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Experimental study of lean spark ignition combustion using gasoline, ethanol, natural gas, and syngas

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A B S T R A C T

In the development of internal combustion engines, engineers and researchers are facing the challenge of improving engine efficiency while reducing harmful exhaust emissions. Previous research has shown that lean combustion is one of the viable techniques that can improve engine efficiency while effectively reducing exhaust emissions. Lean burn engines operate at low burned gas temperatures and can achieve high thermal efficiency based on favorable mixture thermodynamic properties. However, under high dilution levels, a lean misfire limit is reached where the combustion process becomes unstable and incomplete combustion starts to occur. Instability significantly affects engine efficiency, driveability, and exhaust emissions, which limit the full potential of lean burn engines. The lean misfire limit is not only dependent on engine design but also on fuel properties. Therefore, fuels that are conducive to lean combustion can provide the opportunity for enhanced efficiency and reduced emissions. Spark ignited (SI) combustion with conventional gasoline has shown to have relatively narrow range of fuel-air equivalence ratio; therefore, it is desired to explore the lean limit of SI combustion by using alternative fuels, which can also contribute to the reduction of greenhouse gas emissions from transportation and power generation.

Experiments were conducted on a Cooperative Fuel Research (CFR) engine with varying fuel-air equivalence ratio (φ) to assess the engine performance and emissions with three alternative fuels, natural gas, ethanol, and syngas, at compression ratio of 8:1 and engine speed of 1200 rev/min. Equivalence ratio was varied by decreasing the mass of fuel while keeping the mass of air the same. The lean misfire limit was defined as the equivalence ratio where the CoV of IMEP across multiple consecutive engine cycles was greater than 5%. It was found that syngas can maintain stable combustion at extremely lean conditions and has the lowest lean misfire limit. Natural gas combustion achieved a lower lean misfire limit than gasoline and ethanol. Gasoline and ethanol had similar lean misfire limits, but it was found that gasoline helped the engine to achieve higher load and fuel conversion efficiency compared to the three alternative fuels.

1. Introduction

Spark ignited engines are the most common power source for light-duty vehicles and some small power equipment. With increasing oil prices, depleting fossil fuel reserves, and growing concerns of environmental effects from burning fossil fuels, alternative fuels can offer solutions for sustainable future transportation and power generation. Although SI engines have been widely used in commercial applications, their thermal efficiency has been limited in part by low compression ratio and stoichiometric mixture composition. Lean operation can offer significant thermal efficiency benefits due to high ratio of specific heats (γ) and low heat transfer losses, while low burned gas temperatures reduce NOx emissions. Although lean burn SI engines can benefit from improved engine efficiency and reduced emissions, certain challenges need to be overcome before the full potential of lean burn SI engines can be realized. As the fuel-air mixture becomes lean, the heat release is reduced and the laminar flame speed decreases, resulting in increased burn duration, which in turn may offset any thermal efficiency benefits [1,2]. Also, at very lean conditions combustion becomes unstable and the ignitability of the fuel-air mixture is poor, leading to incomplete combustion or misfire, which greatly affects emissions and thermal efficiency [3,4]. This lean limit can be defined as the equivalence ratio at which the Coefficient of Variation (CoV) of Indicated Mean Effective Pressure (IMEP) is above 5%. The lean misfire limit is not only dependent on engine design, but also depends on fuel-air mixture properties such as the laminar flame speed, ignition energy requirement, and the latent heat of vaporization. Therefore, a fuel which enables very lean combustion can provide the opportunity for enhanced efficiency and reduced emissions. Gasoline SI combustion has been shown to have a relatively narrow lean operating range [1], but alternative fuels such as ethanol, natural gas, and synthesis gas (syngas) can extend the lean limit of SI engines based on their favorable properties. In order to investigate the extension of the lean misfire limit using alternative fuels, it is important to understand their combustion characteristics, and how their different fuel properties can promote lean mixture ignition and flame development.

Several studies have focused on SI engine operation with alternative fuels and have compared them with conventional gasoline. *Natural gas* is considered the most common and promising alternative gaseous fuel and is primarily composed of methane at concentration that ranges

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between 85 and 97% by volume [5]. It has high H/C ratio (~3.8), which results in lower CO_2 emissions [6,7], and a high research octane number (RON~130), which enables operation at high compression ratio for increasing thermal efficiency and power output [8]. It also has a lower lean flammability limit than gasoline, which promotes lean ignition in SI engines [2]. The lower heating value (LHV) of natural gas is slightly higher than gasoline, however the gaseous nature of the fuel can result in lower engine volumetric efficiency and thus power output [5], caused by the gas displacing intake air and the lack of evaporative cooling compared to liquid fuels. In addition, SI combustion with natural gas has shown lower heat release rate than gasoline combustion, because of the lower laminar flame speed of natural gas under high temperature and pressure conditions, which increases the burn duration and thus heat transfer loss, resulting in lower thermal efficiency [9].

Synthesis gas or "syngas", is a mixture that consists mainly of hydrogen and carbon monoxide and can be formed at different ratios. Syngas can be produced from natural gas, coal, biomass, or hydrocarbon feedstock. Typical LHV values of syngas are lower than natural gas and hydrocarbon fuels, due to the lower density of hydrogen and carbon monoxide [10,11]. Hydrogen has the highest laminar flame speed and lowest ignition energy among gaseous fuels, and also has a high Research Octane Number (RON) (~120) [12–14]. Since syngas can be composed mainly of hydrogen, running syngas in SI engines is expected to lower the lean misfire limit, which reduces the duration of flame development and flame propagation, and thus improve the engine lean burn capability compared to conventional gasoline. However, syngas also affects the engine volumetric efficiency due to its gaseous state, and typically has a lower heating value compared to liquid fuels. Emissions formation can benefit from using syngas since there is no hydrocarbon content in the fuel. In addition, by enabling lean combustion the burned gas temperatures can be kept below the NOx formation threshold.

Ethanol has traditionally been used as a sole fuel or in blends with gasoline for use in SI engines. Since ethanol can be produced from sugar cane, corn, or wheat it has been used extensively as an alternative fuel or additive to gasoline in the United States, Brazil, and Europe. Ethanol has higher RON than gasoline (~113), which enables the use of higher compression ratio with associated efficiency and performance benefits [15]. Engine volumetric efficiency can be improved due to the higher latent heat of vaporization of ethanol compared to gasoline, which is particularly effective in suppressing end gas knock in direct injection engines [16]. Because ethanol is an oxygenated fuel and has higher laminar flame speed than gasoline, the combustion efficiency and engine-out emissions of total hydrocarbons (THC) and carbon monoxide (CO) can be lower than gasoline. Previous studies have shown that the faster heat release rate and lower flame temperature of ethanol can result in higher engine thermal efficiency due to lower heat loss and higher γ, and also lower NOx emissions due to lower peak combustion temperature compare to gasoline [16–18]. However, ethanol has lower LHV compared to gasoline, which results in higher fuel consumption for producing the same amount of work.

The objective of this study was to conduct experimental testing in order to analyze the effects of fuel properties on the lean misfire limit of SI combustion and assess their effects on engine performance and emissions. The following sections describe the experimental setup and methodology, as well as the results obtained from experimental testing.

2. Experimental setup and methodology

Experiments were performed on a single-cylinder, four-stroke, spark ignited Cooperative Fuel Research (CFR) engine, which has variable compression ratio (6:1–18:1). The engine specifications are shown in Table 1 and the schematic diagram of the experimental setup is shown in Fig. 1. The engine was coupled to an active DC dynamometer for measuring engine speed and torque. The intake air flow rate was measured and controlled with an Alicat MCRW-500SLPM-D/5M mass flow meter mounted upstream of the intake plenum. The liquid fuels were injected into the intake port, but the gaseous fuels were stored in compressed gas tanks and fumigated to the intake plenum to mix with the intake air. A second Alicat MCR-100SLPM-D/5M mass flow meter was used to measure and control the mass flow rate of the gaseous fuels. In-cylinder pressure data was measured using a Kistler 7061B piezoelectric pressure transducer on the cylinder head. A BEI XH25D-SS-1024 crank angle encoder was mounted on the engine shaft for measuring crankshaft position with a resolution of 0.2 crank angle degrees (CAD). For each operating point, engine data for 200 cycles were recorded for and post-processed with in-house, high fidelity, heat release analysis routines. Emissions were measured with a Horiba MEXA-7100DEGR motor exhaust gas analyzer.

Table 1
CFR engine specifications.

Bore (mm)	82.6
Stroke (mm)	114.3
Connecting Rod Length (mm)	254
Displaced Volume (cm^3)	611.7
Clearance Volume (cm^3)	87.4
Compression Ratio	8:1
Fueling Method	Port fuel injection
Engine Speed (rev/min)	1200

The engine was operated at 1200 rev/min, with compression ratio fixed at 8:1 and intake pressure of 75 kPa. Before collecting any data, the engine was conditioned to ensure thermal equilibrium and steady state operation. All operating points shown in this study were collected with spark timing set for Maximum Brake Torque (MBT) at each point. Table 2 below shows the spark timing for each fuel at specific fuel-air equivalence ratio values. For each set of experiments conducted with the four different fuels, the engine was started with the fuel-air ratio set at or close to stoichiometry and then it was progressively decreased until the lean misfire limit was met (COV IMEP > 5%). E10-gasoline was used as the baseline fuel and three different alternative fuels were used during the experiment: (i) 190 proof ethanol (95% ethanol, 5% water vol.), (ii) compressed natural gas (95% CH_4 vol.), and (iii) syngas (60% H_2 and 40% CO% vol.). The relevant fuel properties are listed in Table 3.

3. Results and discussion

3.1. Load and volumetric efficiency

Fig. 2 shows the net indicated mean effective pressure (IMEPn) for the four different fuels as a function of equivalence ratio. As expected, IMEPn decreases with decreasing ϕ due to the lower amount of fuel oxidized and lower heat release. The ϕ ranges for E10-gasoline and ethanol were similar, and combustion with ethanol resulted in comparable IMEPn with gasoline throughout the ϕ sweep. Near stoichiometry, combustion with ethanol resulted in slightly higher IMEPn than gasoline due to the higher in-cylinder pressure for ethanol. This was enabled by the higher laminar flame speed of ethanol and higher octane rating than gasoline, which allowed combustion phasing closer to TDC and more advanced spark timing without knock. However, as the ϕ was reduced, the high latent heat of vaporization of ethanol played an important role and significantly reduced the cylinder temperature, resulting in longer burn duration and lower cylinder pressure than gasoline at lean conditions, thus negatively affecting the work output.

By comparing the two gaseous fuels with gasoline and ethanol, it can be seen that both natural gas and syngas have lower IMEPn than gasoline and ethanol. Natural gas has overall higher IMEPn values than syngas, except for ϕ of 0.63, where the spark advance for natural gas was much higher than syngas, resulting in more heat release late in the compression stroke, which reduced the IMEPn at that point. Natural gas had a ϕ range of 0.63– 0.98. The reduction in IMEPn for natural gas

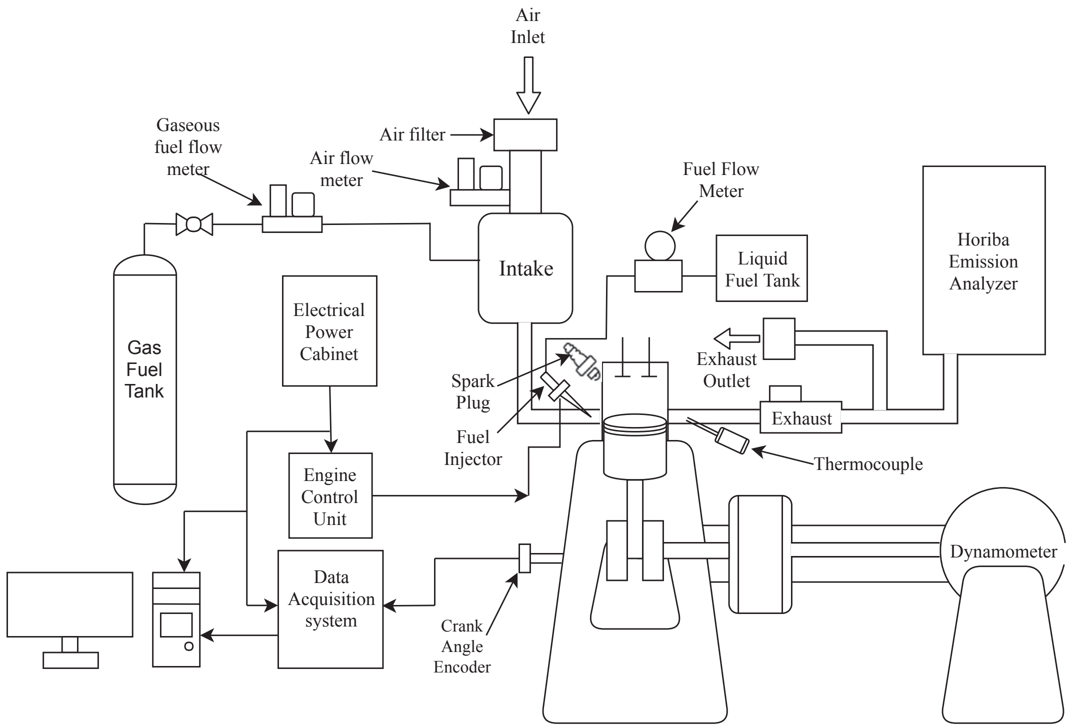

Fig. 1. Schematic diagram of the CFR research engine setup.

Table 2
Spark timing (deg bTDC) for all operating points.

E10-Gasoline		CNG		Ethanol		Syngas	
ϕ	ST (deg bTDC)	ϕ	ST (deg bTDC)	ϕ	ST (deg bTDC)	ϕ	ST (deg bTDC)
0.99	13°	0.98	19°	0.97	18°	0.78	−2°
0.94	14°	0.93	20°	0.93	21°	0.73	0°
0.90	16°	0.89	20°	0.88	22°	0.69	4°
0.85	18.5°	0.83	21°	0.85	23°	0.64	7°
0.79	23.5°	0.79	21°	0.81	26°	0.59	11°
0.74	31°	0.72	22°	0.76	38°	0.54	14°
		0.68	26°			0.50	16°
		0.63	32°			0.46	18°
						0.41	20°
						0.36	22°
						0.31	32°

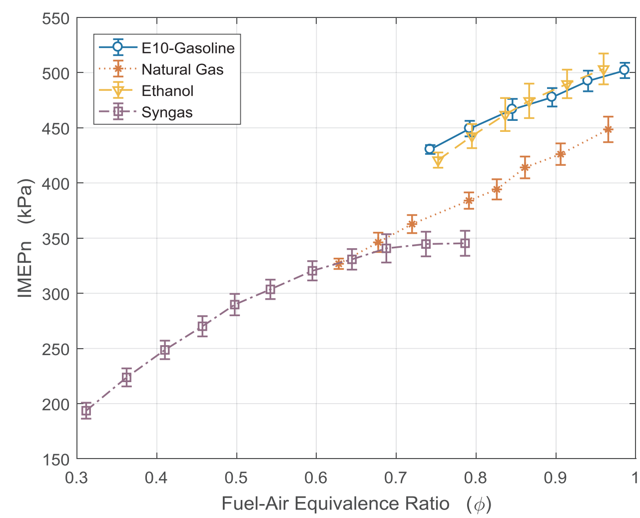

Fig. 2. IMEPn for all fuels with respect to ϕ.

compared to the liquid fuels was mainly due to the gaseous nature of the fuel, which reduced the volumetric efficiency of the engine. The result was lower input fuel energy into the combustion chamber and lower pressure rise during combustion. Natural gas exhibited lower heat release rates than gasoline, which necessitated higher spark advance that negatively affected the useful work output and thus reduced the IMEPn. Syngas had a ϕ range of 0.3– 0.78, limited by the high heat release rates on the upper side. The IMEPn of syngas was the lowest between all fuels at comparable equivalence ratios, due to the lowest volumetric efficiency and lowest LHV.

Fig. 3 shows the engine volumetric efficiency as a function of equivalence ratio for the four different fuels. As the ϕ decreased from close to stoichiometric, the volumetric efficiency increased because of the increasing mass of intake air. Gasoline had the highest volumetric

Table 3
Fuel properties.

Fuels	E10-gasoline	CNG	Ethanol	Syngas
Formula	$C_{6.685}H_{13.386}O_{0.223}$	$C_{1.013}H_{3.965}O_{0.014}$	$C_{1.900}H_{5.800}O_{1.000}$	$C_{0.410}H_{1.181}O_{0.410}$
Molecular Weight [g/mole]	97.3	16.4	44.7	12.7
H/C Ratio	2.002	3.914	3.053	2.880
Lower Heating Value [MJ/kg]	41.8	47.6	25.6	20.5
RON	91–92	~130 [19]	113 [20]	~120 [21]
Stoich AFR	14.0	16.3	8.8	5.4

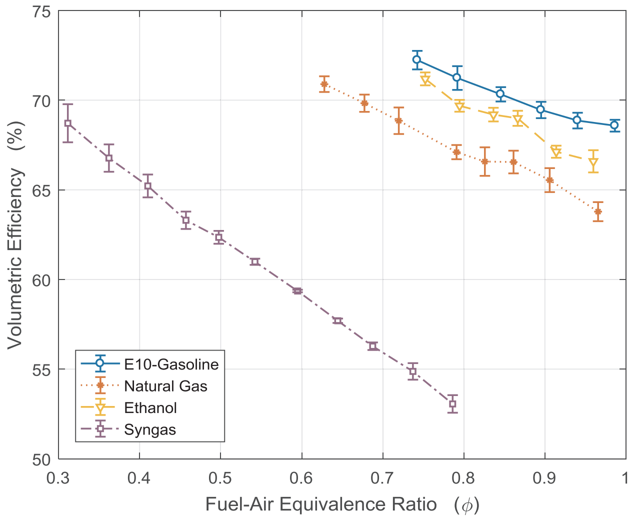

Fig. 3. Volumetric efficiency for all fuels with respect to ϕ.

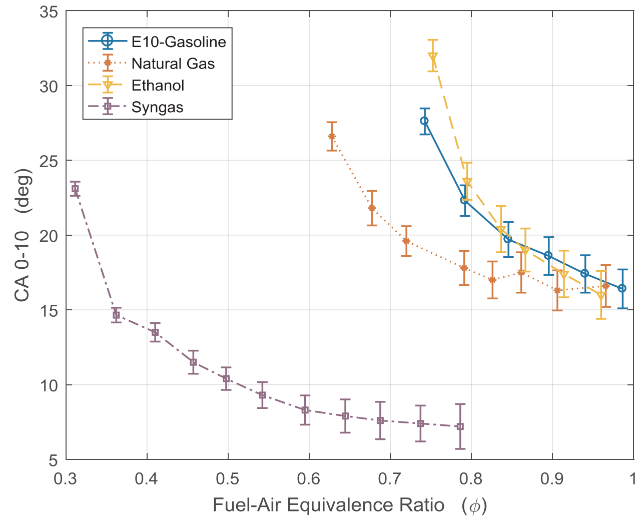

Fig. 5. CA0-10% for all fuels with respect to ϕ.

efficiency among the four fuels, followed by ethanol, natural gas, and syngas. Liquid fuels benefit from their latent heat of vaporization which cools the intake charge thus increasing its density and consequently increasing the volumetric efficiency of the engine. On the other hand, gaseous fuels displace air in the intake manifold, thus reducing volumetric efficiency and eventually heat release and work production. The second factor that affected volumetric efficiency was the different stoichiometric A/F ratio between the four fuels. Fuels with lower stoichiometric A/F ratio required higher mass of fuel injected at the same φ, which reduced the volumetric efficiency. Ethanol has a lower stoichiometric A/F ratio than gasoline, which resulted in lower volumetric efficiency, despite the fact that ethanol has higher latent heat of vaporization than gasoline and thus benefits more from evaporative cooling. Natural gas has the highest stoichiometric A/F ratio, but the gaseous nature of the fuel results in lower volumetric efficiency than both gasoline and ethanol. The lowest volumetric efficiency was measured when the engine was operated on syngas, since syngas had the lowest stoichiometric A/F ratio among all fuels and is a gaseous fuel.

3.2. Heat release rates

Fig. 4 displays the gross heat release rate as a function of crank angle degree for the four different fuels studied at similar ϕ (∼0.8). The E10-gasoline fuel showed the highest heat release rate, followed by syngas, then ethanol, and lastly by natural gas. By comparing the heat release rates of the E10-gasoline with ethanol, it was found that the E10-gasoline has higher heat release rate than ethanol, which was attributed to the lower unburned gas temperature of ethanol (evaporative cooling) and its somewhat lower adiabatic flame temperature. Syngas is shown to have higher heat release rate than ethanol and natural gas due to its highest burning velocity and highest diffusivity among all fuels. The spark timing for syngas was retarded after TDC to avoid high pressure rise rates during combustion. Natural gas had generally the slowest heat release rates, which were primarily attributed to the fact that methane has the lowest burning velocity compared to the other fuels at high pressure and temperature conditions. The result was burn duration being the longest for natural gas as will be shown later.

3.3. Burn duration

Figs. 5 and 6 show the flame development period (0–10% burned mass fraction) and the flame propagation period (10–90% burned mass fraction) as functions of ϕ for all the fuels studied. From Fig. 5 it can be seen that for all the fuels studied the CA0-10% period increases as ϕ decreases due to the lower laminar flame speed and lower heat released during this period. Syngas exhibited the shortest flame development period, ranging from 3 to 24 CAD with ϕ ranging from 0.78 to 0.3

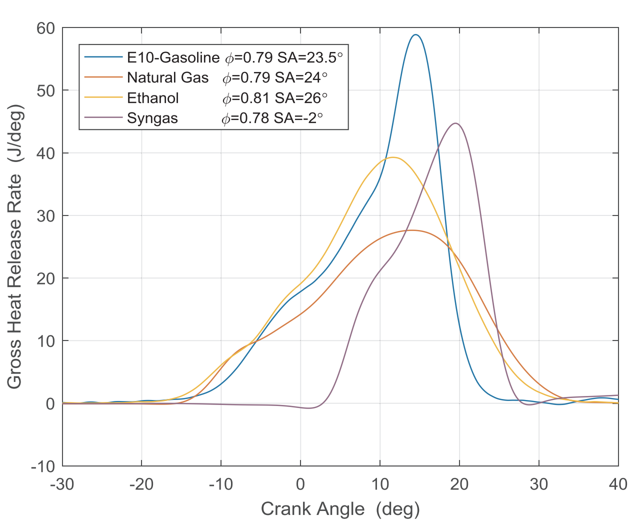

Fig. 4. Instantaneous gross heat release rate for all fuels at φ∼0.8.

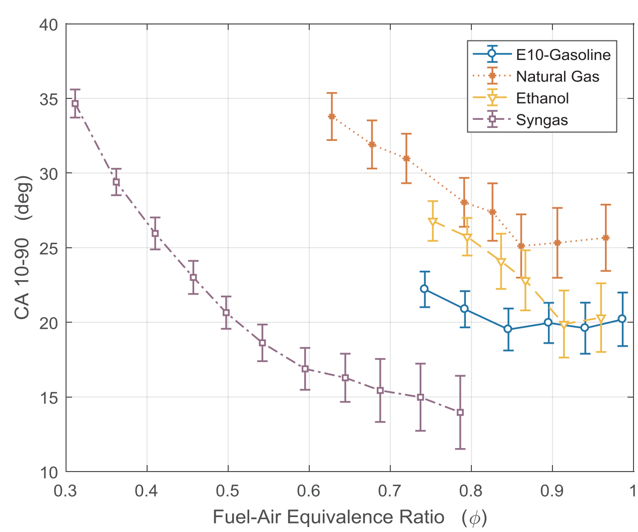

Fig. 6. CA10-90% for all fuels with respect to ϕ.

respectively, which was a result of the high flame speed of hydrogen and its high diffusivity.

The ability of the engine to establish a flame quickly under lean conditions with syngas was the main enabler of its very lean misfire limit. Natural gas showed the second shorter flame development period ranging from 16 to 26 CAD with ϕ ranging from 1 to 0.63 respectively. Natural gas had faster flame development than gasoline and ethanol, due to its higher laminar flame speed at low temperature and pressure conditions. Ethanol and E10-gasoline had similar flame development periods for stoichiometric and near stoichiometric mixtures, but as the mixtures became leaner the flame development period of ethanol increased more than that of E10-gasoline. This effect was attributed to the laminar flame speed of ethanol, which has a sharper decrease with ϕ than the laminar flame speed of gasoline [22]. In addition, the higher latent heat of vaporization of ethanol had an effect on reducing the unburned gas temperature of lean mixtures at spark timing and thus further reducing the laminar flame speed. Based on the experimental data obtained from testing of the four fuels it was concluded that the combustion stability was dependent upon the flame development period, with shorter period resulting in more stable combustion and therefore extended lean misfire limit.

The CA10-90% duration shown in Fig. 6 followed a similar trend to that of CA0-10%. For all fuels studied, the burn duration was minimum at stoichiometry (ϕ of 0.8 for syngas) and increased as the mixtures became progressively leaner. This effect was partially attributed to the slower flame development period at lean conditions and necessitated increasing the spark advance in order to keep the engine operating at MBT. In addition, the lower heat release at lean conditions reduced the fuel oxidation rates, and the lower unburned gas density reduced the unburned mixture entrainment rates in the flame region. Syngas had the fastest flame development period which led to having the fastest flame propagation period as well, aided by the high diffusivity of hydrogen and its high adiabatic flame temperature. The ethanol and E10-gasoline followed a similar trend to the CA0-10% periods shown above. Their CA10-90% durations were similar at conditions close to stoichiometry, but at leaner conditions the higher heat of vaporization of ethanol seemed to be the main factor that affected the unburned mixture temperature and the rate of heat release. The natural gas showed the highest burn duration and the lowest heat release rates, despite the fact that its flame development period was shorter than ethanol and E10-gasoline. Researchers have attributed this effect to the flame speed of natural gas being lower than gasoline at high temperature (>1000 K) and pressure conditions [9]. The result was a need for further spark advance in order to keep the engine at MBT when operated with natural gas.

3.4. Indicated efficiency and combustion phasing

Fig. 7 shows the net indicated efficiency for the four different fuels as a function of ϕ. As expected, the net indicated efficiency generally increased as the mixtures became progressively leaner due to the increasing ratio of specific heats (γ) of the mixtures. Exceptions to this trend were the very lean points that exhibited poor combustion stability and thus affected the net indicated efficiency as well. The E10-gasoline showed overall the highest indicated efficiency levels at its ϕ range and ethanol had marginally lower efficiency values. Natural gas achieved lower efficiency than E10-gasoline and ethanol, primarily because of its longer burn duration, which reduced the expansion ratio. This trend is consistent with the IMEPn behavior presented in Fig. 2 earlier. At ϕ of 0.5 and above, the syngas had lower net indicated efficiency than the other three fuels. In these cases, the heat loss to the walls was greater than the other three fuels, which can be attributed to the shorter quenching distance of hydrogen, which enables the flame to travel closer to the cylinder walls. However, syngas combustion could also be completed at ϕ lower than 0.5, and in these cases the net indicated efficiency increased because of the favorable γ.

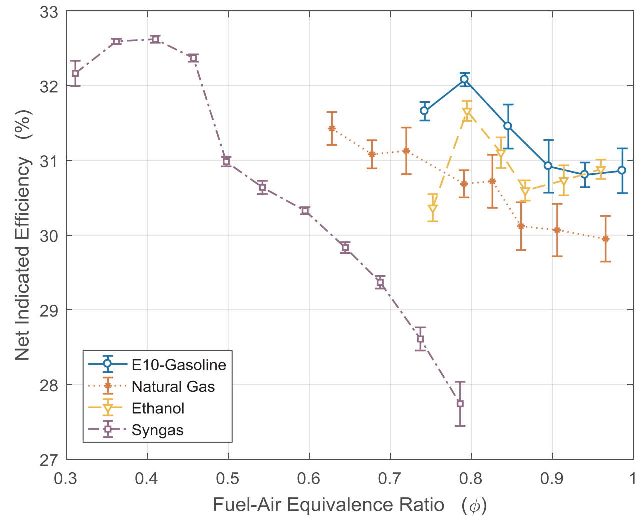

Fig. 7. Net indicated efficiency for all fuels with respect to ϕ.

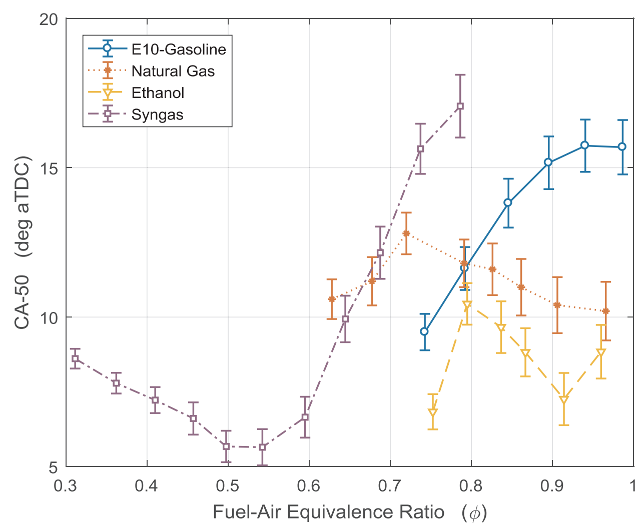

Fig. 8. CA50 for all fuels with respect to ϕ.

Fig. 8 shows the crank angle of 50% burned fraction (CA50) as a function of ϕ for all the fuels studied. The CA50 values shown do not follow the same trend for each fuel because they were dependent on the heat release profile and the spark advance set for MBT at each point. For E10-gasoline, CA50 was advanced as ϕ decreased due to increasing spark advance for MBT. Combustion with ethanol had shorter CA10-90% duration than E10-gasoline, which in turn resulted in earlier CA50 than E10-gasoline. The CA50 of ethanol could be somewhat retarded as ϕ decreased, which helped to slightly increase the net indicated efficiency shown in Fig. 7. At the leanest case, the CA50 of ethanol was again advanced because the spark timing needed to be further advanced. Natural gas had CA50 values that ranged between 10 and 13 CAD aTDC, owing to the low heat release rates and high CA10-90% burn duration, which also impacted the net indicated efficiency. Combustion with syngas was subject to high heat release rates and thus high pressure rise rates at ϕ of 0.7 and above, which necessitated retarding the spark timing in order to control them. The result was retarded CA50 for these cases. For ϕ below 0.7, combustion with syngas reached a minimum CA50 of ~6 CAD aTDC, which was again retarded as the mixture became leaner because of the increased CA10-90% burn duration.

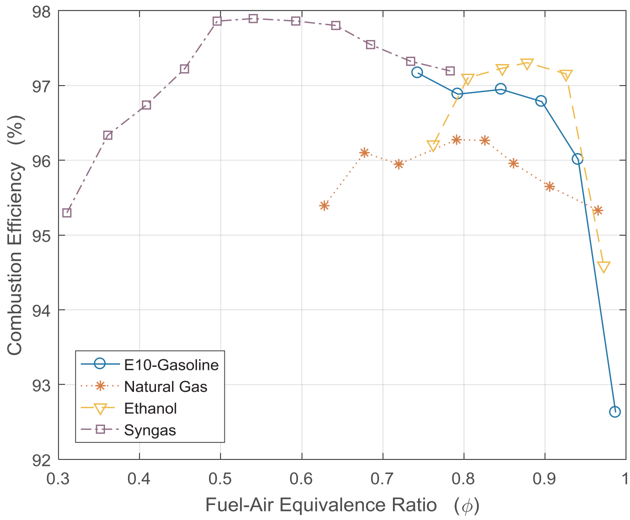

Fig. 9. Combustion efficiency for all fuels with respect to ϕ.

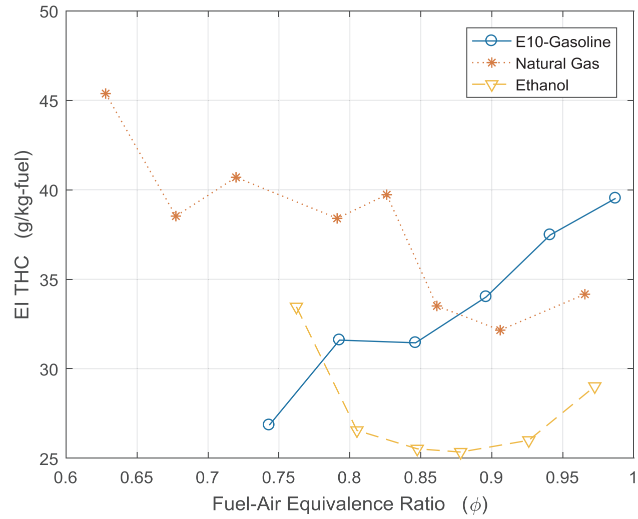

Fig. 10. Emission index of THC for all fuels with respect to ϕ.

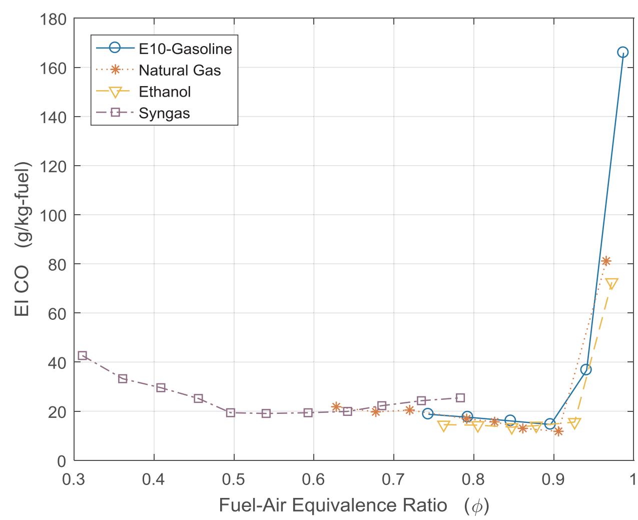

Fig. 11. Emissions index of CO for all fuels with respect to ϕ.

3.5. Combustion efficiency and THC, CO emissions

Fig. 9 shows the combustion efficiency as a function of ϕ for the four different fuels studied. For the E10-gasoline and ethanol, combustion efficiency was the lowest at close to stoichiometric conditions (92.6% for E10-gasoline). This was a result of the CFR engine architecture, which has a quiescent combustion chamber, a side mounted spark plug, and large crevice volumes. However, as ϕ was reduced from stoichiometry, the combustion efficiency increased for both E10-gasoline and ethanol, which was primarily attributed to the oxygen availability that promoted CO to CO_2 conversion. However, as the mixture became too lean, combustion efficiency decreased in all cases because the cylinder temperatures dropped drastically, and flame propagation was slow.

Syngas exhibited the highest combustion efficiency among all fuels due to its chemical composition which contains no hydrocarbon species, its highest flame speed, as well as due to the short quenching distance of hydrogen, which enables the flame to propagate very close to the combustion chamber walls. Ethanol showed higher combustion efficiency than E10-gasoline due to its chemical composition, which contains lower order hydrocarbon species than E10-gasoline as well as oxygen. Natural gas had generally the lowest combustion efficiency (95.3–96.3%), which was attributed to its low heat release rates and long burn duration that led to flame propagation later in the expansion stroke.

Figs. 10 and 11 show the total hydrocarbon (THC) and CO emission indices, respectively, as functions of ϕ for all the fuels studied. THC emissions were direct results of incomplete combustion and were dependent upon the fuel composition. As ϕ was reduced from stoichiometry, combustion efficiency for E10-gasoline increased and thus THC emissions were reduced. The dependence of THC emissions on combustion efficiency was not as strong in the cases of ethanol and natural gas, primarily because of the chemical structure of the fuels, which included lower order hydrocarbon species than E10-gasoline. Combustion with ethanol also resulted in somewhat lower THC emissions than E10-gasoline resulting from the fuel-bound oxygen that promoted hydrocarbon oxidation reactions. Natural gas exhibited the lowest overall combustion efficiency and thus highest THC emissions among all fuels, which was attributed to the low heat release rates. The composition of syngas resulted in zero THC emissions, since it contains no hydrocarbon species in the fuel.

CO formation is also a direct outcome of incomplete combustion. From Fig. 11 it can be seen that the highest CO emissions were measured when the engine was operated close to stoichiometry with gasoline, followed by natural gas and ethanol. However, as the equivalence ratio was decreased, CO emissions quickly reduced because of the high oxygen availability in the mixture and the sufficiently high cylinder temperatures that ensured the successful conversion of CO to CO_2. In the leanest cases, CO started to increase again, which was a direct result of incomplete combustion and poor combustion stability. Syngas exhibited marginally higher CO emissions than the other three fuels studied, despite having the highest overall combustion efficiency. This was a result of the CO content of the syngas mixture (40% CO vol.), thus any unburned fuel was either hydrogen or CO in the exhaust.

3.6. Burned gas temperature and NOx emissions

Fig. 12 shows the peak bulk temperature as a function of ϕ for the four fuels studied and Fig. 13 shows the emission index of NOx for the same cases. In all fuels studied, the peak bulk temperature was reduced as equivalence ratio was reduced because of the resulting reduction in heat release. Combustion with E10-gasoline resulted in the highest peak bulk temperature compared to the other three fuels. Cylinder temperature was directly related to the total mass trapped in the cylinder, the lower heating value of each fuel, and the temperature of the unburned mixture in each case. The E10-gasoline had the highest volumetric efficiency, which resulted in the highest amounts of air and fuel mixture were trapped in the combustion chamber. It also has the second highest LHV among the four fuels studied, and the combination of these

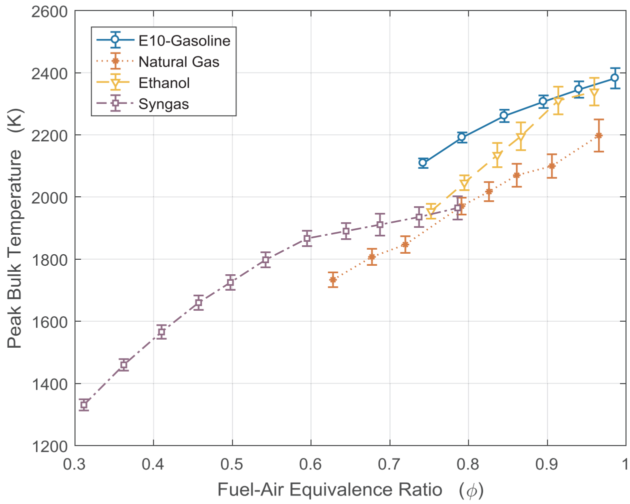

Fig. 12. Peak bulk temperature for all fuels with respect to ϕ.

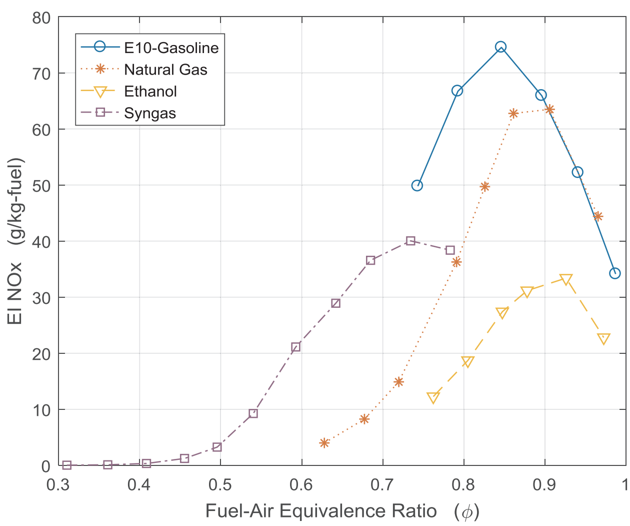

Fig. 13. Emission index of NOx for all fuels with respect to ϕ.

two factors resulted in the highest peak bulk temperatures. Ethanol showed comparable peak temperature to gasoline close to stoichiometry, due to the more advanced spark timing and faster flame speed. However, at leaner cases, the lower heating value of ethanol as well as its high latent heat of vaporization resulted in lower peak bulk temperatures than E10-gasoline. These effects showed to be dominant as the ϕ was further reduced, resulting in larger differences with gasoline in peak bulk temperature.

The lower volumetric efficiency of gaseous fuels resulted in less mass trapped in the cylinder compared to the liquid fuels, which in turn resulted in lower peak temperature, even though the LHV for natural gas was the highest in this study. In addition, the natural gas exhibited the lowest heat release rates, which resulted in lower pressure and temperature rise in the cylinder compared to the other fuels. Previous studies have shown that CO has the highest adiabatic flame temperature among the fuels studied, followed by hydrogen, gasoline, ethanol, and natural gas [23–25]. The high adiabatic flame temperature of hydrogen combined with its high flame speed resulted in syngas combustion having higher peak bulk temperature than natural gas when ϕ ranged from 0.6 to 0.8, despite the lower LHV and lower volumetric efficiency of syngas.

The NOx emission index trends shown in Fig. 13 are direct results of the peak bulk temperature trends shown above. NOx emissions peaked at slightly lean conditions (ϕ ~ 0.9), due to the excess oxygen combined with sufficiently high peak bulk temperatures in all cases (> 1800 K) [26]. As ϕ was decreased, the peak bulk temperatures in all cases decreased accordingly, and eventually were kept below the NOx formation threshold, resulting in very low levels of NOx. Syngas and natural gas had lower lean misfire limits than the liquid fuels, which resulted in lower peak bulk temperatures and virtually no NOx formation at the leanest cases. Syngas was the most effective fuel in reducing NOx emissions because it enables stable SI combustion down to ϕ of 0.3.

Ethanol had higher peak bulk temperatures compared to natural gas and syngas at its ϕ range, but the NOx emission index of ethanol was actually lower than those two fuels because the mass flow rate of ethanol into the engine was higher to compensate for its low LHV and A/F ratio.

4. Conclusions

This study presented a comparison between E10-gasoline, ethanol, natural gas, and syngas as fuels for lean spark ignited combustion. Experiments were conducted on a CFR engine, operated at compression ratio 8:1 and at 1200 rev/min. The equivalence ratio for each fuel was swept from stoichiometry to the lean misfire limit. The key findings can be summarized as:

- Combustion with syngas had the lowest lean misfire limit (ϕ of 0.3), which enabled the engine to achieve high combustion and thermal efficiencies with minimal CO and virtually no NOx emissions at the leanest cases.
- The lean ignition limit was dictated by the ability of the engine to establish a flame in lean mixtures. The ultra-lean operation with syngas was enabled primarily by the high laminar flame speed, diffusivity, and adiabatic flame temperature of the hydrogen present in syngas.
- Natural gas offered the second lowest lean misfire limit, which was lower than that of gasoline and ethanol. Natural gas exhibited short CA0-10% flame development period due to its high laminar flame speed at low temperature and pressure conditions. However, natural gas also had the lowest heat release rates and thus the highest CA10-90% burn duration among all fuels studied. The reasons behind the low heat release rates of natural gas require further investigation.
- Ethanol achieved similar load, combustion efficiency, and net indicated efficiency levels as E10-gasoline. The CO_2 emissions from ethanol combustion were lower than those of E10-gasoline due to the higher H/C ratio of ethanol. However, the CO emissions were comparable with E10-gasoline.
- Peak bulk temperatures resulting from ethanol combustion were lower than E10-gasoline, leading to lower NOx emission index, which also accounted for the higher flow rates of ethanol into the engine.
- The gaseous fuels could not achieve the same load level as the liquid fuels, primarily because of the limited volumetric efficiency of the engine.

The comparison presented in this study was based on the 8:1 compression ratio set at the CFR engine, which was selected in order to remove end gas knock as a limitation of combustion phasing. However, the three alternative fuels studied have higher octane ratings than E10-gasoline, which can be used to promote operation at higher compression ratio, thus enabling higher thermal efficiency without triggering end gas knock in modern, downsized, boosted SI engines. Investigation of natural gas and syngas combustion at higher compression ratios will be a subject of future work.

References

[1] Ayala FA, Gerty MD, Heywood JB. Effects of combustion phasing, relative air-fuel ratio, compression ratio, and load on SI engine efficiency (No. 2006-01-0229). SAE

Technical Paper; 2006.
[2] Cho HM, He BQ. Spark ignition natural gas engines—a review. Energy Convers Manage 2007;48(2):608–18.
[3] Ayala FA, Heywood JB. Lean SI engines: the role of combustion variability in defining lean limits (No. 2007-24-0030). SAE Technical Paper; 2007.
[4] Daniel R, Wang C, Xu H, Tian G. Effects of combustion phasing, injection timing, relative air-fuel ratio and variable valve timing on SI engine performance and emissions using 2, 5-dimethylfuran. SAE Int J Fuels Lubr 2012;5(2):855–66.
[5] Korakianitis T, Namasivayam AM, Crookes RJ. Natural-gas fueled spark-ignition (SI) and compression-ignition (CI) engine performance and emissions. Prog Energy Combust Sci 2011;37(1):89–112.
[6] Bach C, Lämmle C, Bill R, Soltic P, Dyntar D, Janner P, Boulouchos K, Onder C, Landenfeld T, Kercher L, Seel O. Clean engine vehicle a natural gas driven euro-4/sulev with 30% reduced co2-emissions (No. 2004-01-0645). SAE Technical paper; 2004.
[7] Das A, Watson HC. Development of a natural gas spark ignition engine for optimum performance. Proc Inst Mech Eng Part D: J Automobile Eng 1997;211(5):361–78.
[8] Sofianopoulos A, Assanis DN, Mamalis S. Effects of hydrogen addition on automotive lean-burn natural gas engines: critical review. J Energy Eng 2015;142(2):E4015010.
[9] Ben L, Raud-Ducros N, Truquet R, Charnay G. Influence of air/fuel ratio on cyclic variation and exhaust emission in natural gas SI engine (No. 1999-01-2901). SAE Technical Paper; 1999.
[10] Raman P, Ram NK. Performance analysis of an internal combustion engine operated on producer gas, in comparison with the performance of the natural gas and diesel engines. Energy 2013;63:317–33.
[11] Szwaja S, Kovacs VB, Bereczky A, Penninger A. Sewage sludge producer gas enriched with methane as a fuel to a spark ignited engine. Fuel Process Technol 2013;110:160–6.
[12] Furuhama S. State of the art and future trend of hydrogen fueled engine. JSAE Rev 1981:53–68.
[13] Karim GA. Hydrogen as a spark ignition engine fuel. Int J Hydrogen Energy 2003;28(5):569–77.
[14] Bauer CG, Forest TW. Effect of hydrogen addition on the performance of methane-fueled vehicles. part I: effect on SI engine performance. Int J Hydrogen Energy 2001;26(1):55–70.
[15] Costa RC, Sodré JR. Hydrous ethanol vs. gasoline-ethanol blend: engine performance and emissions. Fuel 2010;89(2):287–93.
[16] Schifter I, Diaz L, Rodriguez R, Gómez JP, Gonzalez U. Combustion and emissions behavior for ethanol–gasoline blends in a single cylinder engine. Fuel 2011;90(12):3586–92.
[17] Nakata K, Utsumi S, Ota A, Kawatake K, Kawai T, Tsunooka T. The effect of ethanol fuel on a spark ignition engine (No. 2006-01-3380). SAE Technical Paper; 2006.
[18] Li L, Liu Z, Wang H, Deng B, Xiao Z, Wang Z, Gong C, Su Y. Combustion and emissions of ethanol fuel (E100) in a small SI engine (No. 2003-01-3262). SAE Technical Paper; 2003.
[19] Sierens R, Rosseel E. Variable composition hydrogen/natural gas mixtures for increased engine efficiency and decreased emissions. J Eng Gas Turbines Power 2000;122(1):135–40.
[20] Awad OI, Mamat R, Ali OM, Sidik NAC, Yusaf T, Kadirgama K, et al. Alcohol and ether as alternative fuels in spark ignition engine: a review. Renewable Sustainable Energy Rev 2017.
[21] Han T, Lavoie G, Wooldridge M, Boehman A. Effect of syngas (H 2/CO) on SI engine knock under boosted EGR and lean conditions. SAE Int J Eng 2017;10(2017–01-0670):959–69.
[22] Wang X, Chen Z, Ni J, Liu S, Zhou H. The effects of hydrous ethanol gasoline on combustion and emission characteristics of a port injection gasoline engine. Case Stud Therm Eng 2015;6:147–54.
[23] Borate NS. Flame Temperature Analysis and Nox Emissions for Different Fuels. Michigan Technological University; 2014. Retrieved on 24.
[24] Reed RJ. North American Combustion Handbook. North American Mfg. Co.; 2014.
[25] Law CK, Makino A, Lu TF. On the off-stoichiometric peaking of adiabatic flame temperature. Combust Flame 2006;145(4):808–19.
[26] Heywood JB. Internal Combustion Engine Fundamentals. Singapore: Mc Graw Hill International Editions; 1988.