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Comparative Study on Methanol (M100) and Ethanol (E100) Fueled Otto Cycle Engine

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Abstract

An experimental investigation was conducted to study the combustion, performance, and emissions of an Otto cycle engine fueled with ethanol (E100) and methanol (M100). The experimental tests were performed at speeds of 3500 rpm, 3800 rpm, and 4200 rpm at their respective maximum torque with both fuels. The torque obtained was maximum (7 Nm) at 3800 rpm. It was observed from the experimental results that In-cylinder peak pressure at all the speeds was higher with M100 than E100. The cumulative heat release curve indicated that the start of combustion advanced with M100. The crank angle for 10% and 50% mass fraction burnt occurred earlier with M100 than E100. The Brake thermal efficiency of the engine increased by more than 30% with M100 compared to E100. Volumetric efficiency of the engine increased marginally (5-9%) with E100 than that of M100. Carbon monoxide (CO) emission decreased by 32.9%, 52.9 %, and 7.3% at 3500 rpm, 3800 rpm, and 4200 rpm, respectively with M100 compared to E100. Hydrocarbon (HC) emission with M100 decreased by 14.2, 60%, and 67% at 3500 rpm, 3800 rpm, and 4200 rpm respectively. NOx emission with M100 increased by five times than E100 at 3500 rpm, while at 4200 rpm, it increased by 60%. It is concluded that the engine's combustion, performance, and emission characteristics were better with methanol (M100) than ethanol (E100).

Keywords: Combustion, Ethanol, Emission, Methanol, Performance

Research Article

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1. Introduction

Energy security is a concern worldwide. Conventional fuels are limited while the energy demand is rising rapidly. Therefore, various alternative fuels are being tested to replace the existing petroleum-based fuels. Alcohols are one of the alternative fuels to conventional petrol and diesel which can substitute them in internal combustion (IC) engines. Methanol, ethanol, propanol, and butanol are some of the alcohols used as fuels in spark ignition (SI) engines. Methanol is the simplest alcohol containing a single carbon atom. It is generally produced from coal and can be produced from renewable resources [1,2]. Ethanol can be produced from various biomass feedstocks such as sugarcane, corn, agricultural residue, etc. [3]. Ethanol and methanol have less carbon to hydrogen ratio than gasoline. Therefore, carbon-based emissions are less with the fuels. The octane number of both ethanol and methanol is more than 100 and, therefore can be used in a high compression ratio SI engine. Hence, higher thermal efficiency could be achieved with them. In addition to this, the flame velocity of both fuels is higher than gasoline. Research octane number (RON) and Motor octane number (MON) of ethanol are 108.6 and 89.7, while methanol is 108.7 and 88.6. Both these alcohols can be blended with gasoline in various ratios. The performance of the SI engine is enhanced with the blends. Moreover, emissions of carbon monoxide (CO), hydrocarbon (HC), and oxides of nitrogen (NOx) are decreased with the use of blends [4-6]. Chen et al. [7] compared the combustion and cycle by cycle variations in a SI engine fueled with ethanol, methanol, and butanol. The engine speed was constant at 1600 rpm, and the air-fuel ratio was varied from 1 to 1.5. They reported that methanol yielded the maximum In-cylinder peak pressure and the highest heat release rate, followed by ethanol and butanol, respectively. Methanol exhibited the highest burning rate and the lowest cycle by cycle variations compared to other fuels. Tian et al. [8] reported that the addition of alcohols to gasoline increases the brake thermal efficiency (BTE) and reduces carbon monoxide and carbon dioxide (CO2) emissions from the engine. The effects were prominent with methanol as compared to ethanol and gasoline. Balki et al. studied the SI engine's performance, combustion, and emissions fueled with ethanol, methanol, and gasoline at variable speeds and maximum throttle opening. They reported that with alcohols, maximum torque increased by

3.7% and 4.7%, respectively, with ethanol and methanol. The brake-specific fuel consumption with methanol increased by 84% than gasoline. BTE of the engine was almost equal with ethanol and methanol at all the engine speeds. The Combustion efficiency was the highest with methanol than that of ethanol and gasoline [9]. Pourkhesalian et al. [10] compared various fuels including hydrogen, methane, propane, methanol, ethanol, and gasoline for the performance and emissions of a SI engine. They reported that liquid fuels produced more power than gaseous fuels and brake specific fuel consumption with methanol was the highest. Brake-specific NOx emission was the lowest with methanol. Balki and Cayin [11] reported the effects of variable compression ratio (CR) (varied from 8, 8.5, 9, and 9.5:1) on performance, emission, and combustion of a SI engine fueled with ethanol, methanol, and gasoline. At 8.5:1 CR, BMEP increased by 5.2% and 10.5% with ethanol and methanol respectively. BTE increased by 3.65% and 4.5% respectively with ethanol and methanol compared to gasoline. Overall, HC, CO, and NOx emissions decreased with ethanol and methanol at all CRs. Verma et al. [12] reported that a blend of ethanol (6.25% by vol.) and gasoline (93.75% by vol.) was better as compared to E20 and E80 blends. The performance and the emission characteristics of the engine were improved with the blend. Furthermore, Prasad et al. [13] reported that the engine with a blend of methanol and gasoline decreased the emissions and increased the engine's thermal efficiency by 25%. In addition to this, extensive research work on various blends of methanol and ethanol with gasoline has been studied [14-16]. Even though many studies are available on methanol and ethanol-gasoline blends, the studies pertaining to the use of 100% methanol (M100) and 100% ethanol (E100) in spark-ignition engines are scanty. Hence, a study is tried to compare specifically the behavior of two alcohols E100 and M100 as fuel in a spark ignition engine. The performance, emission, and combustion characteristics including mass fraction burnt and average cylinder temperature profiles of the engine were studied with E100 and M100.

2. Methodology and Experimental Details

A single-cylinder otto cycle engine having a cylinder capacity of 250 cm3 was used in this study. Table 1 indicates the specifications of the engine.

Table 1. Specifications of the engine

Engine	Single cylinder, four-stroke air cooled
Cylinder bore	74 mm
Stroke	58 mm
Compression ratio	9.8:1
Swept Volume	250 cc
Length of connecting rod	112 mm
Max. Torque	20 Nm @ 6000 rpm
Lubrication	Forced
Fuel injection	Port type
Dynamometer	Water-cooled eddy current dynamometer

The experimental setup used for this study is shown in Figure 1 which consists of a port fuel injection system and ECU for controlling the engine parameters**.**

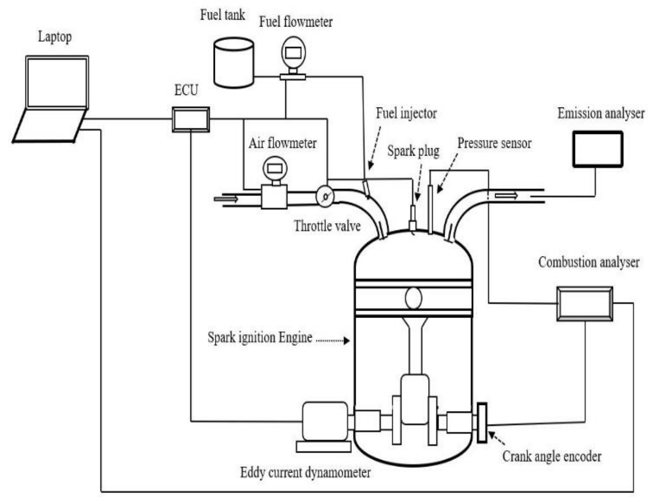

Fig. 1. Experimental setup

An eddy current water-cooled dynamometer was used for loading the engine. The AVL make exhaust gas emission analyser was used for measuring exhaust gas emissions. The engine was run at various speeds ranging from 3500-4200 rpm. The maximum torque obtained (7Nm) was at 3800 rpm. Therefore, combustion characteristics are described at this point. The In-cylinder pressure was measured using a piezoelectric pressure transducer mounted in the cylinder head. A crank angle encoder was used to acquire crank angle data. AVL Indicom V2.9 software was used to measure pressure and crank angle signals. An average of 100 cycles was taken for pressure-crank angle data. A Coriolis based mass flow meter was used to measure the fuel flow and air flow rate. The heat release rate was calculated from pressure crank angle data. It was further used to calculate cumulative heat release.
Heat release per degree crank angle (Q_θ) was calculated using Equation 1 [17].

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} P \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dP}{d\theta} \quad (1)$$

The cumulative heat release (Q_(cum.)) was calculated using Equation 2.

$$Q_{cum.}(\theta) = Q_{\theta} + Q_{\theta-1} \quad (2)$$

The inlet valve closure of the engine takes place at 1320 bTDC, and 1080 aTDC is the exhaust valve opening. Mass fraction burnt was calculated using cumulative heat release. The average cylinder gas temperature was calculated during the time period of closure of both valves. Exhaust valve closure was considered the reference point for temperature calculation (by assuming polytropic process) by Equation 3.

$$T_{cal.} = P_{cal.} V_{cal.} \frac{T_{ref.}}{P_{ref.} V_{ref.}} \quad (3)$$

3. Results and Discussion

Figure 2 indicates the In-cylinder peak pressure with E100 and M100. Peak pressure at all the speeds is higher with M100 than E100 by 1-2 bar. The laminar flame speed of methanol (52.3 cm/s) is higher than ethanol (39 cm/s). Flame development and propagation play a major role in combustion. Due to high flame speed, combustion becomes faster, resulting in a shorter combustion duration. The piston position is near to TDC and pressure increases to a high level. High In-cylinder pressure results in a high temperature. The peak pressure is more towards the TDC with M100 than that of E100.

Figures 3 and 4 show the pressure-crank angle and heat release rate (HRR) curves for E100 and M100 at 3800 rpm respectively. HRR revealed that for M100 the rate of mass burnt is rapid and thus, heat release is also rapid than E100. The combustion pathway for methanol involves fewer steps, and fewer intermediate products are produced. With E100, since carbon number is more than methanol, the combustion pathway is comparatively complex and more intermediate products are formed. Cumulative heat release (Fig.5) represents the heat generated inside the cylinder per cycle of the fuel inducted. The maximum heat release with E100 was higher than M100. The start of combustion was earlier with M100 since the slope formation in the M100 curve is earlier than E100. Early start of combustion could be triggered due to lower minimum ignition energy of M100 (0.14 mJ) which is 0.23 mJ in E100.

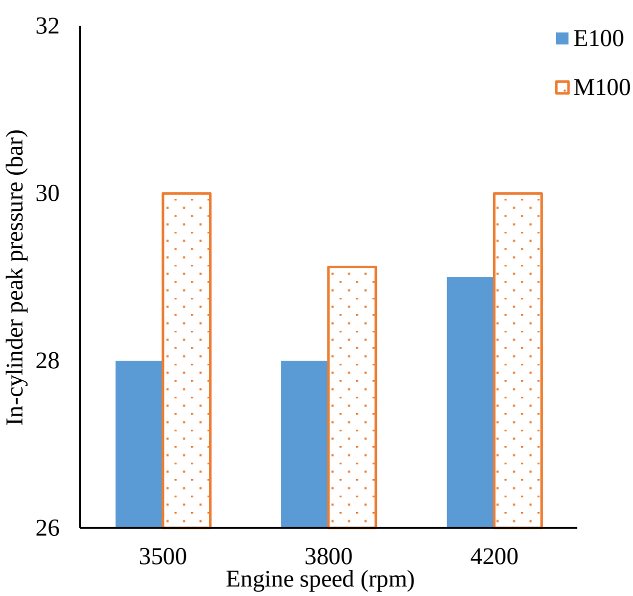

Fig.2. In-cylinder peak pressure at various engine speeds

It is clear from the profile of mass burnt fraction (Fig. 6) of both fuels. Crank angle for 10% and 50% mass fraction burnt occurred earlier with M100 as compared to E100. Mass fraction burnt profiles indicate the progression of combustion inside the engine. Combustion affects the in-cylinder pressure, temperature, thermal efficiency, and emissions from the engine. Shorter duration for mass burnt with M100 reflects better combustion with M100. The In-cylinder temperature profile is shown in Figure 7 at 3800 rpm. The average cylinder temperature with M100 was higher than E100.

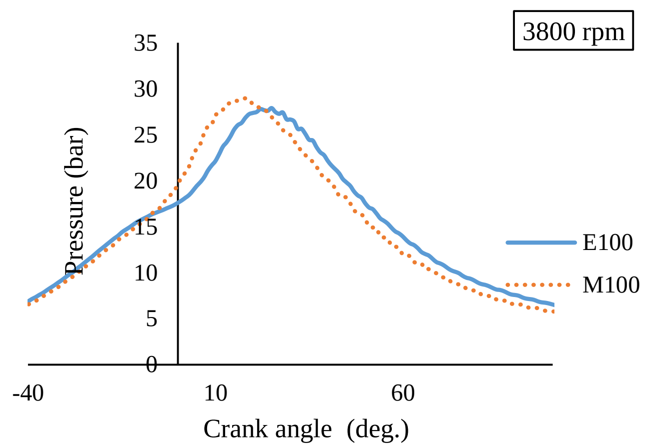

Fig. 3. Pressure crank angle curve for E100 and M100 at 3800 rpm

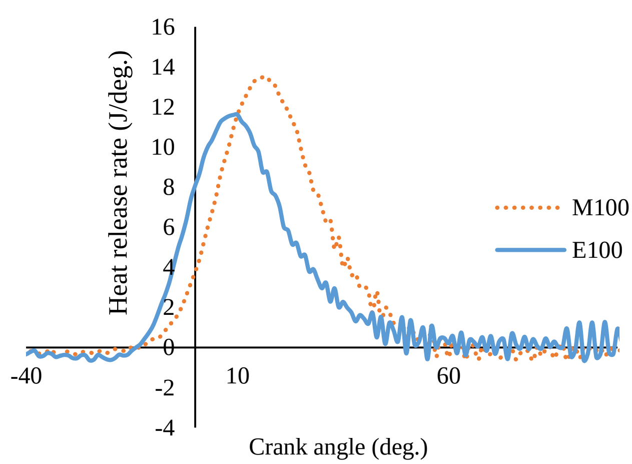

Fig.4. Heat release rate for E100 and M100

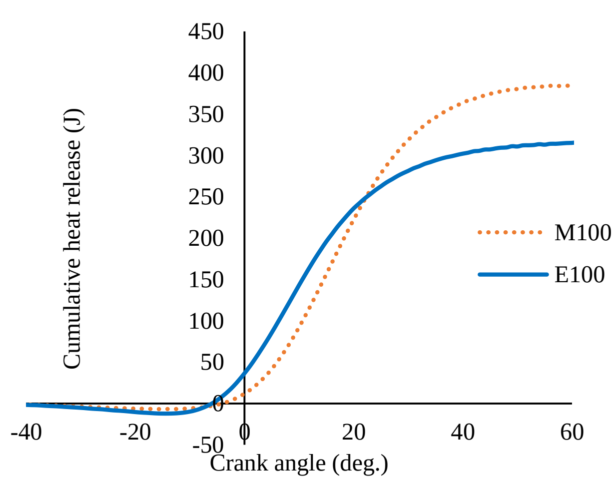

Fig. 5. Cumulative heat release curve for E100 and M100

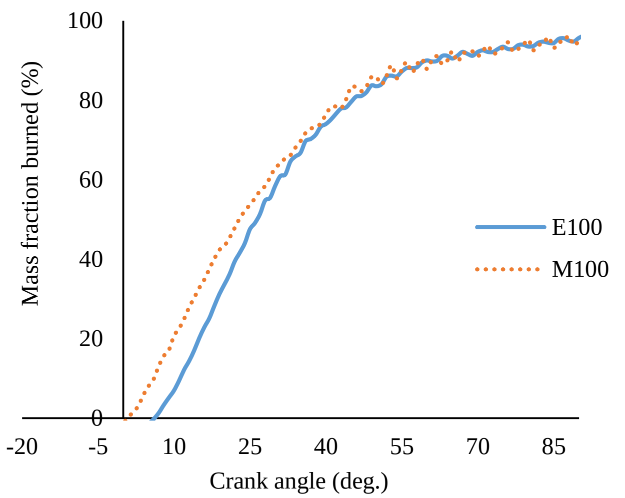

Fig. 6. Mass fraction burnt with crank angle in E100 and M100

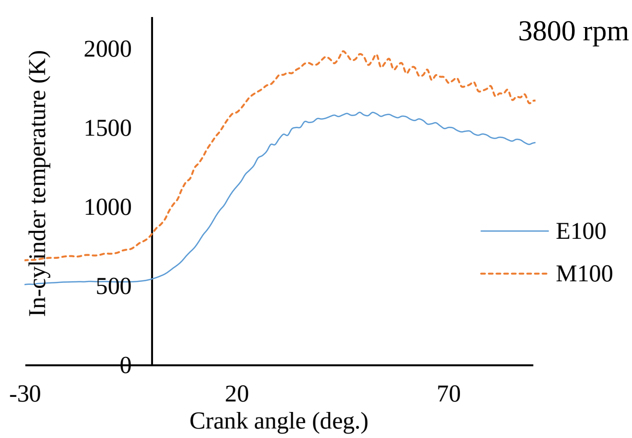

Fig. 7. In-cylinder temperature profile at 3800 rpm

Fig. 8 indicates the variation of maximum brake power of the engine with both the fuels. At 3800 rpm, Brake power obtained was maximum (7 Nm) with E100 and M100.

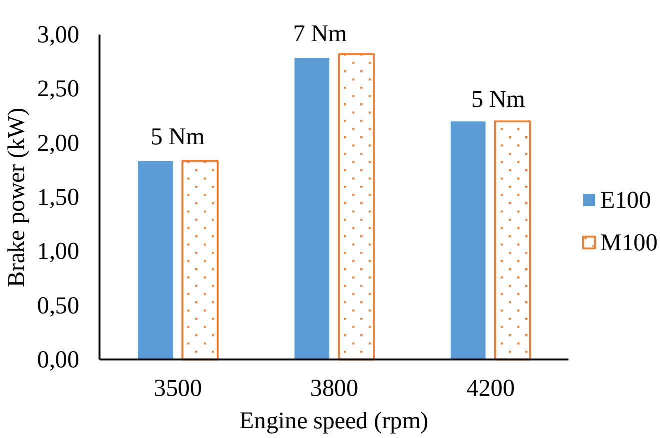

Fig. 8. Brake power with the engine speed

Fig. 9 indicates the brake thermal efficiency of the engine with E100 and M100. BTE with M100 was 39% higher than E100 at 4200 rpm. BTE was higher than E100 at all the speeds by more than 30%. Brake thermal efficiency is the ratio of output thermal energy to the input. Since the brake power was equal to both the fuels, input energy with M100 was relatively lower than E100. The calorific value of methanol is 20 MJ/kg, while that of ethanol is 26.8 MJ/kg. Fig. 10 shows the brake-specific fuel consumption (BSFC) with E100 and M100. It was observed that BSFC was nearby similar to both the fuels. Combustion was better with methanol than gasoline, as observed through the high brake thermal efficiency of the engine. It can be understood through the combustion kinetics of both the alcohols. Combustion products are simpler in methanol while during ethanol combustion, a variety of stable intermediate products are produced. As the number of species is increased, combustion irreversibility is enhanced. The reaction pathway of methanol combustion involves the formation of formaldehyde which yields formyl radical leading to the formation of CO and CO2 via the following reaction sequence (Equation 4) [18]

$$CH_2O \rightarrow HCO \rightarrow CO \rightarrow CO_2 \quad (4)$$

The pathway of the combustion reaction of ethanol involves more number radicals. C-C and C-O bond break at high temperatures, leading to the formation of various stable intermediate products such as ethylene, acetaldehyde, and formaldehyde (Equation 5).

$$\mathrm{C_2H_5OH} \rightarrow \mathrm{Ethylene} + \mathrm{Acetaldehyde} + \mathrm{Formaldehyde} \quad (5)$$

Conversion of acetaldehyde to various radicals takes place via different routes (Equation 6).

$$\mathrm{Acetaldehyde} \rightarrow \mathrm{Ethanol/carbon\ monoxide/\ ketene\quad via\ different\ routes} \quad (6)$$

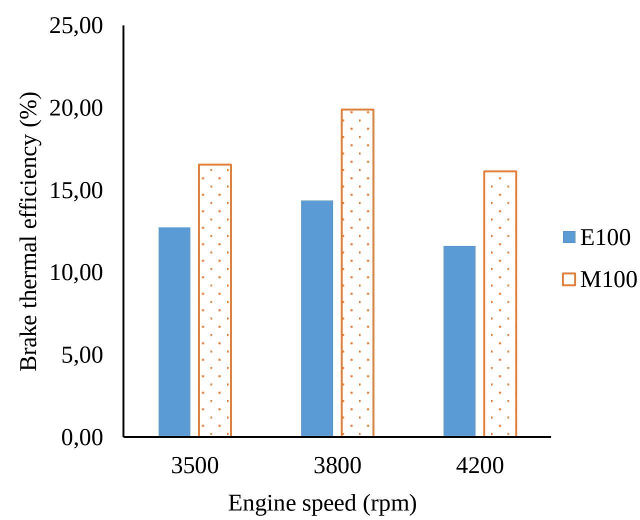

Fig. 9. Brake thermal efficiency variation with E100 and M100

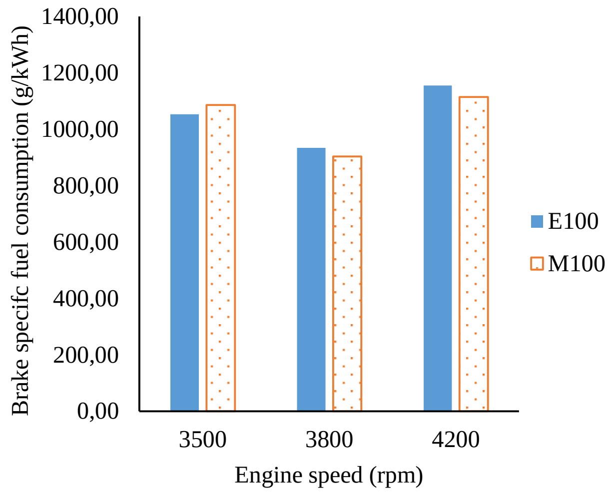

Fig. 10. BSFC with E100 and M100

Fig. 11 shows the volumetric efficiency of the engine at various speeds. The volumetric efficiency of both the fuels was very close at each speed. Since alcohols have high latent heat of vaporisation which increases the density of incoming air and more air could be inducted by the engine. The volumetric efficiency of the engine was lower due to the part throttle operation of the engine. Volumetric efficiency with E100 increased by 5% and 9% at 3800 rpm and 4200 rpm respectively. The stoichiometric air-fuel ratio of E100 (9:1) is higher than that of M100 (6.5:1). Therefore, the amount of air inducted with E100 is more significant than M100. Hence, the volumetric efficiency of the engine was slightly higher with E100.

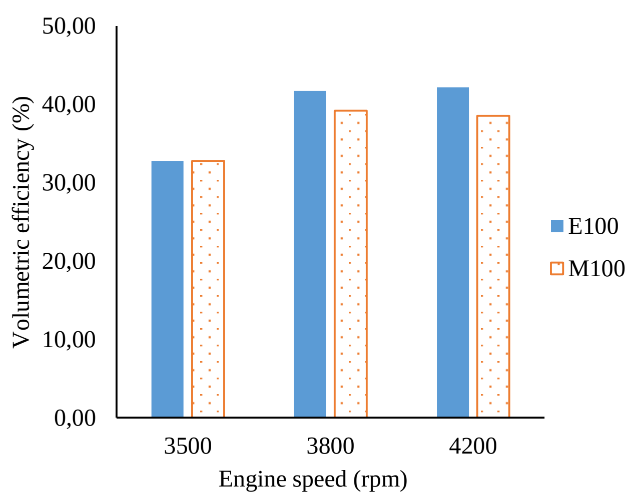

Fig.11. Volumetric efficiency of the engine with E100 and M100

Fig. 12 shows the CO emission with E100 and M100 at different speeds. It was observed that CO emission decreased by 32.9%, 52.9 %, and 7.3% with M100 as compared to E100. Methanol contains 50% (by mass) oxygen while it is present by 35% in E100. The presence of oxygen is helpful in the oxidation of CO to CO_2. Hence, CO emissions decreased with M100. The opposite trend was observed with CO_2 emission (Fig. 13). CO_2 emission was higher with M100. It was due to the efficient conversion of CO to CO_2 with M100.

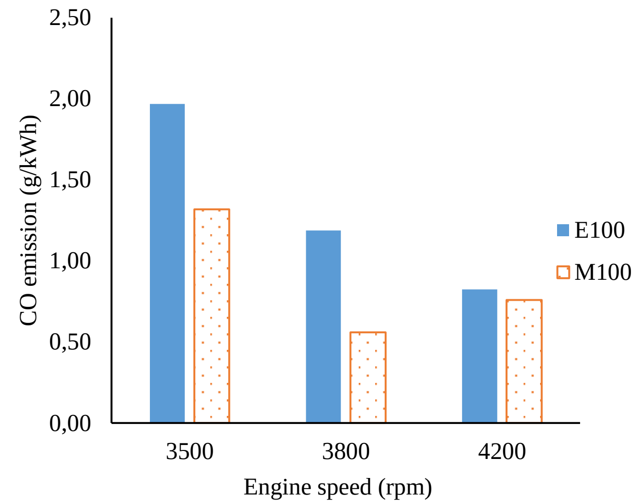

Fig. 12. CO emission with E100 and M100

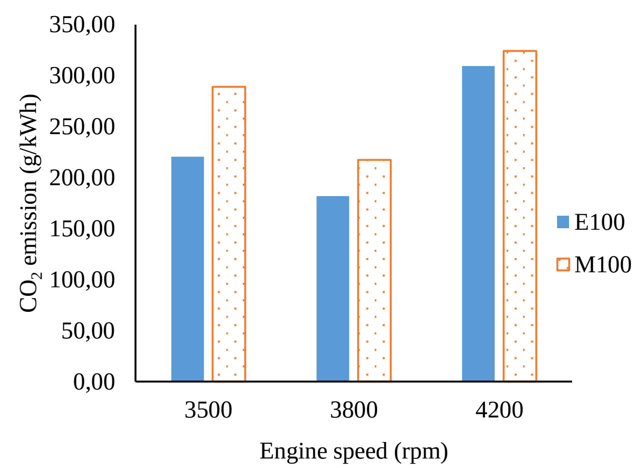

Fig. 13. CO_2 emission with E100 and M100

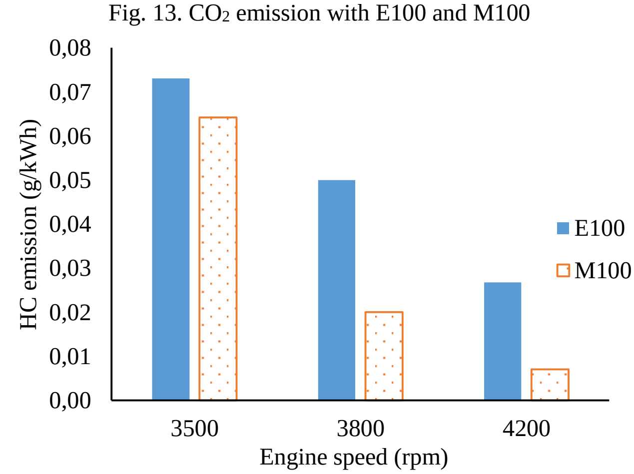

Fig. 14. HC emission with E100 and M100

Fig. 15 indicates the NOx emission with E100 and M100. It was observed that at the lowest speed, NOx emission with M100 increased by 5 times than E100 while at the highest speed, it increased by 60%. At middle speed, NOx emission with both the fuels was almost equal. The presence of oxygen, residence time, and temperature affect the NOx formation. According to the Zeldovich mechanism, cylinder temperature strongly influences NOx formation. The cylinder temperature was higher with M100 than E100 (Figure 5). Another factor was the presence of more oxygen (50% by mass) with M100 which could enhance the NOx formation.

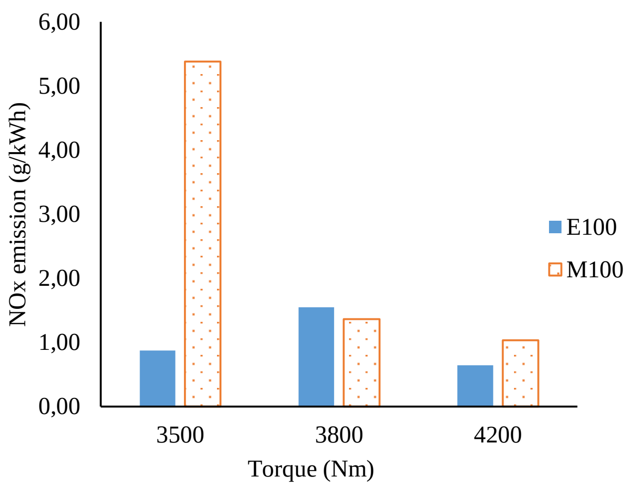

Fig. 15. NOx emission with E100 and M100

4. Conclusions

An experimental investigation was conducted to study an otto cycle engine's performance and emission characteristics fueled with methanol (M100) and ethanol (E100). The following outcomes are drawn from the study:

- The high flame velocity of methanol resulted in high In-cylinder peak pressure and temperature at all the speeds compared to E100.
- The start of combustion advanced with M100 compared to E100 due to its lower minimum ignition energy, which initiated the combustion earlier than E100.
- Crank angle for 10% and 50% mass fraction burnt occurred earlier with M100 than E100 due to the high flame velocity of methanol.
- Better combustion quality due to the production of simpler intermediate product species resulted in higher brake thermal efficiency of the engine with M100 than E100.
- The engine's volumetric efficiency increased marginally with E100 on account of its higher stoichiometric air-fuel ratio with E100 than that of M100.
- CO emission decreased by 32.9%, 52.9 %, and 7.3% at 3500 rpm, 3800 rpm, and 4200 rpm respectively with M100 compared to E100 due to the presence of more oxygen (by mass), which improved the oxidation of CO to CO2.
- HC emission with M100 decreased by 14.2, 60%, and 67% at 3500 rpm, 3800 rpm, and 4200 rpm respectively. The formation of simpler intermediate products, higher in-cylinder temperature, and presence of molecular oxygen contributed to decreased HC emission with M100.
- NOx emission with M100 increased by five times than E100 at 3500 rpm while at 4200 rpm, it increased by 60%. High cylinder temperature and presence of oxygen contributed to higher NOx emission with M100.

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Conflict of Interest Statement

The authors declare that there is no conflict of interest in the study.

CRediT Author Statement

Nidhi: Conceptualization, Writing-original draft, Validation, Data curation, Formal analysis
K. A. Subramanian: Conceptualization, Supervision

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