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Effects of ethanol–unleaded gasoline blends on cyclic variability and emissions in an SI engine

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Abstract

One important design goal for spark-ignited engines is to minimize cyclic variability. A small amount of cyclic variability (slow burns) can produce undesirable engine vibrations. On the other hand, a larger amount of cyclic variability (incomplete burns) leads to an increase in hydrocarbon consumption and emissions. This paper investigates the effects of using ethanol–unleaded gasoline blends on cyclic variability and emissions in a spark-ignited engine. Results of this study showed that using ethanol–unleaded gasoline blends as a fuel decreased the coefficient of variation in indicated mean effective pressure, and CO and HC emission concentrations, while increased CO_2 concentration up to 10 vol.% ethanol in fuel blend. On the other hand, after this level of blend a reverse effect was observed on the parameters aforementioned. The 10 vol.% ethanol in fuel blend gave the best results.

Keywords: Cyclic variations; SI engines; Exhaust emissions

1. Introduction

From the measurements of the pressure–time history of consecutive cycles in the combustion chamber in an SI engine, it can be easily seen that variations from one cycle to another exist.

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One of the many factors that must be considered in the design and control of spark-ignited engines is the minimization of cyclic variability. Since the pressure rate is uniquely related to the combustion, the pressure variations are caused by variations in the combustion process. In the fast combustion cycles there will be an increased tendency of knocking, which means that it is the fastest combustion that determines the upper compression ratio for chosen fuel. On the other hand in the slow combustion cycles, there is a risk of uncompleted combustion when the exhaust valve opens, which will result in higher unburned hydrocarbon (UHC) emissions and lower efficiency. The cyclic variation is specially a problem for lean burn operating engines.

The cyclic variations are caused by both chemical and physical phenomena. Of these phenomena, the variations in the residual gas fraction, the fuel–air ratio, the fuel composition and the motion of unburned gas in the combustion chamber can be taken into consideration [1]. The cyclic variations for five consecutive cycles in cylinder pressure can be seen in Fig. 1.

Cyclic variability is recognized as a limit for operating conditions with lean and highly diluted mixtures. Previous studies showed that if cyclic variability could have been eliminated, there would be a 10% increase in the power output for the same fuel consumption and power pollution of emissions from the engine [2].

One important measure of cyclic variability, derived form pressure data, is the coefficient of variation in indicated mean effective pressure (imep). It is the standard deviation in imep divided by the mean imep, and is usually expressed in percent as defined

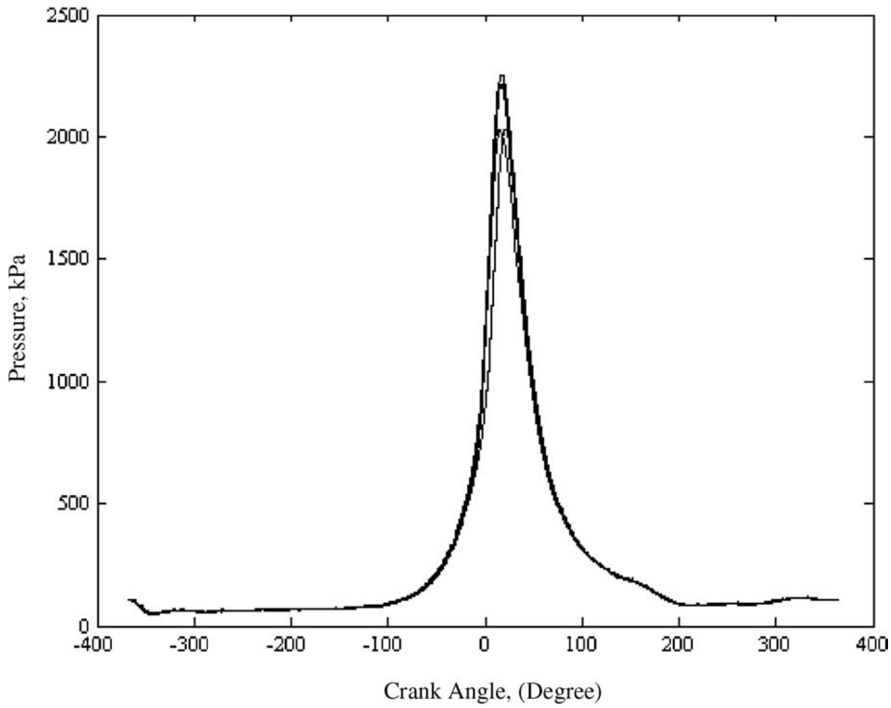

Fig. 1. An example of the cyclic variations for five consecutive cycles in cylinder pressure.

$$\mathrm{COV}_{\mathrm{imep}} = \frac{\sigma_{\mathrm{imep}}}{\mathrm{imep}} \times 100$$

This percentage defines the variability in indicated work per cycle, and it has been found that vehicle driveability problems usually result when $\mathrm{COV}_{\mathrm{imep}}$ exceeds about 10% [3]. The indicated mean effective pressure is easy to calculate and it provides a measure of the work produced for an engine cycle. imep is defined as:

$$\mathrm{imep} = \frac{W_c}{V_d}$$

where V_d is the engine displacement volume and W_c is the work per cycle, which is defined as

$$W_c = \oint P\,dV$$

The anti-knock quality of the fuel can be enhanced by the addition of lead alkyls. The lead additives enhance the antiknock quality of gasoline, but on the other hand, result in the formation and emission of toxic lead compounds. Another disadvantage of using lead additives is their damaging effect on the active materials of the catalytic devices used to control emissions [3]. For these reasons, unleaded or reduced-lead fuels are currently required in many countries around the world.

A more recent practice is to enhance the anti-knock property of the fuel by using certain high-octane oxygen-containing compounds called oxygenates. One of the commonly used oxygenates is ethyl alcohol or ethanol (C_2H_5–OH) [4].

Ethanol was the first fuel among the alcohols used to power vehicles in the 1880s and 1890s. Presently, ethanol is prospective material for use in automobiles as an alternative to petroleum based fuels. The main reason for advocating ethanol is that it can be produced from natural products or waste materials, compared with gasoline which is produced from non-renewable natural sources. In addition, ethanol shows good anti-knock characteristics. However, economic reasons still limit its usage on a large scale. At the present time, instead of pure ethanol, a blend of ethanol and gasoline is a more attractive fuel with good anti-knock characteristics [5]. The ethanol has different chemical and physical properties when compared to gasoline. These differences are expected to influence the performance and combustion of gasoline–ethanol blends.

Al-Hassan [5] studied the effect of ethanol–unleaded gasoline blends on spark ignition engine performance, and CO, CO_2 and HC emissions at three-fourth throttle opening position and variable engine speed operating conditions. The results showed that blending unleaded gasoline with ethanol increased the brake power, torque, volumetric and brake thermal efficiencies and fuel consumption, while it decreased the brake specific fuel consumption and equivalence air–fuel ratio. The CO and HC emission concentrations in the engine exhaust decreased, while the CO_2 concentration increased. The 20 vol.% ethanol in fuel blend gave the best results for all measured parameters at all engine speeds. Bata and Roan [6] experimentally investigated the effects of using ethanol with unleaded gasoline on CO, CO_2 and HC exhaust emissions. The concentration of CO was reduced by about 40–50% at an equivalence ratio on the lean side near stoichiometry. Also, the concentration of CO decreased as the percentage of ethanol increased in the fuel blend. El-Kassaby [7] studied the effect of ethanol–gasoline blends on SI engine performance. The performance tests were conducted using different percentages of ethanol–gasoline up to 40% under variable compression ratio conditions. The results showed that engine indicated power improved with the ethanol addition, the maximum improvement occurring at the 10% ethanol and 90% gasoline fuel blend. Palmer [8] used various blend rates of ethanol–gasoline fuels in engine performance tests. The results showed that 10% ethanol blend in the fuel increased the engine power output by 5%, and octane number could be increased by 5% for each 10% ethanol added. Hsiech et al. [9] investigated the effects of various blends rates of ethanol–gasoline fuels on engine performance and pollution emission. The results showed that blending unleaded gasoline with ethanol slightly increased the torque output and fuel consumption; CO and HC emissions decreased dramatically as a result of leaning effect caused by ethanol addition; and CO_2 emission increased because of the improved combustion. Finally, it was noted that NO_x emission depends on the engine operating condition rather than ethanol content.

This paper aims to help in understanding the effects of blending ethanol with unleaded gasoline on cyclic variations and the emissions of typical automotive engines.

2. Experimental apparatus and procedure

The engine used in the present study is the FIAT, 1.801 dm^3, carburetted and four-stroke spark ignition engine. The specifications of the engine are listed in Table 1.

The engine is fully equipped for measurements of all operating parameters. Pressure time history was measured by a piezo-electric pressure transducer (KISTLER, 6117BFD17 type), and crankshaft degree angle sensor was connected to the relevant amplifiers.

A data acquisition system was used to collect the relevant data and store the data in a personal computer for offline analysis for which a computer program in Q-BASIC language was written to collect the data.

The pressure signal fed into a charge amplifier and the crank angle signal collected by a degree maker shape channel were fed into a acquisition card attached to the personal computer. The acquisition card could collect data at the rate of 100 kHz.

The experiments have been carried out after running the engine until it reached steady state, where the oil temperature was at 50 °C ± 5, and cooling water temperature was at 70 °C ± 5. The engine was operated at 2000 rpm and loaded by hydraulic dynamometer. Water inlet pressure of hydraulic dynamometer was held constant at 0.5 bar. The variations in the engine speed caused

Table 1
Technical specifications of the engine

Engine	FIAT
Number of cylinders	4
Bore (mm) × Stroke (mm)	86.4 × 67.4
Cycle	4-stroke
Compression ratio	9.2:1
Displacement volume (dm^3)	1.581
Maximum power	62 kW at 5800 rpm
Maximum torque	13 da Nm at 2900 rpm
Cooling system	Water-cooled

by different ethanol–unleaded gasoline blends were regulated by changing the throttle valve opening position. The spark timing was adjusted to yield minimum spark advance for best torque timing (MBT) at each ratio of ethanol to unleaded gasoline.

The unleaded gasoline was blended with ethanol to get five test blends ranging from 0% to 20% ethanol with an increment of 5%. The fuel blends were prepared just before starting the experiment to ensure that the fuel mixture was homogeneous and to prevent the reaction of ethanol with water vapor. Before running the engine with a new fuel blend, it was allowed to run for a sufficient time to consume the remaining fuel from the previous experiment.

3. Results and discussion

Figs. 2–6 show the indicated mean effective pressure normalized with the average of the actual pressure for 50 consecutive cycles from 0% to 20% ethanol–unleaded gasoline ratios with a 5% step increment, and Fig. 7 shows the coefficient of variation of indicated mean effective pressure at the same ratios. COV_{imep} was observed as 3.077, 2.970, 2.352, 3.085 and 3.317 for pure gasoline and 5%, 10%, 15% and 20% ethanol in fuel blend experiments, respectively.

The percentage of ethanol in fuel blend affects the volatility and the latent heat of fuel blend. The latent heat of ethanol ($840\,kJ\,kg^{-1}$) is higher than that of gasoline ($305\,kJ\,kg^{-1}$) which makes the temperature of intake manifold lower, and increases the volumetric efficiency. Moreover, ethanol decreases the stoichiometric air–fuel ratio of the fuel blends, because the stoichiometric relative air–fuel ratio of ethanol fuel (8.96) is lower than that of the unleaded gasoline fuel (15.13),

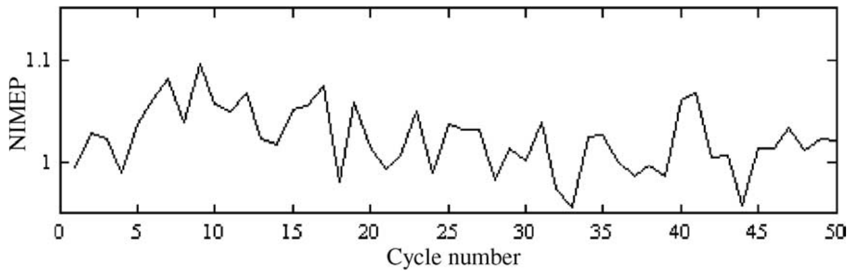

Fig. 2. Normalized IMEP during 50 consecutive cycles (0% ethanol).

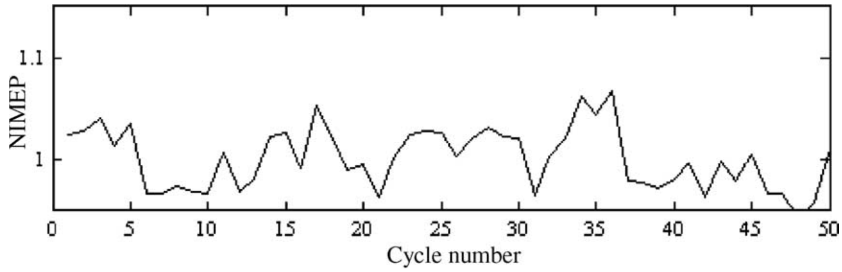

Fig. 3. Normalized IMEP during 50 consecutive cycles (5% ethanol).

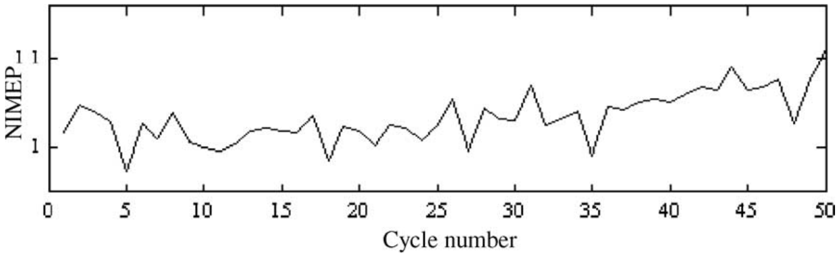

Fig. 4. Normalized IMEP during 50 consecutive cycles (10% ethanol).

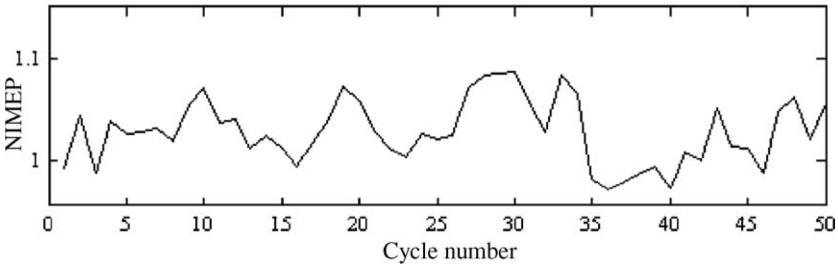

Fig. 5. Normalized IMEP during 50 consecutive cycles (15% ethanol).

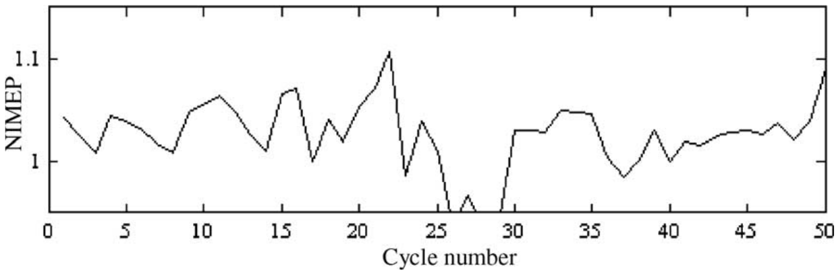

Fig. 6. Normalized IMEP during 50 consecutive cycles (20% ethanol).

and increases the actual air–fuel ratio of the blends as a result of the oxygen content in ethanol. However, these effects continued until the percentage of ethanol reaches 10 vol.%. The attitude of relative air–fuel ratio agreed well with the early studies [8,9], which were about the effects of alcohols on engine performance and emissions. It can be seen from Figs. 7,8 that the relative air–fuel ratio was the highest at the 10 vol.% ethanol ratio and COV_{imep} was the lowest. After the 10 vol.% ethanol in blend, the relative air–fuel ratio started to decrease and COV_{imep} started to increase in the experiments due to the increase in the temperature of the intake manifold and decrease in the volumetric efficiency.

Figs. 9–11 show the effect of the ethanol ratio in fuel blend on the HC, CO and CO_2 emissions, respectively. It can be seen from these figures that as the ratio of the ethanol to ethanol–unleaded gasoline blend increased up to 10%, the HC and CO emissions decreased and CO_2 emissions increased. When the ethanol ratio exceeded by about 10% level, HC and CO emissions increased

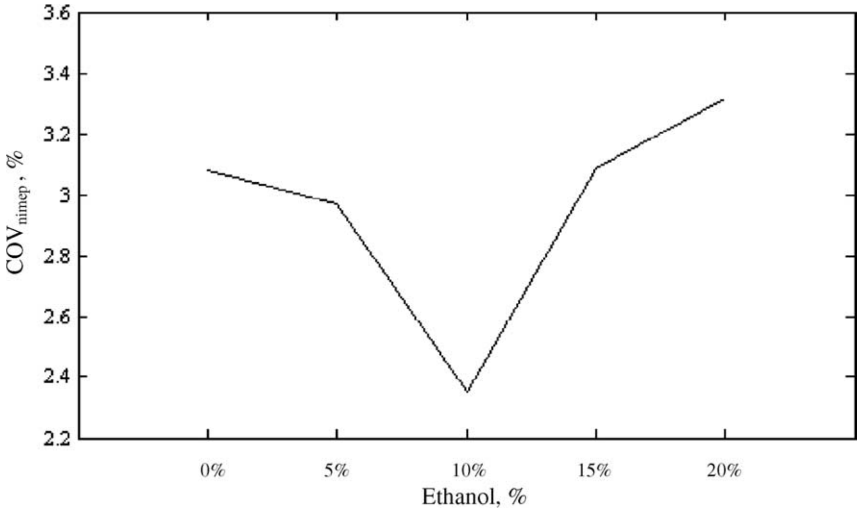

Fig. 7. Coefficient of variation of indicated mean pressure at different ethanol mixtures.

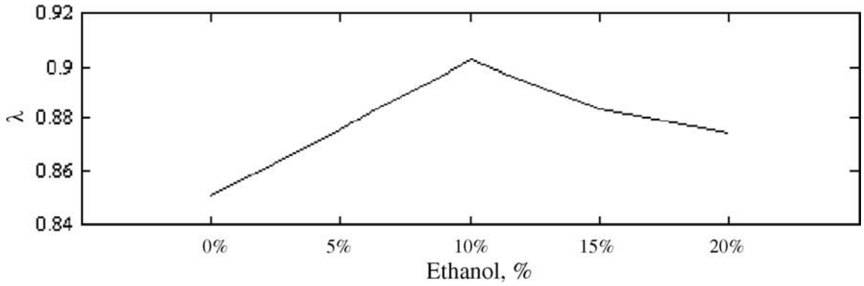

Fig. 8. The effect of ethanol addition on relative air–fuel ratio.

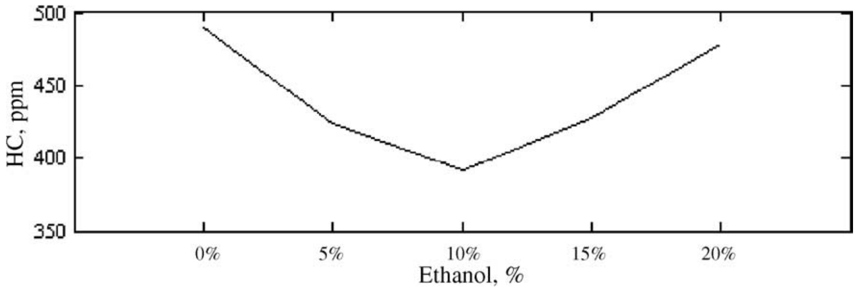

Fig. 9. The effect of ethanol addition on HC emission.

and CO_2 emissions decreased. COV_{imep} was higher, however, CO and HC emissions were lower at pure gasoline experiments than those of 15% and 20% ethanol–unleaded gasoline blends and this behavior agrees with the relative air–fuel ratio and COV_{imep} shown in Figs. 7 and 8.

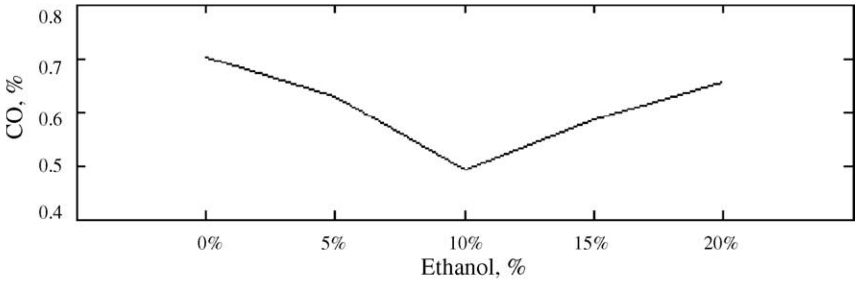

Fig. 10. The effect of ethanol addition on CO emission.

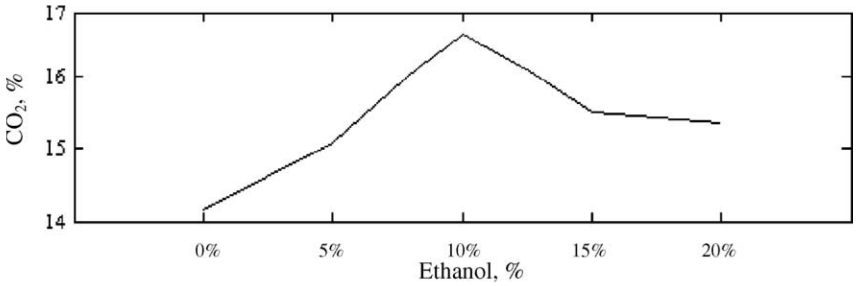

Fig. 11. The effect of ethanol addition on CO_2 emission.

4. Conclusions

1. The 10 vol.% ethanol–unleaded gasoline blend presented the best results for the COV_{imep} and exhaust emissions.
2. COV_{imep} was observed as 3.077 and 2.352 for pure gasoline and 10 vol.% ethanol blend experiments, respectively.
3. Although the COV_{imep} was especially a problem for lean burn condition and the relative air–fuel ratio reached the highest value at the 10 vol.% ethanol ratio, COV_{imep} reached to the minimum value.
4. Use of ethanol–unleaded gasoline blend as a fuel leads to a significant reduction in exhaust emissions by about 20.2% and 30.01% for HC and CO emissions, respectively, at 10 vol.% ethanol ratio compared to pure gasoline experiments.
5. After the ratio of ethanol to unleaded gasoline–ethanol blend exceeded the 10 vol.%, COV_{imep},HC and CO emissions showed an increasing tendency due to the increase in the temperature of the intake manifold and the decrease in the volumetric efficiency. However, no problem existed when the engine was operated at the 20 vol.% ethanol ratio.

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