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PERFORMANCE AND EMISSION ANALYSES OF SPARK IGNITION ENGINE ON MICRO-GENERATION TO GASOLINE AND NATURAL GAS

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Abstract.

Distributed energy generation to diesel has been more environmentally impacting than other energy sources. In this context, Compressed Natural Gas (CNG) has been widely applied as an alternative fuel to reciprocating Internal Combustion Engines (ICE) for several decades, because it is cheaper and cleaner-burning. Therefore, there was interest in investigating an Otto engine converted either to operate with gasoline or optionally with CNG under different load conditions. Each scenario was analyzed based on the in-cylinder pressure, composition of exhaust gases and fuel consumption. Results have shown that CNG combustion presents lower values of combustion chamber temperature and pressure, Indicated Mean Effective Pressure (IMEP), Specific Fuel Consumption (SFC) and CO, CO₂, HC and NO_x emissions than on gasoline combustion, but a higher thermal efficiency on CNG combustion has been achieved. Finally, it has been concluded that adequately converted engines to operate with CNG have also shown to be appropriate to gaseous fuel combustion, allowing researching other fuels such as biogas.

Keywords: Efficiency, Emissions, In-cylinder pressure, Natural gas, Spark ignition engine.

1. INTRODUCTION

The gradual reduction of the petroleum reserves and the growing need to reduce pollutant emissions have promoted the search and development of alternative and renewable energy sources which help to reduce the current oil dependency. In the field of Internal Combustion Engine (ICE), gaseous fuels have become a good alternative to traditional petroleum derived fuels, such as diesel and gasoline. In this way, natural gas has been successfully used for many years as fuel in engines, due mainly to it is abundant in nature and its extraction is inexpensive (Chandra *et al.*, 2011).

Generally, diesel generators are more cost-effective than gasoline and ethanol, but diesel engines have high particulates and sulfur compounds emissions in their exhaust gases. Compressed Natural Gas (CNG) can be used in generator sets, because this fuel presents a cleaner-burning, a significant pollutants emissions reduction, and a better thermal efficiency compared to diesel and gasoline (de Oliveira *et al.*, 2011).

Since methane (CH₄) is the primary constituent of Compressed Natural Gas, it has a high octane number and a wide flammability range, permitting the development of high compression ratio engines. The thermal efficiency of a lean-burn CNG engine can be improved dramatically thanks to increased specific heat ratio, low combustion chamber temperature, high compression ratio and reduced throttling losses (Manivannan *et al.*, 2003).

CO₂ emissions of CNG engines can be reduced by more than 20% compared with gasoline engines at equal power (Kato *et al.*, 2000). Moreover, engine out unburned hydrocarbons (HC) emissions can also be reduced below the corresponding levels for gasoline engines, since the gaseous state of CNG avoids wall wetting effects on intake manifold and cylinder liner, especially at cold start conditions, which improves cold startability of CNG engines and induces fuel consumption savings (Kato *et al.*, 1999). Also, NO_x emissions from a CNG engine can be reduced below the corresponding levels of gasoline engine due to lower combustion chamber temperature compared to that of gasoline engine (Manivannan *et al.*, 2003).

In this regard, it is important to evaluate the impact of the fuel change between gasoline and CNG on the emissions, combustion and engine performance characteristics to ensure the viability of CNG as engine fuel from an environmental and energetic point of view.

2. METHODOLOGY

2.1 Experimental procedure

The experimental apparatus was composed by a Kohlbach Electric Generator (EG) and a Volkswagen (VW) High Performance (AP) 2.0 engine, whose spark ignition timing is advanced to 15° Before Top Center (BTC). All the main specifications are listed in Tab. 1. The ICE rotational speed is regulated by a Proportional, Integral and Derivative (PID) controller that is widely used in commercial generator sets. The load scenario is managed by connecting the EG to a discrete electric loads system composed of ten 3 kW electric resistances to simulate different scenarios without connecting the EG to the grid. Fig. 1 shows the system diagram and the experimental apparatus utilized in the study.

Table 1. Experimental engine specifications.

Engine type	4 strokes, 4 cylinders
Combustion chamber	cylindrical
Bore x Stroke (<i>mm</i>)	82.5 x 92.8
Displacement (<i>cm</i> ³)	1984
Geometric compression ratio	10
Spark ignition timing	15° BTDC
Fuel injection system	multipoint

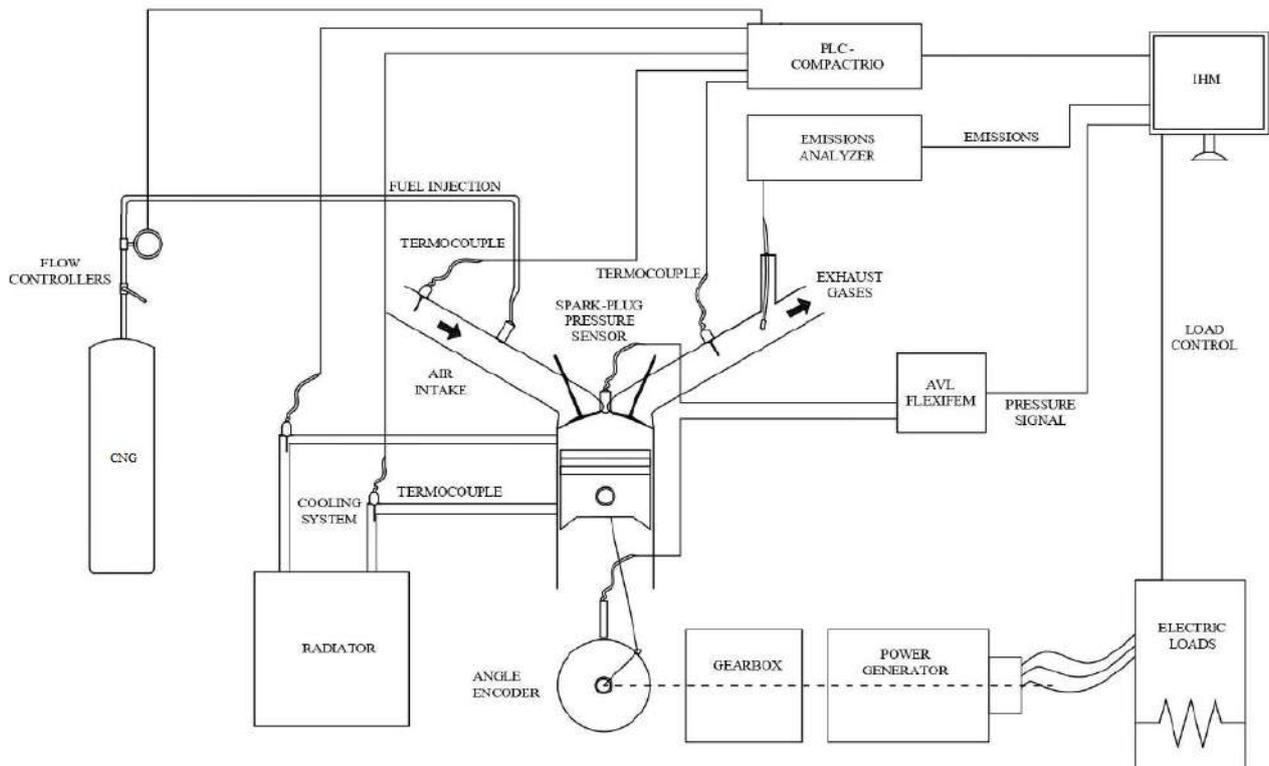


Figure 1. System diagram.

ICE rotation range varies from 0 to 7000 rpm, but there are optimum operating points according to engine application. In the case of a generator set, ICE must operate in engine maximum torque rotational speed. The engine maximum torque lies between 2800 and 3700 rpm, while the EG's rotation must be 1500 RPM to produce 50 Hz power frequency. Then, a set of gears is used between ICE and synchronous generator to assure that the engine's rotation is 3000 RPM, while the EG's rotation is kept at 1500 RPM.

Normally, ICE engines have to maintain a minimal rotation speed necessary to keep all systems running when in idle

mode. However, in the case of this study the PID controller assures that the engine speed is kept constant at 3000 RPM even when there are load and/or fuel variations. An actuator regulates the angle of opening from the throttle valve to keep the intake air flow corresponding to the load required. Finally, by controlling the amount of air injected in the engine, it is possible to control the fuel flow and assure a steady operation of the generator set.

The Compressed Natural Gas injection is a multipoint system and when the ICE runs entirely on CNG the Engine Control Unit (ECU) sends signals to devices that prevent the gasoline and Natural Gas injections at the same time.

A CNG composition was analyzed by a chromatograph (Micro GC 490, Agilent), and the sample data is listed below.

A Crank Angle Encoder (365C, AVL) was attached to the crankshaft to quantify angular displacements up to 0.025°. A spark plug type pressure sensor (ZI45-H7L, AVL) was used in place of the spark plug of cylinder number four to measure the in-cylinder pressure. The FlexIfem acquires and amplifies these instruments measurement signals. Then, it is possible to estimate other parameters from pressure measurement. A gas analyzer (PC-MULTIGAS, NAPRO) is attached to the engine exhaust to measure the volumetric percentage composition of four gases: carbon monoxide (CO), carbon dioxide (CO₂), hydrocarbons (HC) and nitric oxide (NO_x). Finally, flow sensors quantify fuels consumption.

In the experimental study, the engine speed was fixed at 3000 rpm, which enables the system to produce electricity stably and the coolant temperature was maintained below 80 °C. First, the engine ran gasoline in ten different load scenarios: 3 kW up to 30 kW with 3 kW step. Subsequently, the ICE operated with CNG in the same load variation scenarios. As already explained, each scenario operation is analyzed by the in-cylinder pressure, composition of the exhaust gases and fuel consumption.

2.2 In-cylinder pressure data and derivative parameters

Cylinder pressure changes with crank angle as a result of cylinder volume change, combustion, heat transfer to the chamber walls, flow into and out of crevice regions, and leakage. The first two of these effects are the largest. Cylinder pressure is usually measured with piezoelectric pressure transducers. Then, combustion rate information can be obtained from accurate pressure data provided models for the remaining phenomena (Heywood, 1988).

In this work, the In-cylinder pressure was measured over 500 cycles, in order to measure the stability of the combustion by comparing results cycle by cycle. Other engine parameters like Indicated Mean Effective Pressure (IMEP), Rate of Heat Release (ROHR), Integral Heat Released (IHR) and combustion chamber temperature profile are obtained from in-cylinder pressure measurement.

According to Nadaleti and Przybyla (2018), ROHR is given by:

$$\frac{dQ}{d\theta} = \frac{1}{\gamma - 1} \left(V \frac{dp}{d\theta} + \gamma p \frac{dV}{d\theta} \right) \quad (1)$$

Where $\frac{dQ}{dt}$ is the Rate Of Heat Release, γ is the polytropic coefficient and it varies based on temperature and Mass Fraction Burned (MFB), p is the in-cylinder pressure, θ is crank angle and V is the cylinder volume, which, according to Heywood (1988), is given by:

$$V = V_c \left\{ 1 + \frac{1}{2}(r_c - 1) \left[R + 1 - \cos \theta - (R^2 - \sin^2 \theta)^{\frac{1}{2}} \right] \right\} \quad (2)$$

Where V_c is the chamber volume, r_c is the geometric compression ratio and R is the ratio of connecting rod length to crank radius.

Subsequently, the Integral Heat Release can be computed by integrating the release rate in Eq. 1:

$$\int_{t_0}^t \dot{Q} dt \quad (3)$$

The combustion chamber temperature is estimated by ideal gas law according to Yan and Wang (2012):

$$T_{cc} = \frac{pV}{m_{cyl}R} \quad (4)$$

Where T_{cc} is the combustion chamber temperature, R is the ideal gas constant and m_{cyl} is the mass of gas in-cylinder after Intake Valve Closing (IVC), and it is calculated as:

$$m_{cyl} = \frac{\eta_v p_{in} V_{ivc}}{RT_{in}} + \frac{[\beta_0 + (\beta_1 - 1)p_{in}]V_{ivc}}{RT_{exh}} \quad (5)$$

Where p_{in} is the intake manifold pressure, V_{ivc} is the cylinder volume after IVC, T_{in} is the intake manifold temperature, β_0 and β_1 are experimental constants, T_{exh} is the exhaust gas temperature and η_v is the volumetric efficiency. In this study, the gasoline volumetric efficiency is considered 0.9, while for CNG this value is 0.8, which agrees with Pourkhesalian *et al.* (2010) and Ramjee and Reddy (2011).

2.3 Fuel and engine performance parameters

A useful relative engine performance measure is obtained by dividing the work per cycle by the cylinder volume displaced by cycle. This parameter has units of force per unit area and is called the Mean Effective Pressure (MEP) (Heywood, 1988). For 4-stroke engines, IMEP is given by:

$$IMEP = \frac{2000P_i}{V_d N} \quad (6)$$

Where IMEP is Indicated Mean Effective Pressure in kPa, P_i is indicated power in kW, V_d is displacement volume in dm^3 , N is the number of revolutions per second.

Another useful parameter is the Specific Fuel Consumption (SFC) – the fuel flow rate per unit of power output. It measures how efficiently an engine is using the fuel supplied to produce work (Heywood, 1988) and it is expressed by the equation below:

$$SFC = \frac{m_f}{P_b} \quad (7)$$

Where SFC unit is $\frac{g}{kWh}$ and m_f is mass flow fuel in $\frac{g}{h}$.

Futhermore, Heywood (1988) shows another parameter called Thermal Efficiency (η_f):

$$\eta_f = \frac{P_b}{m_f LHV} \quad (8)$$

Where LHV is the Lower Heating Value in $\frac{kJ}{kg}$.

2.4 Emissions characteristics

Levels of emissions of oxides of nitrogen (nitric oxide, NO , and nitrogen dioxide, NO_2 , usually grouped as NO_x), carbon monoxide (CO), unburned hydrocarbons (HC), and particulates are important engine operating characteristics (Heywood, 1988).

According to Heywood (1988), the concentrations of gaseous emissions in the engine exhaust gases are usually measured in parts per million or percent by volume (which corresponds to the mole fraction multiplied by 10^6 and 10^2 , respectively).

3. Results and Discussion

3.1 CNG composition

The CNG composition is shown in Tab. 2. The Compressed Natural Gas used in this study is widely composed by methane (CH_4) as expected and it has 4.689 % in volumetric composition of inert gases (N_2 and CO_2). Other CNG compounds are ethane (C_2H_6), propane (C_3H_8), butane(C_4H_{10}) and pentane (C_5H_{12}).

Table 2. CNG Composition.

Component	Volumetric Percentage
CH_4	91.158
C_2H_6	3.926
C_3H_8	0.094
C_4H_{10}	0.125
C_5H_{12}	0.008
N_2	3.308
CO_2	1.381

3.2 In-cylinder pressure, ROHR, IHR and combustion chamber temperature

In line with Pourkhesalian *et al.* (2010), gasoline pressure peak (Fig. 2) is commonly higher than CNG pressure peak (Fig. 3), which is meeting with this study results. As expected, when load required by EG increases the pressure peak (Fig. 2 and Fig. 3) grows too.

According to Kim *et al.* (2016) and meeting with the results, the ROHR peak is lower and ROHR graph is wider in CNG (Fig. 5) than gasoline (Fig. 4). The main reason is the inert gases which compose CNG, which absorbs fuel released

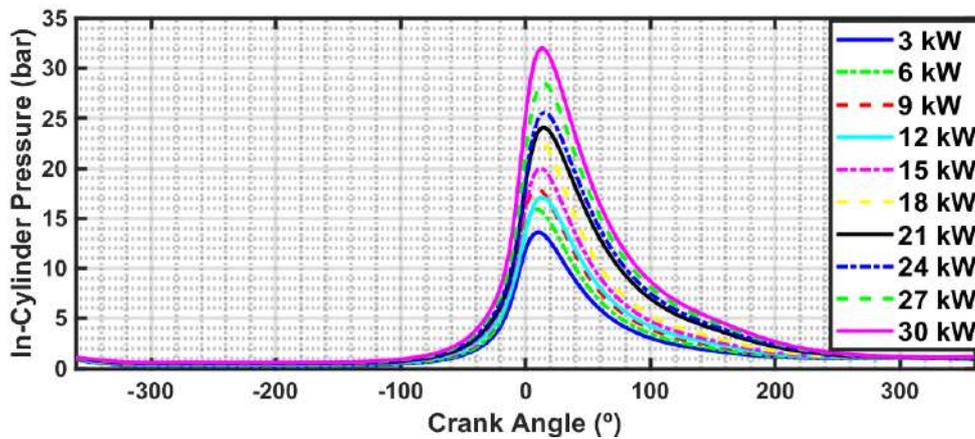


Figure 2. In-cylinder Pressure for Gasoline.

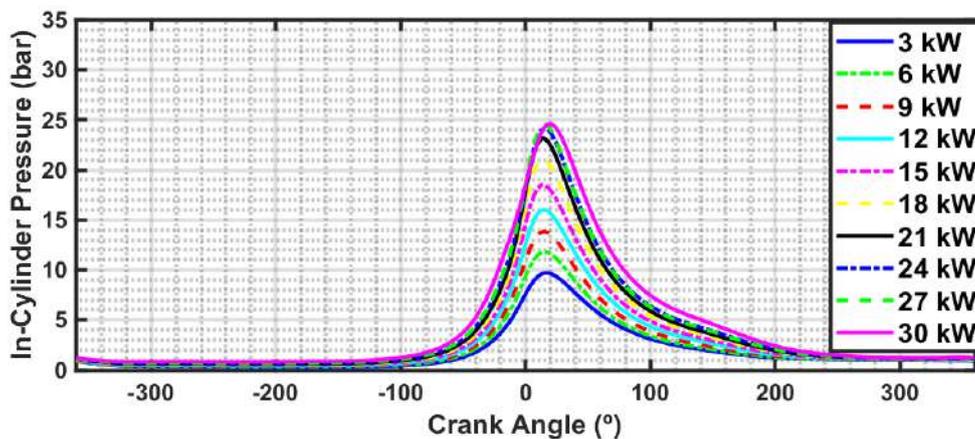


Figure 3. In-cylinder Pressure for CNG.

energy. As shown by Tahir *et al.* (2015), another relevant point is that the amount of mass injected in the manifold is greater in gasoline tests due to the higher density. Due to the higher amount of fuel mass in the mixture and despite of the lower LHV towards CNG, the total amount of energy released and the gases temperature were higher in gasoline tests (Fig. 6 and Fig. 8).

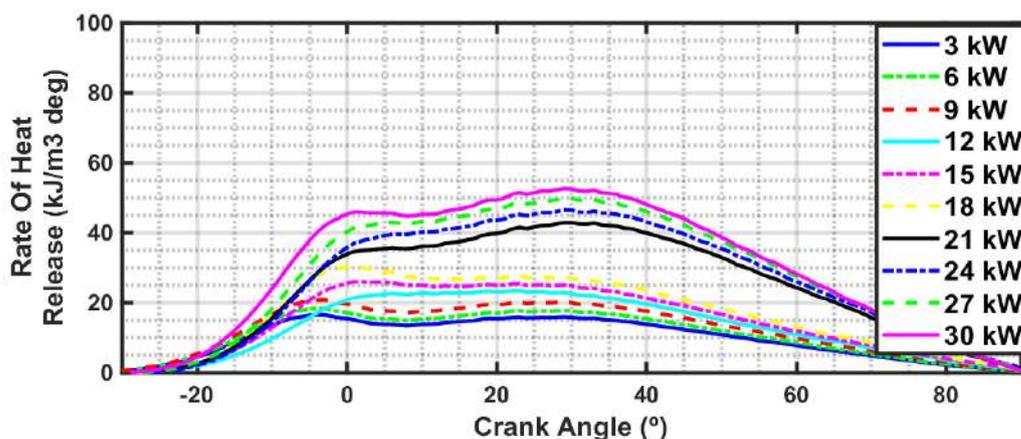


Figure 4. ROHR for Gasoline.

Analyzing ROHR (Fig. 4 and Fig. 5) and IHR (Fig. 6 and Fig. 7) graphs, it is observed that these inert gases also provoke a delay in the combustion, because they create misfires zones. Another factor that implies into the wider ROHR graph for CNG is that this fuel has a lower flame speed than gasoline, besides suffering the effects of inert gases, increasing the combustion duration, which can also be seen in IHR graph. Table 3 shows that CNG combustion start timing (5%

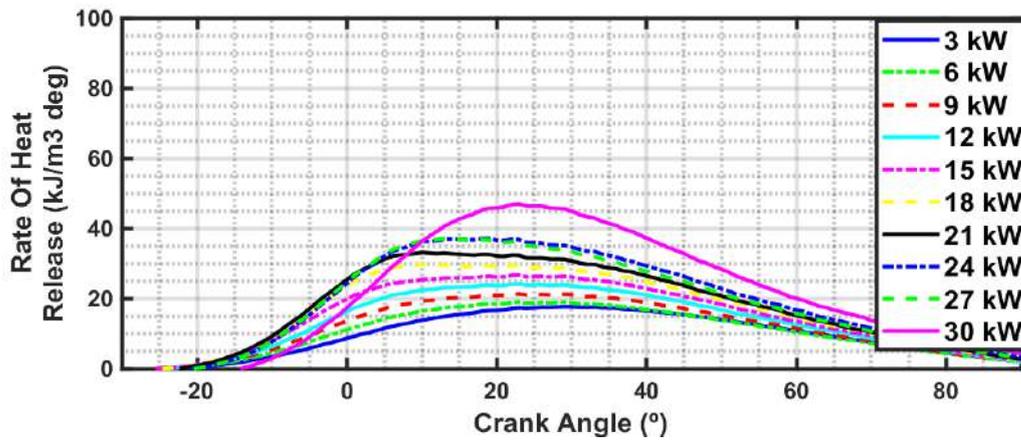


Figure 5. ROHR for CNG.

MFB) is slower than gasoline for all load scenarios because CNG has lower flame speed.

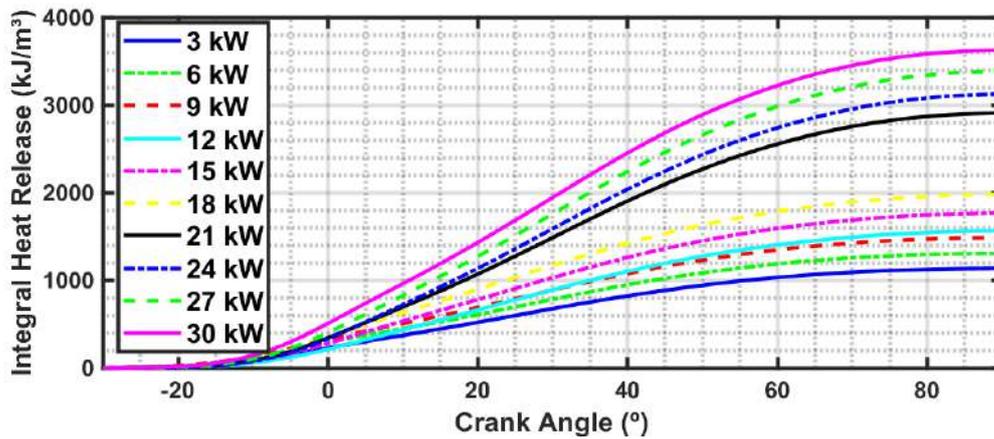


Figure 6. IHR for Gasoline.

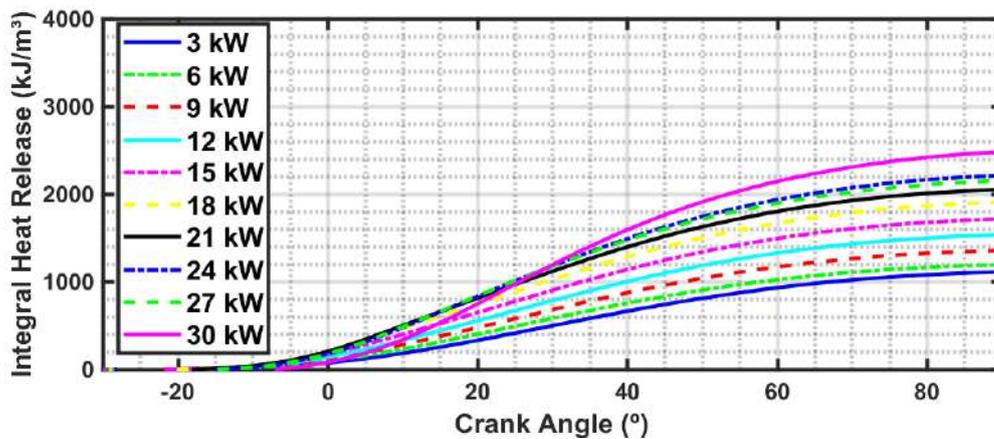


Figure 7. IHR for CNG.

The combustion chamber temperature at the end of combustion is lower in CNG (Fig. 9) than gasoline (Fig. 8). Although gasoline volumetric efficiency is greater than CNG, this behavior is due to the high amount of inertial gases in CNG composition and the lower amount of fuel injected, since the mass flow in gasoline tests are greater than CNG tests, as already been discussed.

Table 3. Combustion start.

Load (kW)	Gasoline (°)	CNG (°)
3	-12.5	-2.15
6	-14.15	-3.75
9	-13.9	-4.5
12	-7.95	-4.6
15	-9.05	-5.4
18	-9.65	-5.3
21	-6.75	-4.9
24	-5.8	-3.05
27	-6.85	-3.25
30	-8.45	-2.35

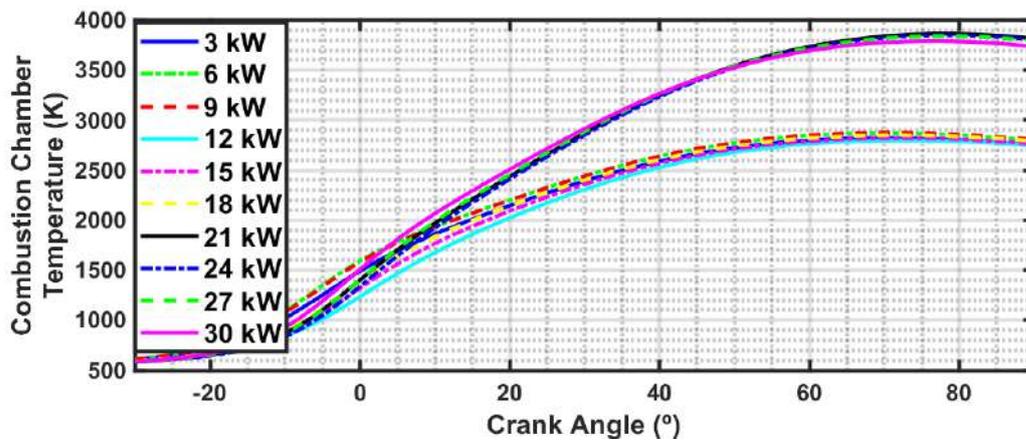


Figure 8. Combustion chamber temperature for Gasoline.

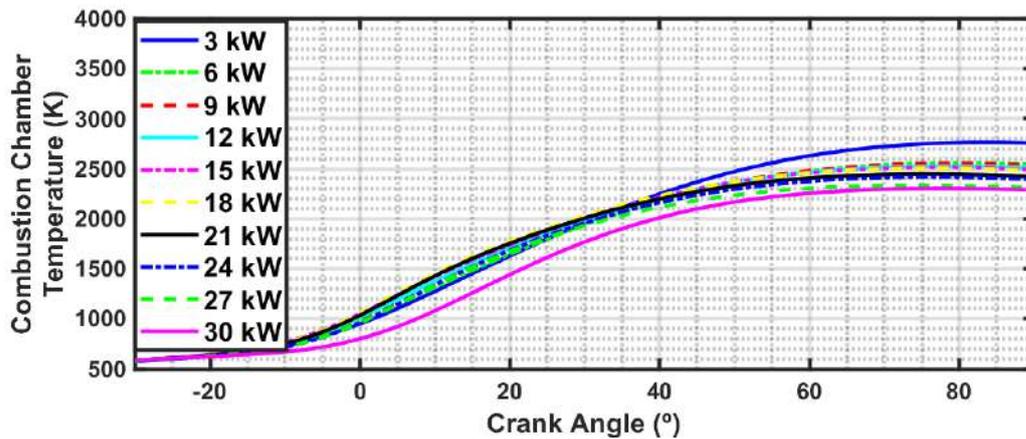


Figure 9. Combustion chamber temperature for CNG.

3.3 Performance parameters: IMEP, SFC and thermal efficiency

As expected, the IMEP (Fig. 10) is increasing with high load scenarios, since more work is demanded. According to Pourkhesalian *et al.* (2010), Ameri *et al.* (2012) and Ramjee and Reddy (2011), the Indicated Mean Effective Pressure at low engine speeds (lower than 4000 rpm) has close values in gasoline and CNG as can be seen in Fig. 10. The greater values of IMEP for gasoline are related with the greater pressure values (Fig 2 and Fig. 3) due to higher amount of heat released by gasoline (Fig. 6).

Specific Fuel Consumption is lowering with load increase (Fig. 11), because of the rising in engine efficiency (Fig. 12) in converting fuel energy into power. According to Pourkhesalian *et al.* (2010), Compressed Natural Gas consumption is expected to be smaller (and the efficiency higher) than gasoline consumption because LHV of methane (the primary constituent of CNG) is higher than gasoline, so the same heat is released with less amount of fuel. Then the total heat

released by CNG is lower than gasoline, as already shown, but the energy released per mass is larger, reflecting in thermal efficiency and SFC improvements.

According to Tambari *et al.* (2014) and Ameri *et al.* (2012), the thermal efficiency of CNG is greater than gasoline, which agrees with the study results (Fig. 12). Ameri *et al.* (2012) says natural gas has better efficiency compared to gasoline owing to its endurance at higher octane numbers, despite having lower volumetric efficiency.

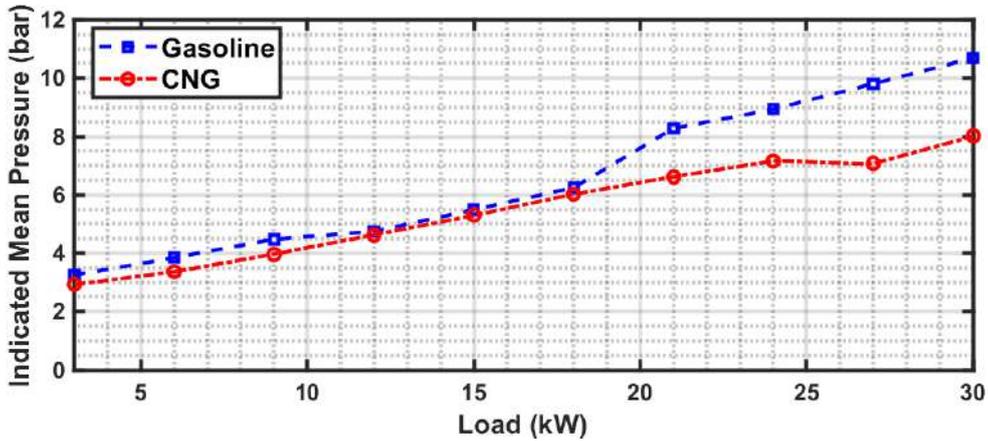


Figure 10. IMEP for Gasoline and CNG.

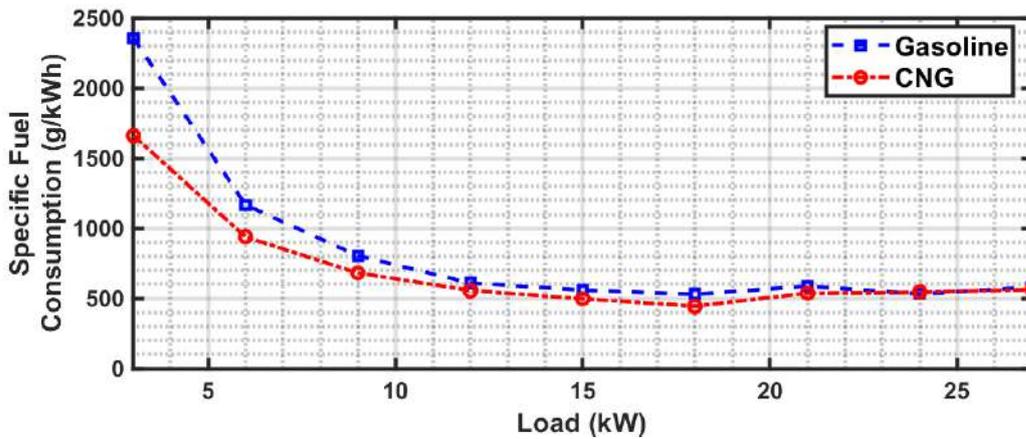


Figure 11. SFC for Gasoline and CNG.

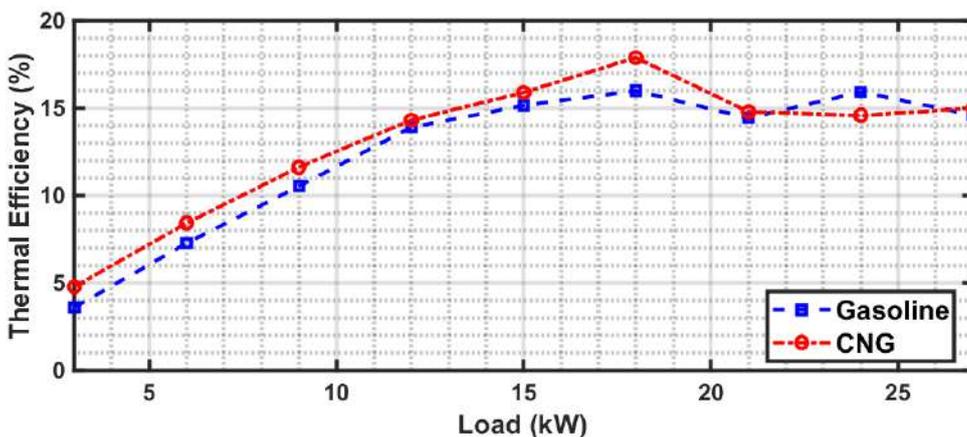


Figure 12. Thermal efficiency for Gasoline and CNG.

3.4 Emissions

The emission chart below (Fig. 13) shows a decreasing level of CO contents with the rising load for both fuels. This occurs due to the thermal efficiency (Fig. 12) growth and, consequently, the CO oxidizing process to CO_2 is intensified, which explains too the rise of CO_2 emissions (Fig. 13). Also, HC emissions (Fig. 14) are lowering by the same reason of CO emissions, the high thermal efficiency values, which leads to more complete combustion process and less unburned material. The CO_2 emissions were at a higher level in gasoline due to carbon-hydrogen ratio. Gasoline has nearly 1.87 hydrogen atoms for each carbon atom. CNG, that can be modeled like its major component, methane, has 4 hydrogen atoms for each carbon atom. Then, stoichiometrically, the gasoline produces more CO_2 molecules per fuel mol. Meeting with the results, Barcellos *et al.* (2003) shows that CO and HC emissions decrease with the fuel change from gasoline to Compressed Natural Gas, because CNG has more complete combustion than gasoline.

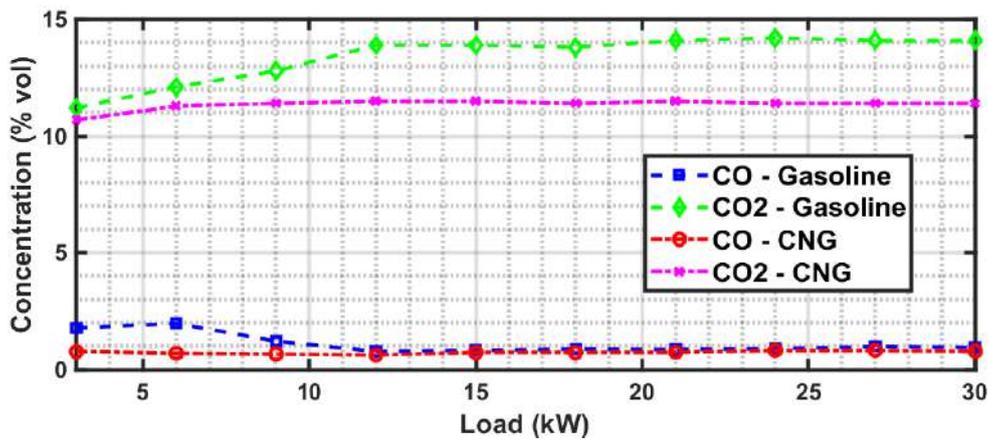


Figure 13. CO and CO_2 emissions for Gasoline and CNG

In contrast, NO_x emissions (Fig. 14) are increasing with load rising for both fuels due to NO_x formation takes place at high temperatures (Pourkhesalian *et al.*, 2010).

In this sense, NO_x emissions of CNG is lower than gasoline, because the combustion chamber temperature levels for CNG (Fig. 9) are lower than for gasoline (Fig. 8). This temperature difference is explained by inert gases in CNG composition, as already discussed. However, these results are different from those conventionally found in the literature, because gasoline and CNG tests are usually performed under different conditions of spark ignition timing. This decision is usually taken to compensate the combustion delay caused by the low flame speed of methane (Pourkhesalian *et al.*, 2010).

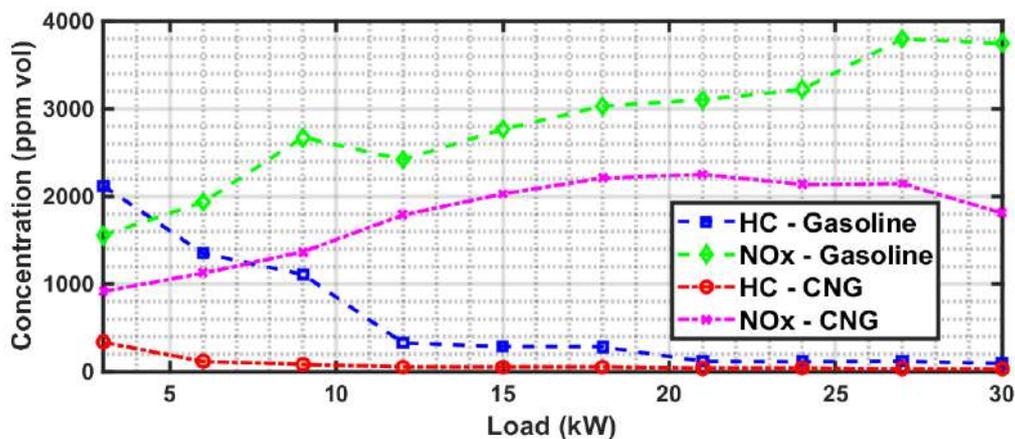


Figure 14. HC and NO_x emissions for Gasoline and CNG

4. CONCLUSION

The combustion behavior and emissions of a spark-ignition engine were studied by varying the fuel and the load of the engine. It was observed that in general, CNG is a good substitute for fossil fuels in terms of power generation, showing lower HC , CO and NO_x emissions and it has relatively stable combustion.

- The combustion delay is more significant in CNG than gasoline;
- Combustion chamber temperature is lower in CNG due to smaller heat released;
- The gasoline shows higher IMEP levels because of the greater in-cylinder pressure;
- CNG proved more efficient in converting fuel into energy, presenting lower SFC values.

Finally, it was concluded that the converted engine operates adequately with CNG and this conversion is appropriate to investigate gaseous fuels combustion, such as biogas.

5. ACKNOWLEDGEMENTS

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