrease in HC concentrations, indicating an increase in partial reactions or better flame propagation.

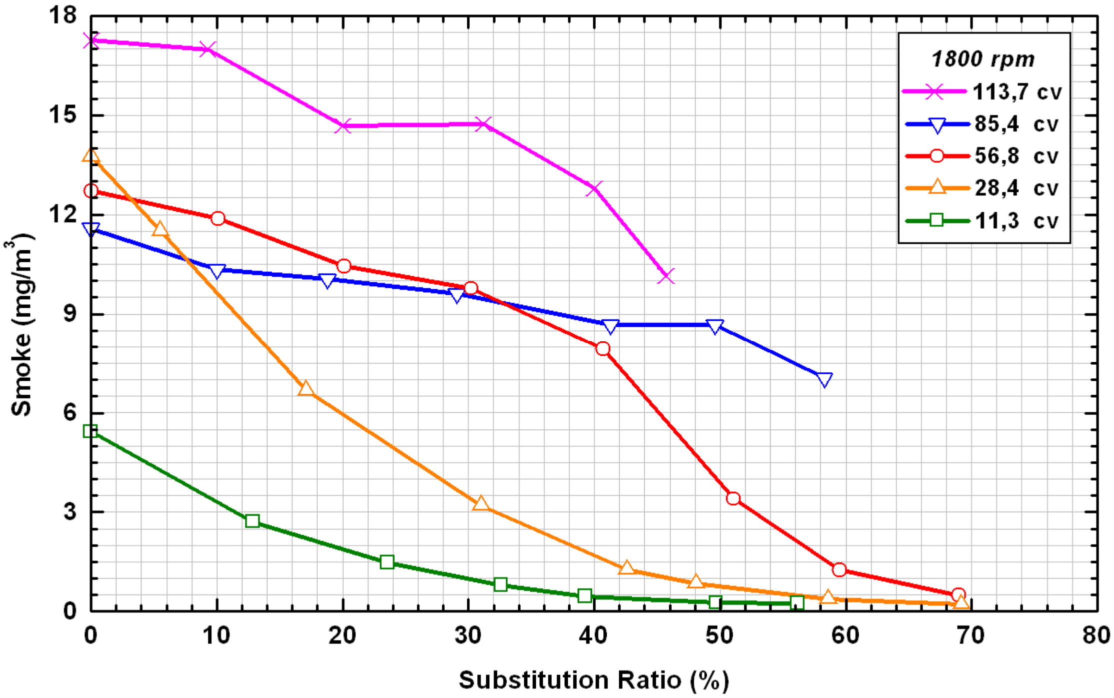

Figure 4. Smoke for different substitution ratios and engine loads. Engine speed: 1800 rpm.

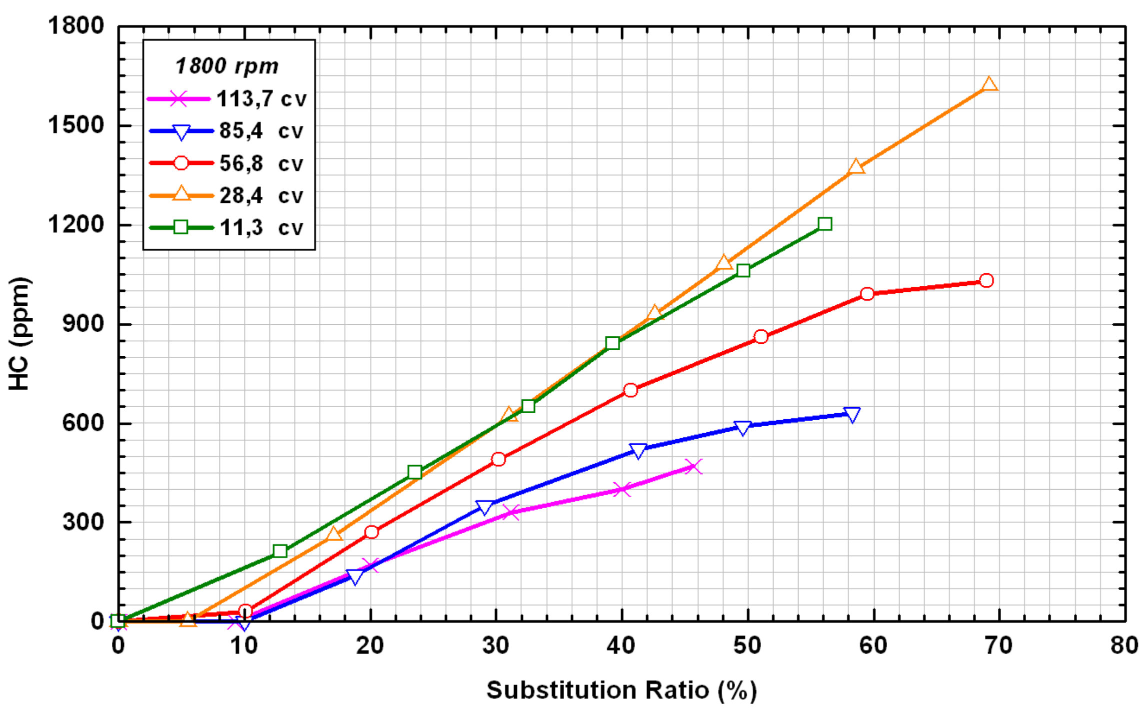

Figure 5. Hydrocarbons for different substitution ratios and engine loads. Engine speed: 1800 rpm.

4. CONCLUSIONS

The experimental investigation conducted in this work identified the performance and emissions characteristics of a alcohol / diesel dual fuel turbocharged aftercooled engine: various new phenomena were observed when compared the corresponding original Diesel engine.

At low loads, for the evaluated engine speed, the brake thermal efficiency under dual fuel conditions is significantly lower compared to original diesel operation. Above these loads, dual fuel engine performance is poorer when more AEHC is added. Only at high loads is the combustion of AEHC more complete and less affected by the substitution ratio, since better BTE results are observed for both operating modes. HC emissions from the alcohol / diesel engine were significantly higher than those of the original Diesel engine. Theses concentrations increase with substitution ratio and a reduction is observed only under high load conditions. Alternatively, over a wide range of the operating conditions (engine speed, load), the dual fuel engine clearly showed the benefits of reduced NO_X and MP emissions; such reductions must be considered if proposed global reductions are to be achieved with conventional Diesel engines.

The results of this study pave the way for future research on optimizing the combustion process of dual fuel commercial engines.

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