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Emission Comparative Analysis of a Bi-fuel Engine Equipped with Transistorized Ignition System

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Authors' contributions

This work was carried out in collaboration between both authors. Both authors read and approved the final manuscript.

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ABSTRACT

As a result of the negative impact of transportation emissions from locomotives, trucks, passenger cars and others on the environment, people's health as well as animals, plants and buildings, there is need to turn to a more environmental friendly fuel for the transport system One of the friendly fuel is to use the compressed natural gas (CNG) in the transportation means, where CNG is cheap and easy obtainable. On the other hand, the transistorized ignition system can gives high voltage in its spark. However, the aim of the present work is to analysis the exhaust gas emissions produce from bi-fuel vehicle engine equipped by either conventional or transistorized breakerless ignition system at cold start, warm-up or waiting for traffic lights. The study involves the measurements of engine

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exhaust gas emission factors such as total hydrocarbon (THC), carbon monoxide (CO), carbon dioxide (CO_2) at no load, while the engine is running under two modes. In the first mode, the engine run on gasoline, while in the second mode, the engine runs on CNG. By using this concept, the influence of the transistorized ignition when used in the bi-engine which runs in the two modes considered on the exhaust gas emission factors has been identified. The results indicate that a significant reduction in exhaust gas emission factors are obtained, the values of THC, CO and CO_2 measured when the engine was equipped by transistorized ignition system are lower than those measured when the engine was equipped by conventional ignition system.

Keywords: Compressed natural gas (CNG); carbon monoxide (CO); carbon dioxide (CO2); total hydrocarbon (THC); transistorized; breaker-less; conventional ignition.

1. INTRODUCTION

The emphasis to date in investigations of lean misfire limit performance has been on the effects of in-cylinder air motion. Several studies have examined the use of various intake port and valve configurations and their effect on the combustion process [1-5]. These studies have shown that increasing the organized charge motion in the cylinder leads to increased turbulence levels. Higher turbulence levels speed up the combustion process, which reduces cyclic variability at lean conditions. Increased charge motion can be obtained through the use of intake ports designed to produce swirl or tumble, or through the use of shrouded intake valves which produce swirl. Various turbulence enhancing combustion chamber geometries have also been studied [8-9]. In particular, pistons, which have unusual bowl shapes designed to generate squish flows or enhance the effects of swirl, have been investigated [10]. Another area, which has received attention, is the application of high-energy ignition systems designed to ignite lean mixtures. Although these enhanced ignition systems have been shown to extend the lean misfire limit, they do possess some disadvantages compared to production systems a key disadvantage is spark plug life; since these high energy systems have to dissipate greater power levels, spark plug life is typically reduced.

One of ignition control strategies is to raise the temperature and the density of the charge by firing a spark plug and start a flame propagating that will further compress the unburned mixture. This would be a very slow flame in the lean, low turbulence mixture and would not have time to propagate across the chamber before conditions in the unburned gases reached the auto ignition point. This type of operation, if required, would in effect be an Otto cycle engine process with “soft knock” due to the very lean operation. Another alternative ignition system is to use direct inject either neat top dead centre (TDC) or at the intake valve a small amount of high cetane fuel to promote earlier auto ignition [6-7].

Historically, gaseous-fueled engines, developed from automotive gasoline engines have used automotive distributor (inductive) ignition systems. Larger industrial/stationary gas engines use magneto generator powered capacitor discharge ignition systems. Often, ignition timing is either fixed or varied only with RPM (speed advance) and sometimes with ignition timing shift based on manifold vacuum of naturally aspirated engines. As electronically controlled ignition systems were developed for both automotive gasoline and industrial gas engines, either transistorized or distributor-less ignition were introduced with timing control determined by simple timing curves. Eventually, calibratible speed and load based on ignition timing using advanced microprocessor controllers were developed.

The effects of hydrogen addition on the natural gas engine operation have been studied [11]. According to the results, adding hydrogen into the CNG-air mixture had negative impact on the combustion delay and increased the combustion burning rate. In addition, the development of a quasi-dimensional model for analysis of combustion process was made in spark ignition (SI) prechamber natural gas engine. Two submodels to simulate turbulence intensity in cylinder and modeling of jet orifices in prechamber. They verified their simulation code with experimental data [12]. Performance and emission characteristics of a bi-fuel Ricardo single cylinder SI research engine have been comparatively with gasoline one. The results show 12 percent reduction of power and 5-50 percent reduction of emissions when the engine is fuelled by natural gas. A developed General Motors 2.2L CNG bi-fuel passenger car where a computer engine simulation model has been used which was able to predict engine performance, fuel consumption and emissions to reduce system calibration time as well as the cost of testing had been experimented. According to the results of the experiments, CNG engines showed significantly lower non-methane organic gases, carbon monoxide (CO) in their emissions than gasoline operated engines. The lean burn strategy for reducing emissions of natural gas SI engines, where experiments included the study of performance and emissions characteristics of an SI lean burn natural gas engine [13]. In ref. [14-15], effects of the fuel composition, combustion chamber geometry, combustion modeling, burning rate models, pre-chamber and after-treatment on these engines have been considered.

However, the aim of the present work is to evaluate the exhaust gas emissions for a bi-fuel engine equipped by transistorized ignition system. The study involves the measurements of engine exhaust gas emissions such as total hydrocarbon (THC), carbon monoxide (CO), carbon dioxide (CO2), while the engine is running under two phases. In the first phase, the engine run with gasoline while in the second phases the engine runs in CNG. The engine speed varies from 500 revelation per minute up to 4000 revelation per minute. In all cases, the engine runs without load. This condition can be observed in the cases of start, warm-up or waiting for traffic lights.

2. CNG-GASOLINE FUEL CHARACTERISTICS

CNG is a mixture of several gases, the main component of, which is Methane, which exists in concentration of 80% to 90% by volume. In fact, the composition of natural gas is not constant and can vary of source to source and also in the same site at different times. Non-methane hydrocarbons are composed mainly from Ethane and lower concentrations of Propane and Butane. Small concentrations of inert gases such as Nitrogen and carbon dioxide are also often present. The composition and properties of CNG and gasoline used in these tests were obtained from Refs [16-18]. Gasoline properties are shown in Tables 1 and 2. Natural gas properties and composition are shown in Tables 3 and 4 (test method: ASTM D-1945-03).

Table 1. Gasoline composition

Component	Symbol	Mass fraction*100
Carbon	C	85.65
Hydrogen	H	12.94
Oxygen	O	1.39
Sulphur	S	0.0003

Table 2. Thermodynamic properties of gasoline

Stoichiometric ratio	14.19
Octane number	95.8
Higher heating value (MJ/kg)	45.03
Lowe heating value (MJ/kg)	42.21
Density @ 25ºC (kg/m^3)(DIN 51757)	749
Molecular weight (kg/kmol)	106.22

Table 3. Thermodynamic properties of natural gas

Stoichiometric ratio	16.5
Higher heating value (MJ/kg)	50.79
Lowe heating value (MJ/kg)	45.71
Molecular weight (kg/kmol)	18.10

Table 4. Natural gas compositions

Component	Symbol	Volumetric %
Methane	CH_4	88.2
Ethan	C_2H_6	4.2
Propane	C_3H_8	1.36
Butane	C_4H_{10}	0.3
Iso-Butane	C_4H_{10}	0.28
Pentane	C_5H_{12}	0.06
Iso-Pentane	C_5H_{12}	0.09
Hexane	C_6H_{14}	0.03
Carbon dioxide	CO_2	0.3
Nitrogen	N_2	5.2

3. ENGINE IGNITION SYSTEM

Engine using gasoline or compressed natural gas needs a spark to start Engine Ignition System combustion process. The spark timing is critical if maximum power and fuel economy are required. History of combustion internal engine shows that the jump-spark system used was gradually developed through the stage of hot tube, break spark and tremble coil, with each step showing a definite improvement over its previous one. The conditions inside the engine's cylinder at the time of ignition govern the voltage required to produce a spark where as it needs only a few hundred volts to make a spark jump across the spark plug gap. Nowadays, most ignition systems are capable of supplying the required voltage. Fig. 1 presents a block diagram illustrating such systems both for conventional and electronic [19-20].

4. EXPERIMENTAL APPARATUS AND PROCEDURES

Emissions and performance characteristics of the bi-fuel engine are measured over a wide range of engine speeds according to ISO-1585 testing procedure. Test facilities consist of:

- Four cylinder SI engine
- Eddy current dynamometer, Ricardo FE 760-S

- Exhaust gas analyzer, Pierburg HGA 400
- Fuel temperature control device, AVL 753
- CNG mass flow meters, Emerson micro motion elite sensor
- Mazda on-board diagnostics (OBD II) device
- Data acquisition system, Ricardo
- CNG kit, PRINS (VSI)
- CNG storage

The engine and dynamometer specifications are listed in Tables 5 and 6. The test engine is converted from a gasoline engine to a bi-fuel (CNG + gasoline) engine and equipped with a suitable bi-fuelling system. In order to achieve desired data, sensors were mounted in suitable positions. Data were collected simultaneously from sensors and sent to a data acquisition system. Also, data from exhaust gases were recorded, total unburned hydrocarbons (THC), CO and CO2 in exhaust emissions. Electronic control unit (ECU) data such as injection time, injection duration and spark advance were monitored by Mazda OBD II device.

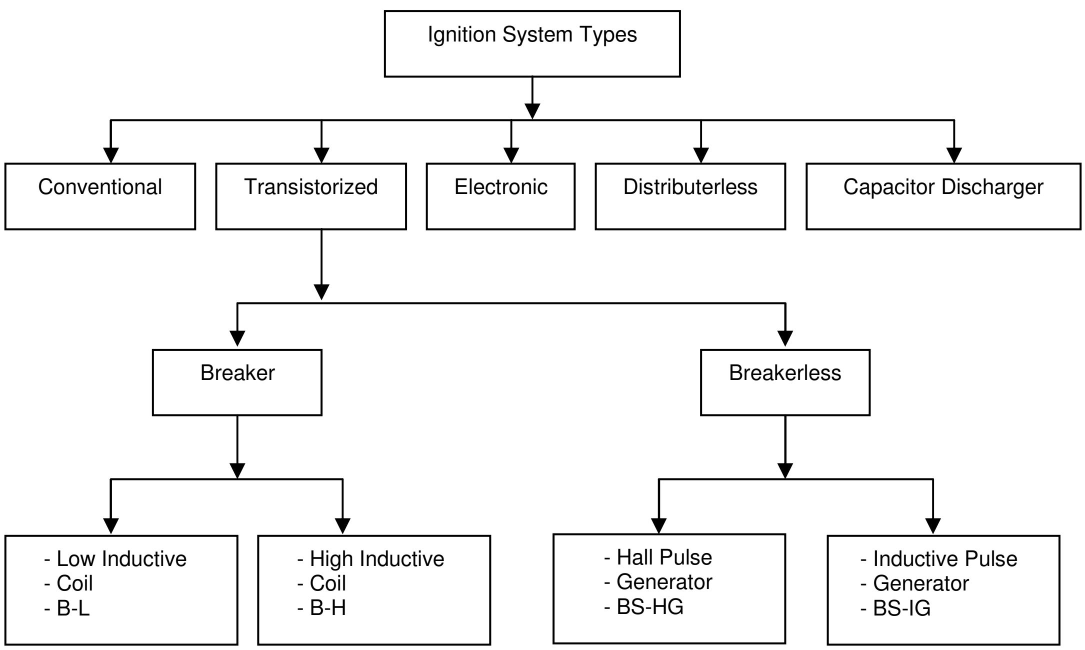

Fig. 1. Block diagram for ignition system types

The engine was equipped with a fuel mixer located upstream of the throttle. Fuelling, ignition and boost pressure were controlled manually. The layout of test engine installation is shown in Fig. 2. The engine was installed in a test cell and connected to the dynamometer at no load. Gaseous exhaust emissions were measured using state-of-the-art equipment, including non-dispersive infrared (NDIR) analyzers for CO2 and CO, a paramagnetic O2 analyzer, a flame ionization detector (FID) for unburned hydrocarbon (THC). The ignition system used in this work is breaker less transistorized type.

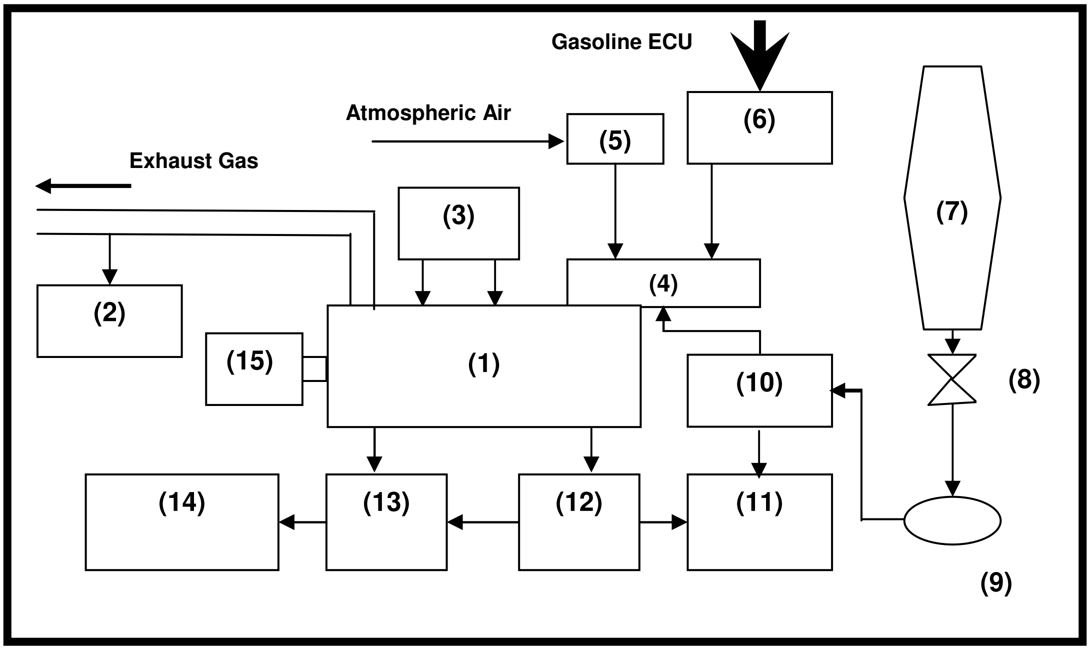

Fig. 2. The layout of test engine installation

1- Engine under test, 2- Exhaust gas analyzer, 3- Ignition systems, 4- Intake manifold, 5- Intake air, 6- Gasoline injection, 7- CNG tank, 8- CNG valve, 9- CNG regulator, 10- CNG injection,11- CNG ECU, 12- Gasoline ECU, 13- Switch box, 14- Data acquisition system, 15- Engine Dynamometer

Table 5. Engine specifications

Engine type	**Four stroke, spark Ignition**
Induction	Naturally aspirated
Number of cylinder	4 cylinder – In line
Bore (mm)	86
Stroke (mm)	86
Connecting rode length (mm)	153
Displacement volume (cm^3)	1998
Compression ratio	8.6
Max. power	70 kw @ 5000 rpm
Max. torque	151 N.m @ 2500 rpm
Valve per cylinder	3
Intake valve opening	10º BTDC
Intake valve closing	49º ATDC
Exhaust valve opening	55º BTDC
Exhaust valve closing	12º ATDC

Portable version of infrared gas analyzer is used during the experimental work. The gas analyzer is equipped with gas sampling probe to collect the exhaust gas from the muffler. The gas is then filtered and dried before entering the analyzer. Magnetic inductive pickup transducer is used also to measure the engine rotational speed in rpm. It is clipped to any of spark plugs cable in order to capture the spark signal. The sparking rate is then considered as linear proportion to the engine speed. Tests have been done for both CNG and gasoline fuels under engine steady state conditions. When CNG kit was installed on the engine, calibration was done for CNG operation. CNG kit consisted of: pressure regulator, common rail injector, CNG ECU, spark advancer, emulator, CNG filter and fuel exchange switch.

Table 6. Dynamometer specifications

Dyno. type	Ricardo FE 760-S
Max. torque (N.m)	610
Max. speed (rpm)	12000
Max. power (kw)	191.17
Inertia (kg/m^2)	0.176
Torsional spring (N.m/rad)*1000	239
Weight (kg)	474

5. TEST RESULTS AND DISCUSSION

The engine has been tested for CNG and gasoline over a range of 500-4000 rpm engine speeds. The tests have been done in full load and part load conditions. Various data such as engine performance parameters, exhaust emissions, pressures and temperatures in some critical points and ECU data have been measured. More details about the layout of the test rig can be found in [10].

In this section, the effect of fuel type on engine exhaust gases has been considered. The presented results show emissions before. Figs. 3 to 5 show relationship of CO2 and CO to engine speed for CNG and gasoline fuels in full load condition. The amount of CO2 in combustion of hydrocarbons is proportional to carbon to hydrogen ratio. The main component of natural gas is methane which has the lowest carbon to hydrogen ratio compared to other hydrocarbons. Therefore, the resulting CO2 in CNG combustion is less than gasoline. The amount of CO is a function of air-fuel ratio. In fact, as air-fuel ratio gets closer to stoichiometric condition, the amount of CO emission becomes less. The air-fuel ratio of CNG fuelled engine is closer to stoichiomertric condition consequently CO emissions are decreased with CNG.

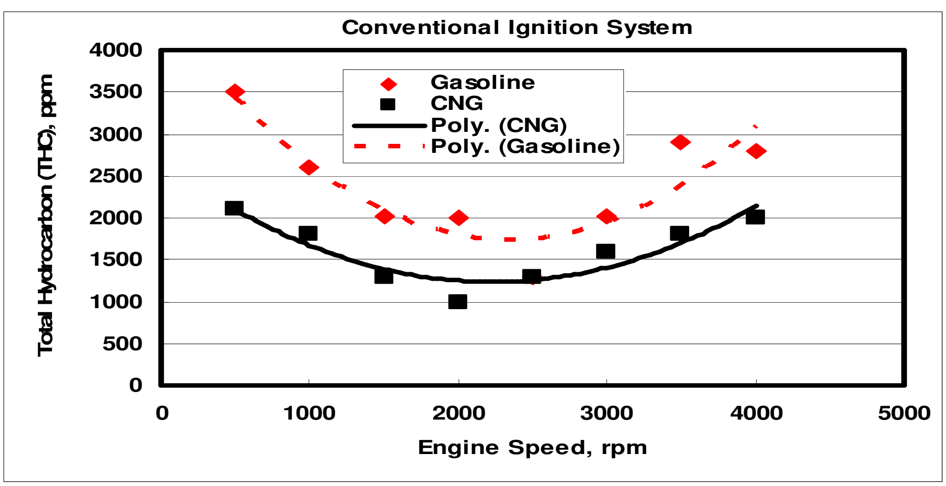

Fig. 3. Comparison of THC in exhaust gases for CNG and gasoline fuels

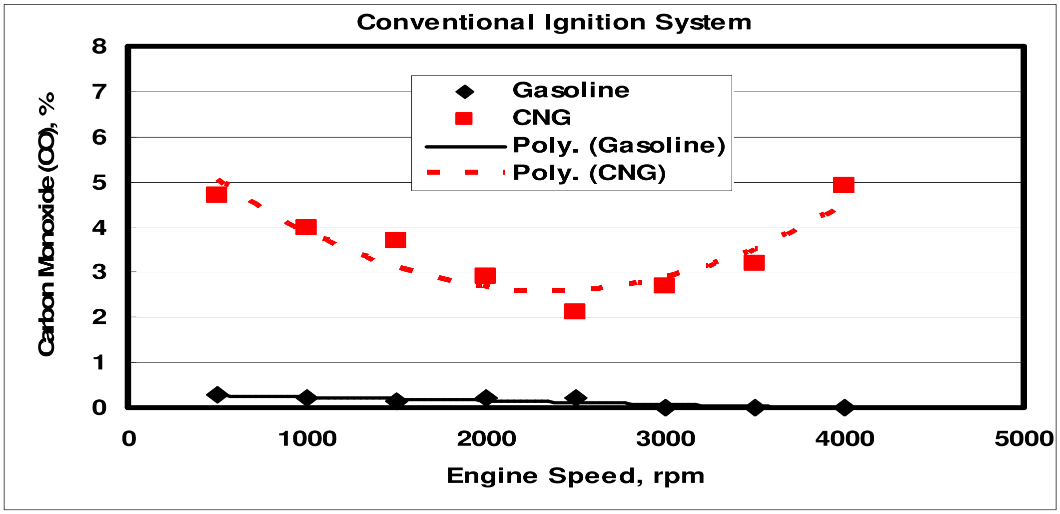

Fig. 4. Comparison of CO in exhaust gases for CNG and gasoline fuels

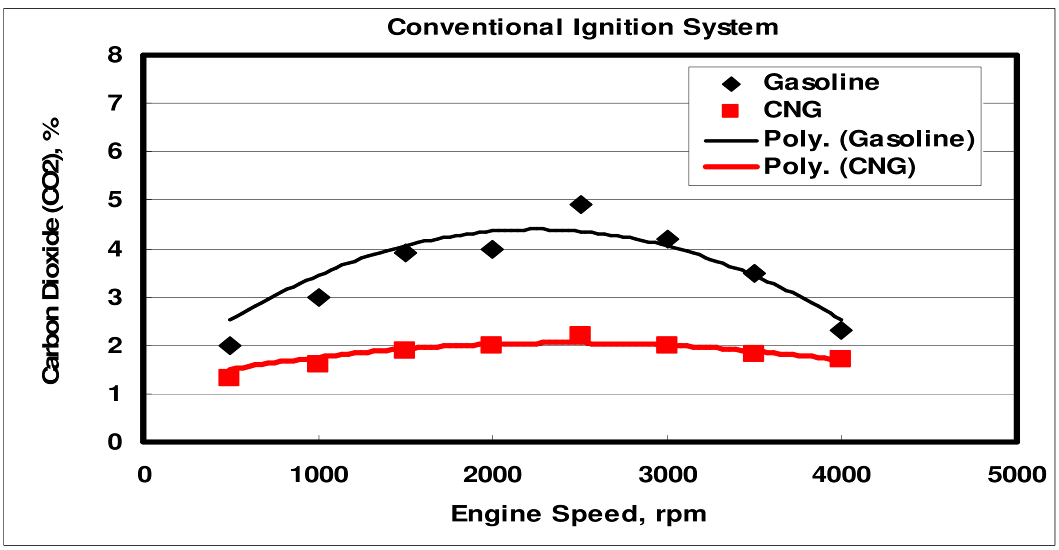

Fig. 5. Comparison of CO_2 in exhaust gases for CNG and gasoline fuels

Figs. 6 to 8 show the value of THC, CO and CO_2 measured for the bi-fuel engine when equipped by either convention or transistorized ignition system with respect to engine rotational speed. The fuel is being compressed natural gas (CNG). In general, the results shown in these figures indicate that the values of THC, CO and CO_2 measured when the engine was equipped by transistorized ignition system was lower than those measured when the engine was equipped by conventional ignition system. The reduction here is relatively higher than those for the conventional ignition system.

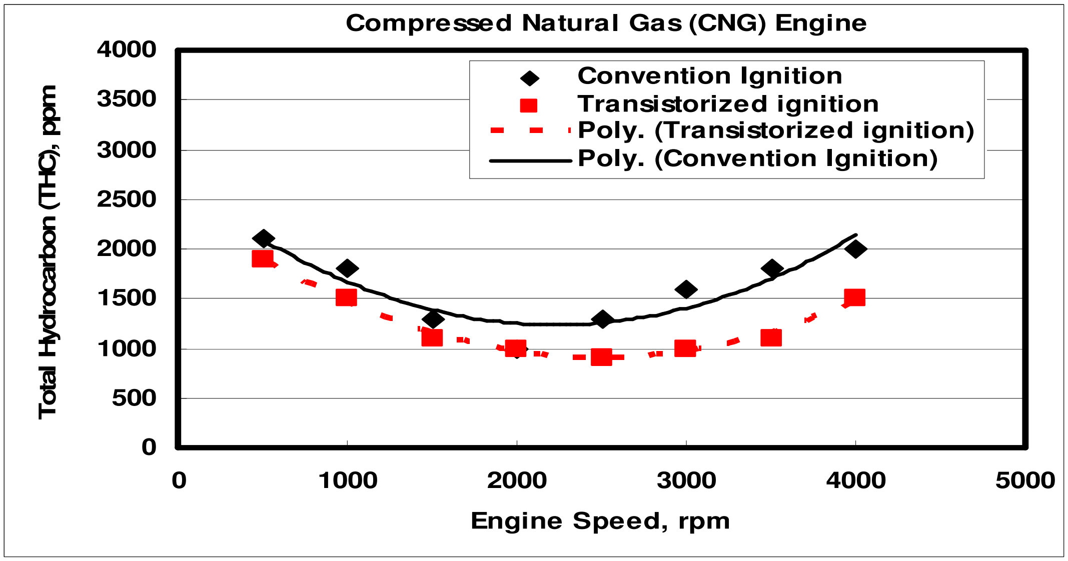

Fig. 6. Comparison of THC in exhaust gases for conventional and transistorized Ignition systems

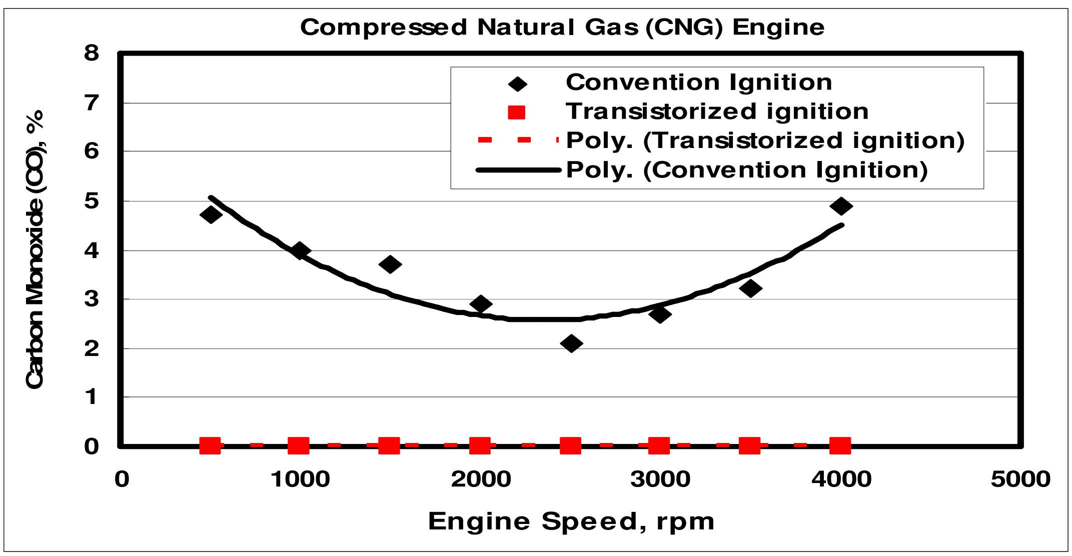

Fig. 7. Comparison of CO in exhaust gases for conventional and transistorized Ignition systems

Figs. 9 to 11 depict the averages of engine emission factors for both fuels and for both ignition systems. It is shown that the average of CO and CO_2 emission factors measured for gasoline phase are lower than those for CNG phase Figs. 10 and 11. Consider Fig. 10 for CO component where the amount of CO is a function of air-fuel ratio. In fact, as air-fuel ratio gets closer to stoichiometric condition, the amount of CO emission becomes less. The air-fuel ratio of CNG fuelled engine is closer to stoichiomertric condition, consequently CO emissions are decreased with CNG. On the other hand, the transistorized ignition system gives accurate spark timing for a much longer period which allow good combustion process particularly for CNG fuel.

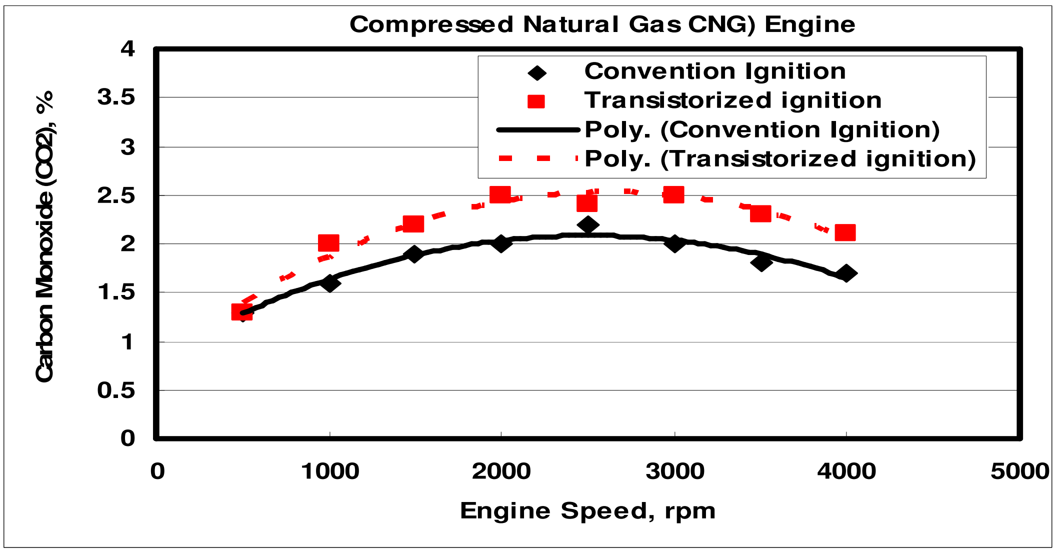

Fig. 8. Comparison of CO_2 in exhaust gases for conventional and transistorized Ignition systems

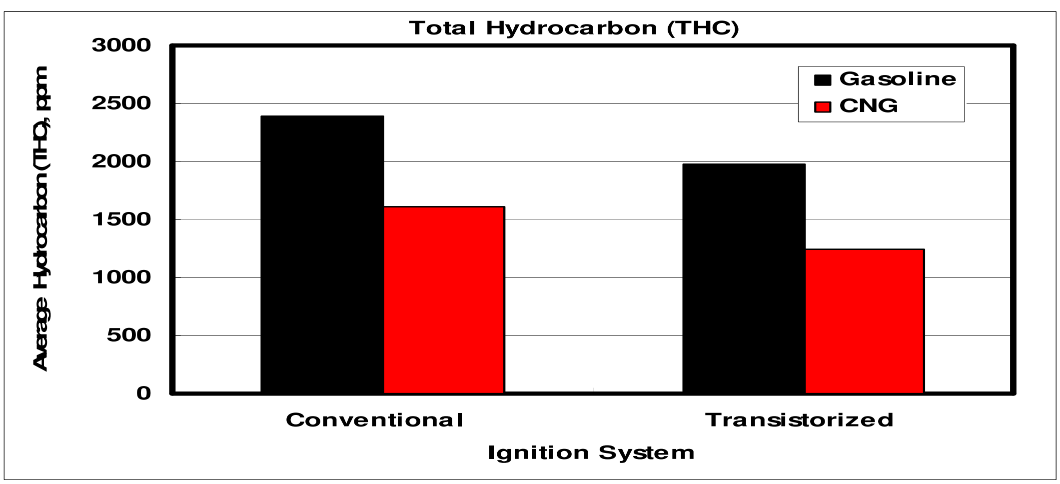

Fig. 9. Comparison of average THC for CNG and gasoline fuels- for conventional and transistorized Ignition system

In Fig. 9 the THC emission factor measured in CNG engine phase is shown, where its average value is lower than that for gasoline. According to these figures, there are some reductions in the THC concentration with CNG operation. These reductions are due to higher temperatures of combustion and exhaust gases and lower fuel trapping phenomenon in crevices while engine operates with CNG.

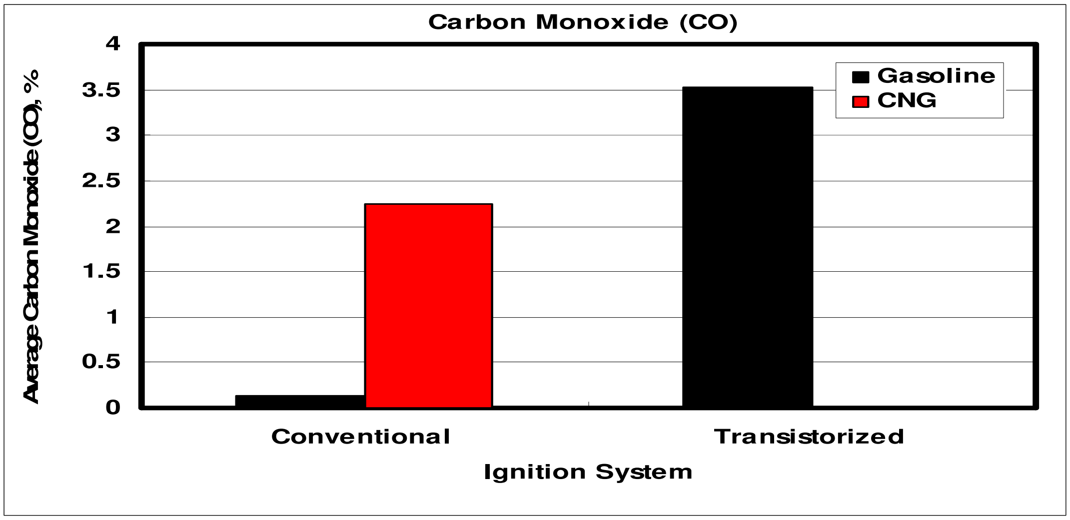

Fig. 10. Comparison of averages CO for CNG and gasoline fuels- for conventional and transistorized Ignition system

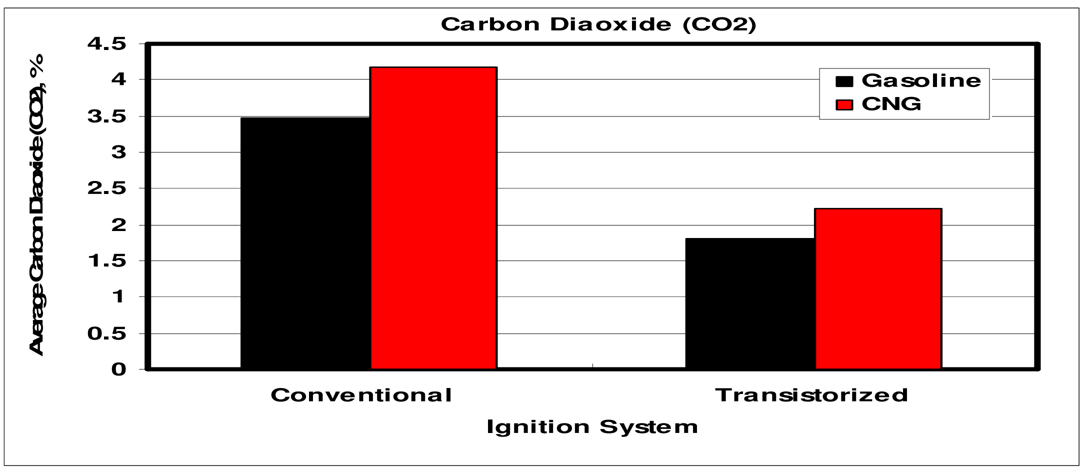

Fig. 11. Comparison of averages CO_2 for CNG and gasoline fuels- for conventional and transistorized Ignition system

The associated uncertainty for the measured emission components are of ±2.5 ppm for CO, ±2.0% for CO_2 and ±2.66 ppm for HC.

6. CONCLUSION

1- The results indicate that the values of THC, CO and CO_2 measured when the engine was equipped by Transistorized ignition system are lower than those measured when the engine was equipped with conventional ignition system.
2- During cold start and subsequent warm-up or waiting for traffic lights operations, gasoline fuel emitted at much higher excess for THC and CO first start compared

with CNG fuel. Moreover, this is attributed to wall wetting, wall quench and/or crevices as well as the increase of humidity.

3- A significant improvement can be gained when the bi-engine runs with either petrol or CNG by using transistorized ignition. Moreover, the transistorized ignition can be considered more effective than the conventional one when the engine runs with CNG fuel.
4- The object of these experiments was the study of emissions characteristics of a gasoline and bi-fuel (CNG + gasoline) SI engine. Individual engine tests have been done in steady state for CNG and gasoline fuels. All results have been measured over a wide range of engine speeds.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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