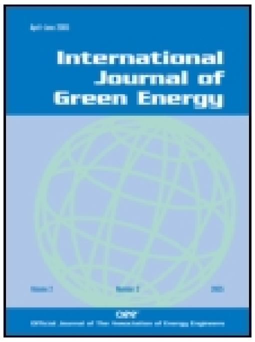

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A Comparison of Engine Performance and the Emission of Fusel Oil and Gasoline Mixtures at Different Ignition Timings

ALPER CALAM[1], YAKUP İÇINGÜR[1], HAMIT SOLMAZ[1], and HASAN YAMIK[2]

[1]*Technology Faculty, Gazi University, Ankara, Turkey*
[2]*Engineering Faculty, Department of Mechanical and Manufacturing Engineering, Bilecik S.E. University, Bilecik, Turkey*

Alcohols have been used as a fuel for engines since 19th century. Among the various alcohols, ethanol is known as the most suited renewable, bio-based and ecofriendly fuel for spark-ignition (SI) engines. In addition, ethanol has higher evaporation heat, octane number, and flammability temperature therefore it has positive influence on engine performance and reduces exhaust emissions. In this study, engine performance and emission levels of unleaded gasoline and unleaded gasoline-fusel oil blends in a spark ignition engine, under variable ignition timings are investigated. Engine torque increased and brake specific fuel consumption (bsfc) decreased with the ignition timings. For F0 and F10 blends, hydrocarbon emissions changed by 22% on average and carbon monoxide (CO) emissions changed by 9.2%. It was also observed that nitrogen oxide (NOx) emissions were reduced.

Keywords: Gasoline, fusel oil, performance, emission

Introduction

The majority of world energy resources are of fossil fuel form and the adverse effects of such fossil fuels on the environment and on human beings in particular, are increasing. In addition, the combustion of fossil fuels contributes to an increase in greenhouse gases in the atmosphere. Furthermore, the availability of fossil fuels in the immediate future is in question. Considering all these factors, the popularity of research into possible alternative fuels is increasing. The main properties of alternative fuels are that they are easy to store, have high energy density, and burn easily with low emissions. There are certain types of fuels which are not originally fossil fuels, but which have better combustion than fossil fuels. These are synthetic liquid fuels, alcohols, and gas fuels (Thring 1983). The alcohols are hydrocarbon oxide. Ethanol has a higher octane than gasoline and can be produced from agricultural products. As they contain oxygen, they also give off better emissions (Can, Çelikten, Usta 2004). Ethanol and methanol mixtures can be used in combustion engines with minor modifications to the engines, and the use of alcohols as alternative fuels also reduces dependency on fossil fuels. Considering the development of internal combustion engines, the main aim is to improve fuel economy and reduce CO_2 emissions with regard to legislation as well as market trends.

Alcohol-gasoline mixtures can be used with or without water. Depending on the ambient temperature, the chemical structures of the components and purity of the chemical components, phase differentiation problems can be observed (Karaosmanoğlu, Işığıgür, Aksoy 2008). As a result of phase differentiation, differences between each cycle and difficulties in combustion and initial movement problems can be observed (Karaosmanoğlu, Işığıgür, Aksoy 1993). In addition, as alcohols contain water, these cause corrosion and wear problems. To avoid phase differentiation problems, experiments are conducted with methanol gasoline mixtures containing 15% methanol and certain chemical additives. As a result of these experiments, it is observed that engine performance improves (Kowalewicz 1993). Similar experiments are also conducted with methanol gasoline mixtures, which contain 20% methanol and, to avoid phase differentiation, fusel oil is added to the mixture. Similar improvements are observed in engine performance as in the previous experiment. In addition to engine performance, emission levels also improved (Karaosmanoğlu et al. 1991). The performance and emissions of gasoline-ethanol mixtures which contain 5%, 10%, 20%, and 30% ethanol are also investigated with a variety of excess air percentages. Engine torque increases by 4% when a 30% gasoline–ethanol mixture is used. As the ethanol ratio increases, improvements in emission values are measured (Wu et al. 2003). As the alcohol ratio increases in the mixtures, a higher engine knocking strength is observed compared with gasoline (Gautam and Martin 2000). Even with an engine compression value increase, no engine knocking is observed when the ethanol ratio is increased in the mixture (Topgül et al. 2006). When ethanol–heptane mixtures are used in a single cylinder engine, with 10%, 20%, 30%, 40%, 50% volumetric ratios, the effective pressure increases from 3.1 bar to 5.1 bar. In addition to this, thermal efficiency increases by 50% under a full load cycle, but decreases in partial load cycles (Xingcai et al. 2006). Ethanol has great

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Address correspondence to Hasan Yamık, Engineering Faculty, Department of Mechanical and Manufacturing Engineering, Bilecik Seyh Edebali University, Bilecik, Turkey. E-mail: hyamik@gmail.com, hasan.yamik@bilecik.edu.tr

heat evaporation. As a result of this, it has a coolant effect on the mixtures, so volumetric efficiency increases (Hsiehi et al. 2002; Al-Baghdadi 2003). In addition, a limited increase is observed in engine torque and emissions. When oxygen in alcohol is burned, it is observed that CO and HC emissions significantly decrease (Gautam and Martin 2000; Hsiehi et al. 2002; Wu et al. 2003; Topgül et al. 2006, Xingcai et al. 2006). (Yao et al. 2007). A new approach to burning methanol in engine is proposed and investigated, in which the engine burns DME and methanol dual fuels in HCCI mode, and DME is converted from methanol. HCCI operation can be obtained over a fairly wide speed and load range, and ultra-low NO_x emissions and high indicated thermal efficiency can be achieved for burning methanol (Metwalley et al. 2012). Two different fuel injection systems are used. The results indicate that most of the carbon monoxide (CO), carbon dioxide (CO_2), and unburned total hydrocarbons (THC) emission appears at higher load as well as near the idling speed (Wang, Zhang, Wang 2007). Diesel engines have higher NO_x and particular matter emissions compared with gasoline engines. Stringent emission regulations on internal combustion engines have been implemented by governments all over the world, especially in USA, Europe, and Japan (Love, Parthasarathy, Gollahalli 2009). Study employed a recently developed experimental technique to investigate the effect of iodine number on NO_x formation in laminar partially premixed flames of three vaporized biofuels: canola methyl ester, soy methyl ester, and methyl stearate. Observed that the peak NO_x concentration significantly increased with the iodine number, indicating a strong correlation between the chemical structure of the fuel and NO_x emission.

Fusel oil is a by-product of ethyl alcohol production with fermentation during the distillation process and is a natural source of amyl alcohols. The composition and amount of the fusel oil depends on the type of carbon used in the alcohol production, fermentation process, preparation method, and decomposition method of the fusel oil in the mixture. Fusel oil consists of low molecular mass alcohols (mainly i-amyl alcohol, butyl alcohol, n-propyl alcohol, n-butyl alcohol, ethyl alcohol, and n-amyl alcohol), water, aldehyde, unsaturated acids and their esters, poly alcohols and terpens (Patil, Koolwal, Butala 2002). The properties of the fusel oil composite are shown in Table 1.

The ignition processes affect the overall performance in the spark ignition engines. It affects the gas temperature inside the cylinder and the exhaust system and on the level of NO_x emission.

In this study, the performance (brake torque, BSFC, effective efficiency, exhaust gas temperature), emissions (CO, NOx, THC) characteristics of an engine were investigated in engines with the same type of injection system and fuelled with gasoline (F0), fusel oil, and their F5, F10, F20, F30, F50 blends. Spark timing values are changed.

Material and Method

Test Equipment

Experiments are conducted with a Hydra brand, four cylinders spark ignition engine. A McClure DC dynamometer and Sun MGA 1500 emission analyzer were installed into the set up. The experiment set up is shown in Figure 1 and the technical properties of the engine are shown in Table 2.

The experiments were conducted at a compression rate of 11/1. The Sun MGA1500 make emission analyzer was capable of measuring NO_x, CO, CO_2, O_2, and HC emissions. Parameters, measurement range, and sensitivity are shown in Table 3.

Table 1. Physical Properties of Fusel Oil Components

Component Test Method	Chemical Formula	Molecular Mass (g/mol)	Density (g/cm^3) ASTM D1007	Boiling Point (°C) ASTM D1007	Melting Point (°C) ASTM D1007	% Volumetric	% Mass
i–amyl alcohol	$C_5H_{12}O$	88.148	0.8104	131.1	−117.2	63.93	61.52
i–butyl alcohol	$C_4H_{10}O$	74.122	0.802	108	−108	16.66	15.87
n–butyl alcoholl	$C_4H_{10}O$	74.122	0.8098	117.73	−89.5	0.736	0.708
n–propyl alcohol	C_3H_8O	60.09	0.8034	97.1	−126.5	0.738	0.704
Ethanol	C_2H_6O	46.07	0.789	78.4	−114.3	9.58	8.98
Water	H_2O	18	1	100	0	10.3	12.23

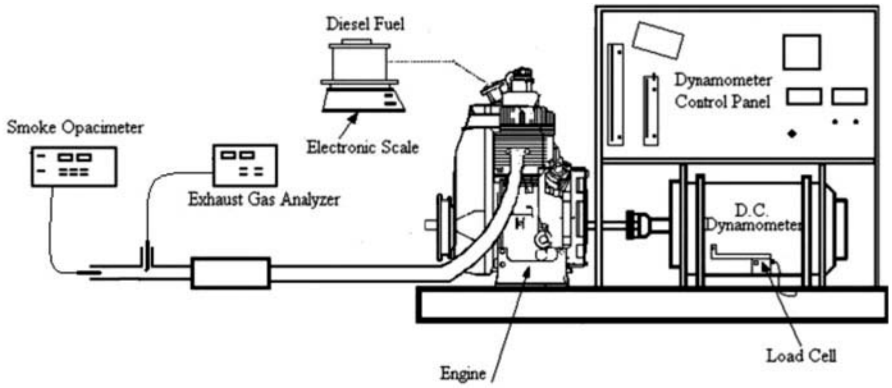

Fig. 1. Schematic layout of the test bench.

Table 2. Technical Properties of the Engine

Engine Type	Hydra, Spark Ignition Engine
Number of Cylinders	1
Bore × Stroke	80.26 × 88.9 (mm)
Max. Speed	5400 (1/min)
Max. Output	15 (kW)
Compression Ratio	5/1–13/1
Fuel System	Fuel Injection
Spark Timing Range	70° BTDC–20° ATDC

Table 3. Emission Measurement Range and Sensitivity

Parameter	Measurement Range	Accuracy
CO	0–14%	0.001%
HC	0–9999 ppm	1 ppm
NO_x	0–5000 ppm	1 ppm
CO_2	0–18%	0.1%
0_2	0–25%	0.01%
λ	0–4	0.001

The engine was warmed to an optimum working condition using experimental fuel. The experiment was conducted under full load conditions of 3500 1/min maximum engine torque with varying advance values. Engine performance and emission values were measured with unleaded gasoline F0 and blends. The impact of varying advance values on engine torque, specific fuel consumption, and emissions (CO, HC and NO_x) were measured. For each measurement at different spark timing values, measurement of the experimental data was completed in a short time period just after the engine torque and emission levels reached a stable value. All of the experiments were conducted with a stoichiometric air/fuel ratio.

Test Fuels

Unleaded gasoline (F0) and unleaded gasoline fusel oil blends were used in the experiments. These blends are shown in Table 4. The fuels used in the experiments were kept at room temperature for 24 hours and no phase change was observed. The fuel properties are shown in Table 5.

Results and Discussion

Figure 2 shows the engine torque change with respect to the spark timing at 3500 rpm. A maximum engine torque was measured at 26° ignition timing at 33.09 Nm for F0 fuel. A maximum engine torque was measured at 24° ignition timing at 33.53 Nm for F10. It was observed that when the spark timing decreased, the differences between the blend torques increased. As alcohols have a high enthalpy of vaporization, the temperature of the air inlet into the engine reduces. As a result of this, the tendency to self-ignition and engine torque reduced.

Table 4. Composition of Fuel Blends

	Composition
F0	Unleaded Gasoline
F5	5% Fusel Oil+95% Unleaded Gasoline
F10	10% Fusel Oil+90% Unleaded Gasoline
F20	20% Fusel Oil+80% Unleaded Gasoline
F30	30% Fusel Oil+70% Unleaded Gasoline
F50	50% Fusel Oil+50% Unleaded Gasoline

Figure 3 shows the specific fuel consumption with respect to ignition timing at 3500 rpm. The variances in minimum specific fuel consumption are measured at 1.85% at 24° ignition timing for F0 and F10. However, the variance in 24° ignition timing for F0-F50 is measured at 18.89%. Retarding the ignition timing, there was insufficient time for ignition. As a result, as the spark timing advance reduced, specific fuel consumption increased. Fusel oil has a low heat value and this result in increased specific fuel consumption.

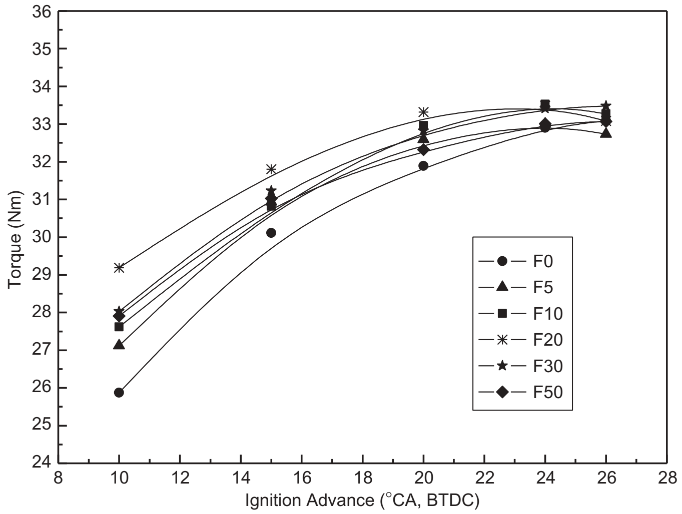

Fig. 2. Variation of engine torque with ignition timing.

Table 5. Fuel Properties

	Test Standard	Unleaded gasoline	Fusel Oil	F5	F10	F20	F30	F50
Density (kg/m^3)	ASTM D 4052	745	849	748	754	759	766	784
Heat Value (kJ/kg)	ASTM D 240	43580	29514	43175	42681	42124	41794	39581
Engine Octane Number	ASTM D 2700	86.51	103.61	87	87.08	87.09	87.1	89.30
Octane Number	ASTM D 2699	96.33	106.82	97.40	97.8	97.84	98.3	98.38
CFPP (°C)	ASTM D 6749	−53	>50	>50	>50	>50	>50	>50

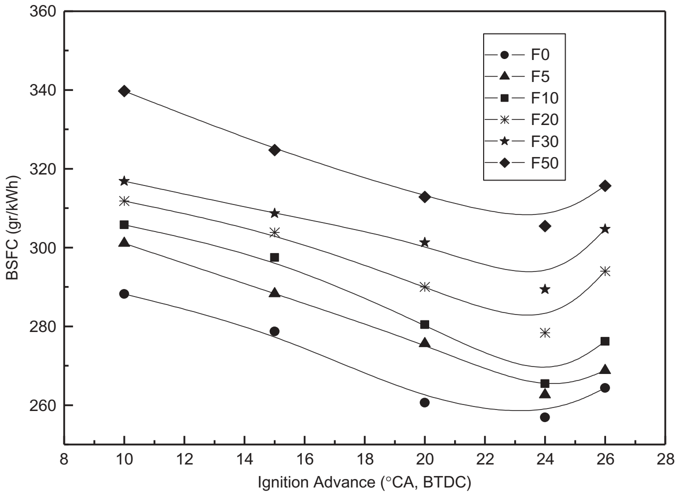

Fig. 3. Variation of specific fuel consumption with ignition timing.

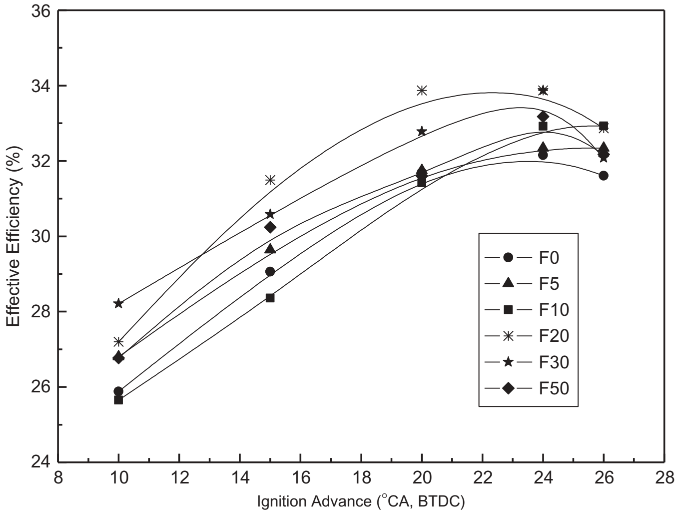

Fig. 4. Variation of effective efficiency with ignition timing.

Effective efficiency is defined as work done by an engine shaft divided by the total energy input. Figure 4 shows effective efficiency with respect to ignition timing. When effective thermal efficiency is high, both specific fuel consumption and engine torque values are low. For both fuels, maximum effective efficiency was measured at 24° ignition timing. The variance in effective efficiency between the F0 and the F10 fuels is in a range between 4.18% and 8.66% depending on the ignition timing. As the fusel oil blend range increases, engine effective efficiency increases. Although fusel oil has a low heat value, it has a high effective efficiency. It is observed that as the spark timing value reduced, the effective efficiency similarly reduced. As the spark retarded for efficient combustion became insufficient, this resulted in reduced effective efficiency. This has an adverse impact on combustion performance.

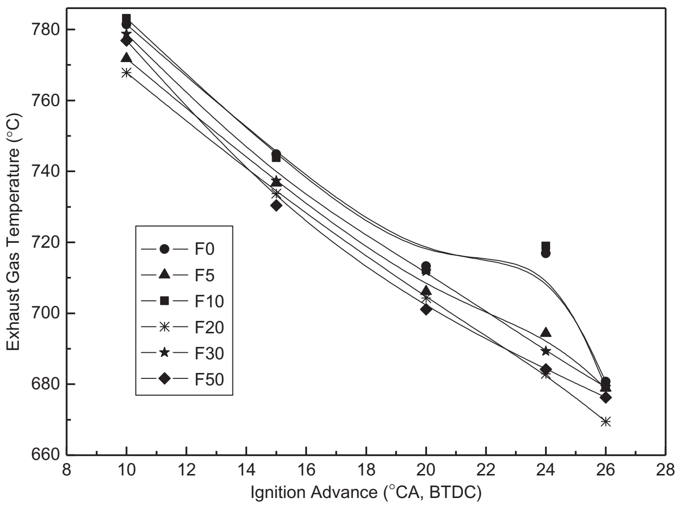

Fig. 5. Variation of exhaust gas temperature with ignition timing.

As the ignition advance increases, the exhaust gas temperature decreases. Figure 5 shows the exhaust gas temperature with respect to ignition timing. The highest exhaust gas temperature difference in F0–F50 was measured as 2.20%. The variance in exhaust gas temperature is less than 1% on average for both fuels. With the early ignition, maximum pressure reached a closer zone to TDC (Top Dead Centre). The piston approaches BDC (Bottom Dead Centre), the gas temperature in the cylinder decreases. As ignition advance decreases, maximum pressure achieved after TDC, this would reduce the efficiency of the process. Therefore retarding the ignition timing is always associated with incomplete combustion and an increase in the exhaust temperature. As a result of retarding the ignition timing there will be a reduction in NO_x formation during the combustion process which is mainly due to drop in the peak temperature.

The reason for HC emission in the exhaust gas is either low temperature during the combustion or inadequate oxygen in the combustion. Also, a fuel rich or lean mixture affects the HC emission. Figure 6 shows the HC emissions with respect to ignition timing for F0 and F10 fuels. The variance in HC emissions between F0 and F10 fuels, at a maximum engine torque, is approximately 11%. Also, the variance in HC emissions

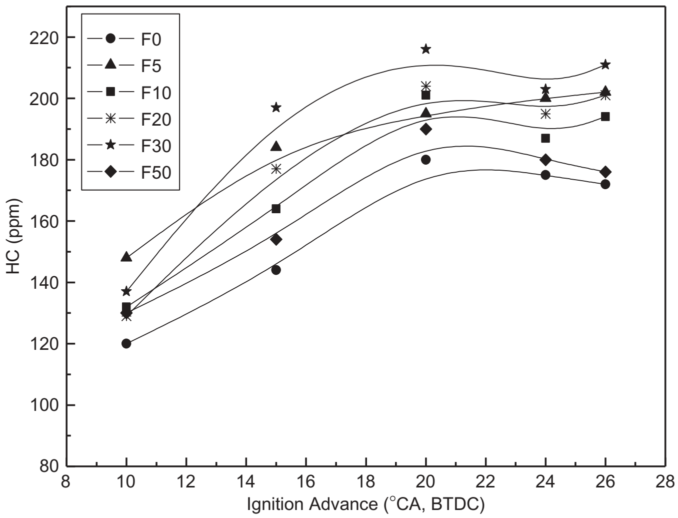

Fig. 6. Variation of HC emissions with ignition timing.

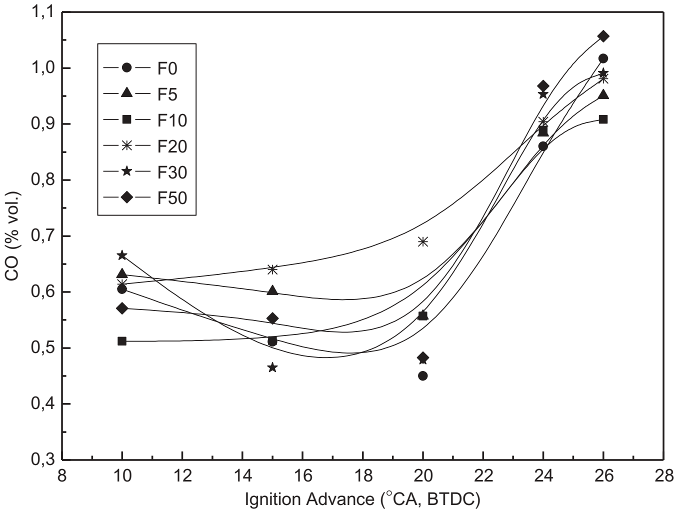

Fig. 7. Variation of CO emissions with ignition timing.

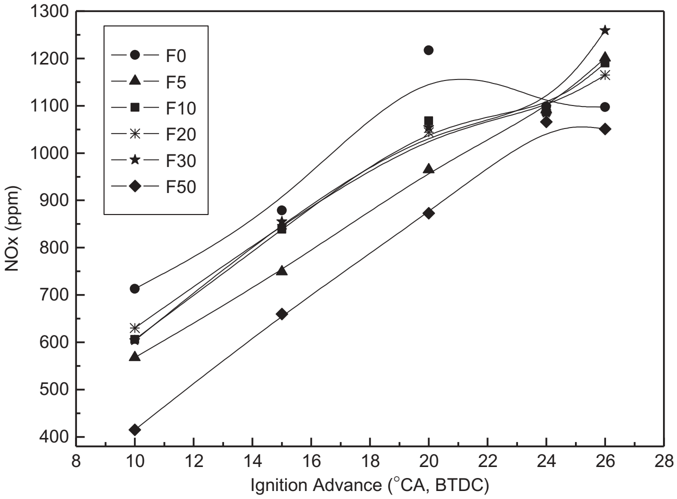

Fig. 8. Variation of NO*x* emissions with ignition timing.

between F0 and F50 fuels is approximately 38%. As ignition timing increases, the combustion cylinder temperature and pressure increases. In addition, as exhaust temperature decreases, HC emissions increase since oxidation and the cylinder surface temperature decrease during the exhaust process.

Figure 7 shows CO emissions with respect to ignition timing for F0 and F10 fuels. The lowest CO emission was measured at 20° spark timing for the F0 fuel, and at 10° for the F10 fuel. The variance of CO emissions between the F0 and F10 fuels at 20° ignition timing was measured as 20.1%. Due to insufficient oxygen and a nonhomogenous mixture in the combustion cylinder, CO formation developed during the combustion process. When engine spark timing is retarded, the CO emissions increase.

NO_x emissions increase with an increased ignition advance. Due to advanced ignition during combustion, the maximum pressure increases. As a result of ignition and high pressure during combustion, a higher temperature was observed after combustion in the cylinder. Due to the impact of advance, it took longer to evacuate the high temperature gas emissions from cylinder. As a result of this, the NO_x amount increased. Figure 8 shows NO_x emissions with respect to ignition advance for both fuels used in the experiment. NO_x emissions vary between 1.1% and 30% depending on the ignition advance. A decrease in ignition advance, causing a reduction of NO_x emissions, on the other hand, also reduces engine torque and increases specific fuel consumption. Advancing the ignition timing for a lean mixture causes the combustion process to occur earlier in the cycle, which results in higher peak cylinder pressure and burned gas temperatures, hence higher rates of NO_x formation.

Conclusions

Fusel oil is a by-product obtained during alcohol production and is not widely-used in industry. However, both the physical and chemical properties of fusel oil indicate that it can be used as an alternative fuel for spark ignition engines. The variance in engine torque under 3500 1/min. and between 10–24° CA for the F0–F10 and F0–F50 fuels are measured as 1.91% and 1.11%, respectively. Similarly, a variance in average effective efficiency between 20°–26° ignition advance is calculated as between 0.8% and 4%. The highest effective efficiency is calculated in the F50 fuel. HC emissions vary between 4% and 40% with respect to ignition advance. The variance in minimum CO emissions under different advance values for the F0–F50 fuels is calculated as 9.2%. High flame temperatures cause NO_x emissions during the combustion process. As ignition advance increases, ignition starts at an earlier phase of the cycle and increases the combustion temperature. The variance in NO_x emissions is around 9% on average for the fuels used in the experiments.

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