Influence of Combustion Duration on the Performance and Emission Characteristics of a Spark-Ignition Engine Fueled with Pure Methanol and Ethanol

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ABSTRACT: In this research, we estimated and summarized the effects of combustion duration on the performance and emission characteristics of a spark-ignition engine using pure methanol and ethanol as fuels, which have not been previously presented. From the results, we demonstrated that an increase in combustion duration causes a decrease in peak firing temperature and peak firing pressure and an increase in trapped residual gas. The level of trapped residual gas when using ethanol as fuel is higher than that of methanol fuel. The indicated mean effective pressure (IMEP) and brake mean effective pressure (BMEP) increase to maximum values and then decrease with increasing combustion duration, while the brake specific fuel consumption (BSFC) reaches a minimum value and then increases. The optimal BSFC improved to 33.31% when the engine used ethanol fuel instead of methanol. The increase in combustion duration helps to reduce NOx and HC emissions, but an increase in CO emissions is observed.

1. INTRODUCTION

Harmful effects on health and the environment led to the introduction of emission legislation that prescribes the allowed emission levels. Most of the vehicle types, along with passenger cars, are required to satisfy European emission standards. Recently, various effective techniques have been applied to improve engine power and toxic gas emissions, such as optimizing operating parameters like ignition timing, injection timing, compression ratio, air–fuel ratio, and valve overlap. Furthermore, alternative fuels have been investigated as an effective method to reduce toxic gas production and solve the fossil fuel crisis. Historically, methanol and ethanol are known as potential alternative fuels for combustion engines, with many benefits compared to gasoline, including the use of a higher compression ratio to improve engine power because of a higher octane number and improve engine emission due to a higher oxygen ratio, a higher flammability limit, and a low carbon-to-hydrogen ratio. That is why methanol and ethanol are excellent alternative fuels for spark-ignition engines.

Mourad et al. studied the effects of ethanol/butanol–gasoline fuel on the emissions and performance of a spark-ignition engine. In their research, various blends of the ethanol/butanol–gasoline ratio were used (2, 5, 10, 15, and 20%). They found that engine emission was improved with an 8.22% decrease in CO, 25.2% in HC, and 8.22% in fuel consumption. However, engine power may decrease up to 11.1% depending on conditions. Elfasakhany et al. presented the effect of the ethanol–methanol–gasoline blends on the performance and emission characteristics of a spark-ignition engine. In this research, the addition of methanol and ethanol was from 3 to 10% in volume. The results of engine performance will be compared between three cases: pure gasoline, methanol–gasoline blends, and ethanol–gasoline blends. They found that when engine fueled ethanol–methanol–gasoline blends, the CO and HC emission decreased and engine torque was improved compared to pure gasoline fuel. With methanol–gasoline blends as fuel, the engine performed the lower 5.5% CO and 6% HC emission when compared to ethanol–gasoline blends. In the further study, the results also showed that with the use of ethanol–methanol–gasoline blends as fuel, the engine power and engine emission (CO and HC emission) were improved when compared to n-butanol–isobutanol–gasoline blends or ethanol–isobutanol–gasoline blends.

In other studies, the performance and emission of the engine using gasoline, pure methanol, and ethanol were presented.
Balki et al.\textsuperscript{15} performed an experiment on a small engine with a single cylinder to study the effects of ethanol and methanol fuels on engine performance and engine emission characteristics. The results show that the engine achieved a higher torque and lower toxic emission (NO\textsubscript{x}, HC, and CO) when using alcohol compared to gasoline. However, higher brake-specific fuel consumption and CO\textsubscript{2} emission were also observed. In another study,\textsuperscript{16} the effects of the compression ratio on the emission and performance of a spark-ignition engine with three kinds of fuels (ethanol, methanol, and gasoline) were evaluated. The testing condition comprised a 2400 rpm engine speed and different compression ratios of 8:0:1, 8:5:1, 9:0:1, and 9:5:1. The result shows that the engine performed better with lower emission when using ethanol and methanol at all compression ratios compared to gasoline as fuel. Further, in high-compression-ratio conditions, a faster cylinder gas pressure increase and heat release rate were noted. Çelik et al.\textsuperscript{17} reported on gasoline engine performance when using pure methanol fuel at a high compression ratio. In their research, a small spark-ignition engine was tested at several compression ratios (6:1, 8:1, 10:1, gradually). They discovered, at a compression ratio of 6:1, reduction of NO\textsubscript{x}, CO\textsubscript{2}, and CO in exhaust emission without engine power loss when methanol was used as fuel instead of gasoline. However, upon increasing the compression ratio from 6:1 to 10:1, the NO\textsubscript{x}, CO\textsubscript{2}, and CO emissions were reduced, while engine power and thermal efficiency increased by 14 and 36%, respectively.

In the combustion stroke of a spark-ignition engine, combustion duration is a sensitive factor that affects thermal efficiency, residual gas, and engine emission characteristics. A shorter combustion duration is due to incomplete conversion of chemical energy to thermal energy.\textsuperscript{18} Conversely, a longer combustion duration leads to increased heat transfer to the piston, cylinder, and exhaust gas.

Yamin et al.\textsuperscript{19} reported the effects of combustion duration on an SI engine with hydrogen as fuel. Five engine speeds were considered: 1000, 1500, 2000, 2500, and 3000 rpm. The combustion duration was from 2 to 7 ms. They found that a longer combustion duration helped decrease NO\textsubscript{x} emission but at reduced engine thermal efficiency because of heat loss. Khao et al.\textsuperscript{20} determined the influence of combustion on gasoline engine performance and emission characteristics. A small gasoline spark-ignition engine was utilized to determine the engine emission characteristics at various engine speeds; the combustion duration band is from 40 to 110 °CA. The results of the research show that a higher optimal combustion duration can be observed at a higher engine speed. At the optimal combustion duration, the engine yields the maximum brake torque and the minimum BSFC. NO\textsubscript{x} and HC emissions decreased with increasing combustion duration, while CO emission increased. Yousufuddin et al.\textsuperscript{21} studied the effect of combustion duration on a dual fuel hydrogen–ethanol CI engine. The engine test speed was 1500 rpm, while the amount of hydrogen was increased from 0 to 80% by volume. The engine gave the best performance at a combustion duration of 35–42 °CA. Lata et al.\textsuperscript{22} presented the influence of combustion duration on dual-fuel diesel–LNG engine performance and NO\textsubscript{x} emission. In their search, they found that combustion duration increased up to 6° when the engine was working at light load. The increase of combustion duration has a sensitive effect on engine thermal efficiency and engine emission. The thermal efficiency increased 16.7% and reducing NO\textsubscript{x} emission was presented.

From the summary of the above published articles, the significant effects of combustion duration on engine thermal efficiency and toxic gas emissions were confirmed. Based on the fuel type, combustion duration presents a difference in thermal efficiency and pollution generation.

Ethanol and methanol are alternative fuels that have a high heat vaporization rate, oxygen ratio per molecule, and flammability limit. Based on these characteristics, particular combustion phenomena (flame speed, optimal combustion duration, homogeneity of the air–fuel mixture, combustion phases, and peak pressure increase during the combustion stroke) are different when compared to other fuel types (e.g., gasoline, diesel, hydrogen). Despite significant understanding, almost all of the previous articles presented the effect of alternative fuel addition rates or pure alternative fuels on engine performance. There is no data literature discussing and comparing the effects of combustion duration on engine performance and toxic gas emissions when using pure ethanol and methanol as fuels. This significant gap in the research of the combustion duration factor is filled in this paper. The effects of combustion on the performance and emission characteristics of an SI engine using pure ethanol and methanol as fuels are addressed in detail. Here, the effect of combustion duration on internal exhaust residual gas is investigated. The increase in combustion duration effects on peak temperature, peak pressure increase, and NO\textsubscript{x}, HC, and CO emissions is investigated and discussed in detail. To achieve this goal, a modified spark-ignition engine utilizing pure methanol and ethanol as fuels is used. The spark-ignition engine is modeled based on AVL Boost software. This simulation model will help precisely control the combustion duration.

2. EXPERIMENTAL SYSTEM AND ENGINE MODELING

2.1. Experimental Setup and Properties of Fuel. In this research, a V-twin cylinder small gasoline spark-ignition engine was investigated. The displacement volume of each cylinder is 137 cm\textsuperscript{3}. Each cylinder has four valves (two exhaust valves and two intake valves). Two separate camshafts were used to control intake valves and exhaust valves. The specifications of the testing engine are shown in Table 2. The experimental fuel properties are shown in Table 2. The experimental setup is shown in Figure 1 and the experimental setup schematics are presented in Figure 2.

| parameter | unit | value |
|-----------|------|-------|
| compression ratio | 11:8:1 |
| cylinders | 2 |
| bore–stroke | mm | 57–53.8 |
| intake valve and exhaust valve | mm | 2 × 2 |
| connecting rod | mm | 107.9 |
| 1 cylinder displacement | cm\textsuperscript{3} | 137 |

An AVL AG150, 150 kW, water brake dynamometer (2) was controlled by a dynamo testing system controller (1). The experimental engine (4) was connected with a dynamometer through a connecting shaft (3). The engine speed signal was determined using an encoder (E4058-1800-3-T-24) (6) on a flywheel (5). A K-type sensor (7) with a response time of less than 10 ms was located on the engine head to determine combustion temperature. A fuel pump (9) transported fuel in the fuel tank (8) through a fuel filter (10) to the injector 16450-
C12-23S (11). An oxygen sensor L36C-18-8G1 (13) and (12) was located on the exhaust gas system to determine oxygen and the temperature of exhaust gas flow. The cylinder pressure was determined using a pressure sensor Kistler 6056A (14). The intake air flow passed into the cylinder through an air cleaner box (15) and a throttle 89452-22090 (16). To maintain the air flow’s temperature from 29.5 to 30 °C and measure the air mass flow, an air heater (17) and an air flow meter (18) were located on the intake system. The ECU was used to control injection fuel mass and ignition timing. A Horiba MEXA-7100DEGR was used to analyze exhaust gas characteristics. A computer (22) and a monitor (19) were used to control and observe data, respectively.

The experimental engine speed was at 5000 rpm. The engine oil temperature was kept at 80 °C and the engine was cooled by air. The injector controller was used to maintain the air–fuel ratio as the stoichiometric ratio for each fuel: 14.7:1 with gasoline fuel, 9:1 with ethanol fuel, and 6.47:1 with methanol fuel. Here, to use methanol and ethanol as fuels, the engine was slightly modified using the suitable supplied fuel system and fuel injectors (10 × Φ0.3 mm holes). The pressure controller was used to manage the fuel pressure through the pressure control valve. Additionally, a LabVIEW program installed on the NI computer was used to control the injection timing and injection duration using feedback from the air mass flow sensor on the intake manifold. The engine worked at steady-state conditions and the throttle angle was at full opening. All of the tested devices were calibrated before the experiment.

### 2.2. Engine Modeling

A simulation model was employed to control the combustion duration and estimate the engine exhaust residual gas under various testing conditions. AVL Boost software was able to simulate almost all combustion engine types: SI engines,24–26 CI engines,27 turbocharged engines,28 and alternative fuel engines.29,30 Figure 3 presents a V-twin engine simulation model. Element E1 allows setup engine working conditions at the steady state or transient state. All of the selected simulation output parameters can be observed on monitor MNT1. To define the system

| Table 2. Comparison of the Properties of Fuels |
|---------------------------------------------|
| property | unit | ethanol | methanol | gasoline |
| chemical formula | C\textsubscript{2}H\textsubscript{5}OH | C\textsubscript{2}H\textsubscript{5}OH | CH\textsubscript{3}OH | |
| molecular weight | g/mol | 46.07 | 32.04 | 95–120 |
| boiling point | °C | 78.3 | 64.5 | 27–225 |
| density | g/cm\textsuperscript{3} at 20 °C | 0.790 | 0.792 | 0.72–0.76 |
| latent heating value | kJ/kg | 26,900 | 20,100 | 44,300 |
| heat of vaporization | kJ/kg | 923 | 1178 | 349 |
| stoichiometric air/fuel ratio | | 9 | 6.46 | 14.6 |
| oxygen | wt % | 34.73 | 49.94 | |
| research and motor octane number | | 108.6–89.7 | 108.7–88.6 | 95–85 |
| vapor pressure | kPa at 20 °C | 5.9 | 12.8 | 45–90 |
| flammable limits | vol % | 3.5–15 | 5.5–36.5 | 1.4–7.6 |
boundary conditions, elements SB1 and SB2 were used. The function of the CL1 element is of an air filter in the intake system. The throttle opening angle can be controlled at element TH1. The R1 element was used to define the restriction in the exhaust and intake tubes. The air flow characteristics in the intake and exhaust tubes can be determined by measuring MP1 and downstream pressures (Pa), respectively, and T1 and T2 are the up- and downstream temperatures (K), respectively.

The trapped residual gases in the combustion chamber are measured as follows:

\[
\text{Table 3. NO}_x \text{ Formative Reactions}
\]

The NO \(_x\) emission is produced from six reactions as shown in Table 3. \(^3\)

The NO, NO\(_x\), HC, and CO emissions will be focused on and mentioned as the composition of the exhaust gas. The formation of the exhaust gas component is defined below.

\[
\text{Table 4. CO Formative Reactions}
\]

\[
\begin{align*}
Q_f &= A\cdot q_{\text{coeff}}\cdot(T_c - T_w) \\
C &= \frac{K}{K - 1} \left[ \frac{P_1}{P} \right]^{2/k} \left( \frac{P}{P_2} \right)^{(K+1)/K}
\end{align*}
\]

where \(A\) is the effective area of flow (\([-\]), \(P_1\) and \(P_2\) are the up- and downstream pressures (Pa), respectively, and \(T_1\) and \(T_2\) are the up- and downstream temperatures (K), respectively.

The trapped residual gases in the combustion chamber are measured as follows:

\[
\text{Table 4. CO Formative Reactions}
\]

\[
\begin{align*}
\text{Table 3. NO}_x \text{ Formative Reactions}
\end{align*}
\]

\[
\begin{align*}
\text{Table 4. CO Formative Reactions}
\end{align*}
\]
The CO emission was calculated by eq 931
\[
\theta = C_{\text{const}} \cdot (r_1 + r_2) \cdot (1 - \theta)
\] (9)
with \( \theta = \frac{C_{\text{CO}, \text{act}}}{C_{\text{CO}, \text{equ}}} \).

The HC emission was determined from eq 1031
\[
m_{\text{HC}} = \frac{P_c \cdot V_{\text{crevice}} \cdot M}{R \cdot T_{\text{piston}}}
\] (10)
where \( V_{\text{crevice}} \) is the total crevice volume (m³), \( P_c \) is the temperature of the cylinder (K), \( R \) is the gas constant (J/(kmol K)), \( T_{\text{piston}} \) is the temperature of the piston (K), and \( M \) is the unburned molecular weight (kg/kmol).

The BMEP was determined from eq 1131
\[
\text{BMEP} = \text{IMEP} - \text{FMEP} - \text{SMEP}
\] (11)
where \( K_{\text{cycle}} \) is the simulation cycle parameter (cycle) and \( V_D \) is the displacement volume (m³).

The effective engine torque was calculated by eq 1231
\[
T_{\text{eff}} = \frac{\text{BMEP} \cdot V_D}{k_{\text{cycle}} \cdot \pi}
\] (12)

The brake-specific fuel consumption (BSFC) was calculated by eq 1331
\[
\text{BSFC} = \frac{m_{\text{air}} \cdot n_c \cdot 2.16 \times 10^9}{AFR \cdot \text{BMEP} \cdot V_D \cdot n}
\] (13)
where \( m_{\text{air}} \) is the air mass flow (kg/s), \( n_c = 2 \) for four-stroke engines (−), and \( n \) is the engine speed (rpm).

3. RESULTS AND DISCUSSION

3.1. Model Validation. The simulation model is validated based on a comparison with the experimental data. In Figure 4, the black curves describe the experimental results with gasoline fuel, while the red curves describe the simulation results.
We can observe the same ignition timing between simulation and experiment data in Figure 4 because the experimental ignition timing is the input data for the simulation model. The difference in simulation and experimental data in air mass flow was 2.5%, so this difference is acceptable because the air mass flow is an average value. The peak firing temperature in the combustion stroke has a sensitive effect on the NO\textsubscript{x} formation, while a high pressure increase helps the engine achieve the maximum cylinder pressure in a short time. This may help reduce heat loss and improve engine efficiency. The differences in peak temperature and the peak pressure increase in the simulation and experimental results were 5.3 and 4.5%, respectively. The difference in engine torque was 1.2% at 5000 rpm and can be observed in Figure 4. In this research, the NO\textsubscript{x} and HC emissions were validated. When comparing simulation NO\textsubscript{x} and HC emissions to experimental data, differences of 6.5 and 4.3%, respectively, were presented.

All of the above simulation and experimental output data were identical, indicating the good accuracy of the simulation model made by AVL Boost software for the prediction of engine performance. Therefore, this simulation model can be used for further study.

3.2. Combustion Duration Influences on the Performance and Emission of the Engine Fueled with Pure Methanol and Ethanol. Following the increase in combustion duration, the peak temperature and peak firing pressure increase show a downward trend. This can be explained by the fact that an increase in combustion duration is due to an increase in the combustion displacement volume and cylinder-piston surface area. A larger combustion displacement volume led to a reduced
peak firing pressure, while an increase in the surface areas of the cylinder and the piston led to increased heat loss and peak temperature. Figures 5 and 6 show that when the combustion duration is increased from 40 to 80 °CA, in the case of using ethanol fuel, the peak pressure and peak temperature decrease from 98.2 to 79.1 bar and 2727 to 2478 K, respectively, and in the case of using methanol fuel, the peak pressure decreases from 95.9 to 78.5 bar, while the peak temperature decreases from 2678 to 2464 K.

The residual gas fraction is known to effectively influence the energy efficiency and pollution of an engine. The amount of the trapped exhaust residual gas will affect the toxic products emitted in the next combustion stroke. In the combustion stroke, a shorter combustion duration may cause burning to occur when the piston moves to the top dead center, resulting in increased pumping loss. Furthermore, a longer combustion duration extends the burning process until the pistons are far from the top dead center or even until the exhaust stroke. This process may result in increased reverse exhaust gas flow because of the high temperature and pressure in the exhaust gas. The reverse flow would restrict the fresh air–fuel mixture into the combustion chamber. These discussions explain the increase in trapped residual gas with increasing combustion duration (Figure 7). When the combustion duration increases from 40 to 80 °CA, the level of residual gas is higher for ethanol fuel in comparison with methanol fuel. This is because methanol has a higher carbon-to-hydrogen ratio but a lower oxygen ratio per molecule than ethanol (Table 2). Considering the same air mass, the unburned fuel and combustion products will be higher.

The residual gas ratio increased from 0.77 to 0.96% with ethanol and from 0.66 to 0.74% with methanol as fuel.
Given the above discussion, the cases of both shorter and longer combustion durations are unexpected because of brake thermal efficiency loss. As an optimal combustion duration can help an engine achieve optimal engine efficiency, when the combustion duration increases from 40 to 80 °CA, the IMEP and BMEP increase to maximum values and then decrease, as observed in Figures 8 and 9.

At an engine speed of 5000 rpm, the engine shows the best IMEP and BMEP at 60 °CA combustion duration with ethanol fuel and at 70 °CA with methanol fuel. In the case of methanol fuel, the maximum IMEP and BMEP were 11.2 and 8.5 bar, respectively. In the case of ethanol fuel, the maximum IMEP and BMEP were 11 and 8.3 bar, respectively. Because methanol has higher heat of vaporization and oxygen content per molecule than ethanol, the engine performs higher IMEP and BMEP when using methanol fuel.

Through eq 11, the effect of IMEP on BMEB is described, explaining why the BMEP and IMEP show a similar trend.

A decrease in BMEP led to a reduced engine torque as calculated by eq 12.

BMEP has a significant effect on engine torque. Similar trends to BMEP and engine torque are present in Figures 9 and 10, respectively. The maximum BMEP and the maximum engine torque are dropped at the optimal combustion duration value. At 5000 rpm, the optimal combustion duration is 60 °CA with ethanol fuel and 70 °CA with methanol fuel. At this optimal combustion duration value, the engine performed a maximum engine torque of 18.4 Nm with ethanol fuel and of 18.5 Nm with methanol fuel.

The brake-specific fuel consumption (BSFC) is an important factor that reflects the fuel efficiency of an engine (eq 13).
With the constant air−fuel ratio and engine speed, the BMEP has the largest effect on BSFC. Opposite trends of BMEP and BSFC are shown in Figures 9 and 11, respectively. The BSFC decreased until reaching a minimum value after that increase even combustion duration still increases. The minimum BSFC was 637 g/kWh with ethanol fuel and 849.2 g/kWh with methanol fuel. The optimal BSFC is improved by 33.3% when using ethanol fuel instead of methanol.

Figure 12 shows the impact of combustion duration on NO\textsubscript{x} emission. The NO\textsubscript{x} emissions tend to decrease with increasing combustion duration for two reasons. The first reason is a decrease in peak firing temperature (Figure 5). The second reason is an increase in residual gas (Figure 7), causing a diluted air−fuel mixture and a lower oxygen level. In diesel engines, exhaust gas recirculation is an effective method to reduce NO\textsubscript{x} emission.\textsuperscript{32,33} In our study, when combustion duration increased from 40 to 80 °CA, NO\textsubscript{x} emission decreased from 12.9 to 0.3 g/kWh with ethanol fuel and from 8.4 to 0.2 g/kWh with methanol. The engine produces less NO\textsubscript{x} emission with methanol compared to ethanol fuel.

A short combustion duration leads to incompletely burned fuel. A long combustion duration allows more time to completely burn fuel. This explains why a decrease in HC emission is observed when combustion duration increases. Figure 13 shows a downward trend of HC emission with methanol and ethanol fuels. When combustion duration increases from 40 to 80 °CA, HC emission decreases from...
10.1 to 3.6 g/kWh with methanol fuel and from 6.06 to 2.4 g/kWh with ethanol fuel.

CO emission is a product of the chemical reaction between hydrocarbon and oxygen, so a decrease in unburned HC leads to an increase in CO emission. Another contributing factor to increasing CO emission when the combustion duration increases is the increase in residual gas because a larger amount of residual gas reduces the amount of the fresh air–fuel mixture in the combustion chamber, and a lack of oxygen for the chemical reaction from CO to CO$_2$ will occur. The two above reasons explain why the CO emission decreases until achieved a minimum value and then increases (Figure 14). The minimum CO emission was 80.9 g/kWh using ethanol fuel and was 41.8 g/kWh using methanol fuel. The lower CO emission for methanol fuel in comparison with ethanol fuel is observed. This is because of the higher oxygen contents per molecule and lower residual gas of methanol fuel.

4. CONCLUSIONS

Through a combination of experimental and simulation methods, the influence of combustion duration on the performance and toxic gas emission of an alternative fuel engine was completely investigated. Higher combustion duration leads to a decrease in peak firing pressure and temperature and an increase in internal residual gas. The level of the residual gas fraction is higher for ethanol fuel in comparison with methanol fuel. The results of the research showed that the optimal combustion durations when an engine uses pure ethanol and methanol as fuels are different. These optimal combustion durations are 60 and 70°, respectively. At these optimal combustion durations, the engine performed the maximum engine brake torque and the BSFC could be improved by 33.3% using ethanol instead of methanol. By increasing the combustion duration, NO$_x$ and HC emissions could be decreased but CO emission could be increased. The combustion duration shows a larger effect on NO$_x$ emission with ethanol fuel than that with methanol fuel. A lower CO emission was observed when the engine used methanol fuel instead of ethanol. In this research, only NO$_x$, HC, and CO were mentioned as the composition of the exhaust gas, and exhaust particulate matter and smoke were not mentioned and discussed, so this is a limitation and this limitation will be eliminated by the research team in future works.

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Notes

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