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Experimental Investigation of Combustion Characteristics of a Spark Ignition Engine Fueled with Methanol-Gasoline Blends (M15 and M85)

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Research Article

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Abstract

Combustion characteristics of a 2.8 kW, constant speed, spark ignition engine fuelled with methanol-gasoline blends (M15 and M85) were studied. It was observed that the maximum in-cylinder pressure and maximum heat release rate with M85 were almost like gasoline while these values were significantly lower with M15. The flame development duration was 30.5% and 5.5% shorter at lower and higher loads respectively with M85 as compared to gasoline. In addition to this, the flame propagation duration was shorter by 37% and 14.28% at a lower and higher load with M85. Similar results were obtained with M15 blend. The crank angle for fifty percent mass burnt fraction was near to the top dead center of the engine with both the blends as compared to gasoline. The methanol-gasoline fuelled spark ignition engine's important combustion characteristics including heat release rate, cumulative heat release, combustion duration, flame development, propagation, and laminar flame velocity are analysed in detail.

Keywords: Combustion; Flame development angle, Flame propagation angle, Gasoline, Heat release rate, Methanol, Spark ignition engine

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1. Introduction

The utilisation of conventional fuels for the transportation of people and goods is contributing to global warming and poor local air quality. Alcohols are alternatives to conventional fuels which help in mitigating the pollution level. Alcohols increase the octane number of the fuel and lower down the carbon monoxide (CO) and oxides of nitrogen (NOx) emissions [1]. They are more robust against the changes in fuelling in an internal combustion engine (IC) as compared to gasoline [2]. Methanol is one of the alcohols which can be a better substitute for gasoline in a spark ignition (SI) engine. It got attention during the 1980s and 1990s when it was demonstrated in fleet trials. The embedded oxygen in the molecule of methanol aids in the conversion of carbon monoxide to carbon dioxide [3]. It is reported that utilisation of water blended methanol with high compression ratio engine improved the fuel economy along with low NOx emission [4]. The adiabatic flame temperature of methanol is lower than gasoline. It ensures lower thermal losses and is thus beneficial in the improvement of the thermal efficiency of the engine. Properties of methanol and gasoline fuels are compared in Table 1. The higher flame velocity of methanol indicates that the engine can tolerate the high amount of exhaust gas recirculation. Furthermore, the high flame velocity ensures fast combustion and further reduces the chances of autoignition. A high autoignition temperature suppresses the onset of knock. Therefore, the thermal efficiency of a methanol fuelled engine can be improved by increasing the compression ratio of the engine. The high thermal efficiency of the engine could offset the effect of the lower calorific value of methanol as compared to gasoline. In addition to this, the storage of methanol is like gasoline. The cold starting issue with the methanol engine could be dealt with by using decomposed methanol i.e. a mixture of carbon monoxide and hydrogen for starting the engine [5]. All these features of methanol fortify its suitability as fuel for spark ignition engine.

1.1 Combustion chemistry of methanol

Alcohols contain a hydroxyl (OH) group attached to the hydrocarbon chain. The OH group is responsible for their unique reaction kinetics. It results in weak carbon to hydrogen bond strength where a carbon atom is bound and hydrogen bonding with OH and HO_2 radicals. Combustion reaction in alcohols involves the H-atom abstraction. At low and intermediate temperatures, OH and HO_2 radicals abstract the H-atom. Under high temperature and fuel-rich conditions, H radicals abstract the H- atom during combustion [7]. The primary consumption pathway for methanol involves the hydrogen abstraction by OH radicals during lean and stoichiometric conditions [8][9]. Lee et al. studied the thermal decomposition rate of methanol using reflected shock waves by atomic resonance absorption spectrometry of H atoms from 1660 to 2050 K [10]. Aranda et al. studied the oxidation mechanism of methanol-oxygen mixture diluted with nitrogen at high pressure (20 to 100 bar) and temperature between 600-900 K. They reported that increasing the pressure lowers the temperature for oxidation. The main mechanism of the reaction involves the abstraction of hydrogen by H and OH radicals [11].

Table 1. Properties of methanol and gasoline [3] [6]

Property	Unit	Methanol	Gasoline
Chemical formula	--	CH_3OH	Various
Density (STP)	kg/m^3	790	740
Oxygen content	% (by mass)	50	0
Research octane number	--	109	95
Heat of vaporization	kJ/kg	1100	180-350
Stoichiometric AFR	kg/kg	6.5	14.7
Lower heating value	MJ/kg	20	42.9
Flame speed (NTP and lambda = 1)	m/s	0.42	0.28
Adiabatic flame temperature	K	2143	2275
Minimum ignition energy in air	mJ	0.14	0.25
Quenching distance	mm	1.85	2.0
Autoignition temperature	K	738	465-743

1.2 Application of methanol in SI engine

Geng et al. [12] reported that the peak in-cylinder pressure and rate of heat release increased with the content of methanol in the blend M15 as compared to base gasoline. The duration of combustion decreased with the blends due to the high flame velocity of methanol. Zhao et al. [13] reported that CO and HC emissions were lower while NOx was higher with M15 blend as compared to gasoline. Zaid et al. [14] reported that the thermal efficiency of the SI engine improved with blends M3, M6, M12, and M15. Furthermore, Ozsezen [15] reported that with methanol/ethanol-gasoline blends, peak in-cylinder pressure and bmep of SI engine were higher than pure gasoline. Yanju et al. [16] reported that the in-cylinder pressure increased with the level of methanol in the blends: M10, M20, M85. CO and NOx emissions decreased while brake thermal efficiency improved with the increase of the percentage of methanol in the blends. Zhang et al. [17] reported that the flame development angle and flame propagation duration decreased with hydrogen addition (by vol. 1% and 2%) in a SI engine fuelled with methanol. Wang et al. [18] reported that in a gasoline direct injection engine fuelled with M15, M25, and M40 blends, CO, HC and NOx emissions were decreased. Similarly, Iliev [19] reported that CO, HC, and NOx emissions decreased with methanol-gasoline blends. Elfasakhani [20] concluded that methanol gasoline blends were better in terms of performance and emission characteristics as compared to ethanol-gasoline blends. Utilisation of Methanol in a diesel engine has also been reported [21], [22]. Literature analysis indicates that the performance and emission characteristics of SI engine fuelled with various methanol-gasoline blends has been studied extensively. However, an exclusive study on combustion characteristics of M15 and M85 is scanty in the literature. This study brings out the combustion related key features of gasoline-methanol blends and focuses to find out a better blend suitable for a spark ignition engine.

2. Experimental details and methodology

A four-stroke single-cylinder SI Genset engine was used for performing the experiment. An alternator with a rated power of 2.1 kVA at 50 Hz and 220 V was connected at the output shaft of the engine. The specifications of the engine are given in Table 2. The throttle opening was controlled by a mechanical governor connected to the crankshaft. The speed of the crankshaft was 3000 ± 50 rpm. The carburettor supplied the air-fuel mixture to the engine. Ignition timing was kept constant at 20 degrees before the top dead center. The engine is equipped with a transistorised coil type of ignition system. A piezoelectric pressure transducer was used to measure the in-cylinder pressure. The sensitivity of the pressure sensor is 45 pC/bar. The crank angle was measured by using an optical angle encoder with a crank angle resolution of 0.1 degrees. The blends were made on a volume basis. M15 is a blend consisting of 85% gasoline with 15% methanol by volume. Similarly, M85 blend was prepared while M100 is pure methanol fuel. Table 3 shows the sensitivities in various instruments used during the experiment. In-cylinder pressure data was acquired through AVL Indicom Mobile 2014 software. An average of 100 cycles was acquired for analysis. Figure 1 illustrates the experimental setup.

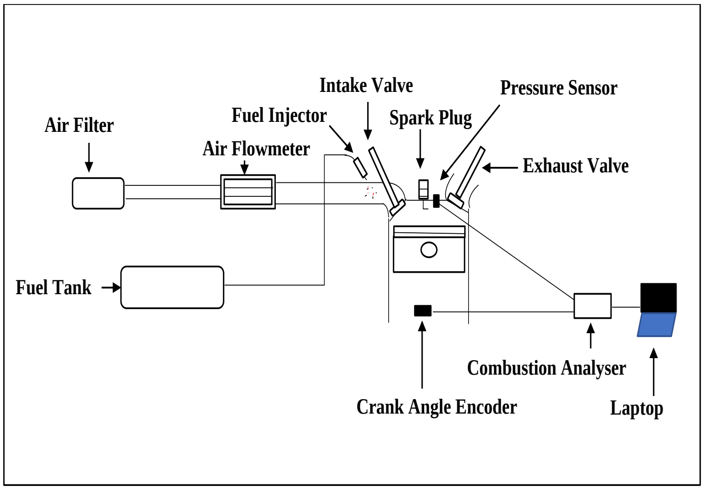

Fig. 1. Experimental test set up

Heat release (Q) per degree crank angle (C.A) was calculated using Equation 1[23]

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} P \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dP}{d\theta} + \frac{dQ_{ht}}{d\theta} \tag{1}$$

where,
P (Pascal) - In-cylinder pressure
V (m^3)- Instantaneous volume of the cylinder
γ - Ratio of specific heats C_p and C_v

Table 2. Specifications of the engine

Specification	Unit	Value
No. of cylinders	--	1
Bore x Stroke	mm x mm	76 x 60
Swept volume	cm^3	272
Compression ratio	--	7.2:1
Rated power	kW	2.8
Type of cooling system	--	Forced Air-cooled

Table 3. Sensitivity in various instruments

Parameter	Sensitivity of instruments
Fuel flow meter	0.01 g/s
Air flow meter	0.001 g/s
Pressure transducer	45 pC/bar
Optical encoder	0.1 ^{0} crank angle

Q_{ht} is the heat transfer through cylinder walls and was calculated using Equation 2

$$\frac{dQ_{ht}}{d\theta} = h(T - T_w)\frac{dA}{d\theta} \tag{2}$$

h (W/m^2K) - Convective heat transfer coefficient (From Woschni's correlation) [23]
A(m^2)- Instantaneous surface area of the chamber
$T_w(K) -$ Mean cylinder wall temperature
Volume per degree C.A was calculated using Equation 3 [23].

$$V = V_c \left\{1 + \frac{1}{2}(C.R - 1)\begin{bmatrix} R + 1 - Cos\theta \\ -(R^2 - sin^2\theta)^{1/2} \end{bmatrix}\right\} \tag{3}$$

where
R- Ratio of connecting rod length to crank radius
V_c - Clearance volume of the cylinder
Rate of pressure rise per degree C.A was calculated using Equation 4.

$$\frac{dP}{d\theta} = P_i - P_{i-1} \tag{4}$$

The cumulative heat release ($Q_{cum.}$) was calculated by using Equation 5.

$$Q_{cum.}(\theta) = \; Q_\theta + Q_{\theta-1} \tag{5}$$

The flame development angle (ignition delay) was calculated by Equation 6.

$$ID = \int_{Spark}^{SOC} d\theta \tag{6}$$

The flame propagation angle (combustion duration) was calculated by Equation 7.

$$CD = \int_{SOC}^{EOC} d\theta \tag{7}$$

where, SOC is the start of combustion considered as the C.A at which 5% heat is released while EOC is the end of combustion considered as the C.A at which 90% heat is released. The angles represent the duration of the event.
The laminar flame speeds for methanol-isooctane blends is calculated using Equation 8 [24]

$$S_{u,fva}^0 = \frac{1}{\left(\frac{1-f_{v,alc}}{s_{C8H18}^0}\right)+\left(\frac{f_{v,alc}}{s_{C8H18U,CxHyO}^0}\right)} \tag{8}$$

where, $f_{v,alc}$ is the volume fraction of alcohol in the blend.

3. Results and discussion

Fig. 2 indicates the laminar flame speed for methanol-isooctane blends. Methanol addition is the most effective to improve flame propagation with isooctane as compared to C_2 to C_5 alcohols [24]. At equivalence ratio (ER) 0.8, the laminar flame speed with M15 and M85 was 8% and 25.6% higher than isooctane. It was 3.1% and 20.2% higher than isooctane in the case of ER 1. These values are obtained at a temperature of 363 K and pressure 0.1 MPa and it gives a basic insight into the reason for the increase in flame speed of blend with methanol content. It can be inferred that the laminar burning velocity did not increase substantially with M15 as compared to M85.

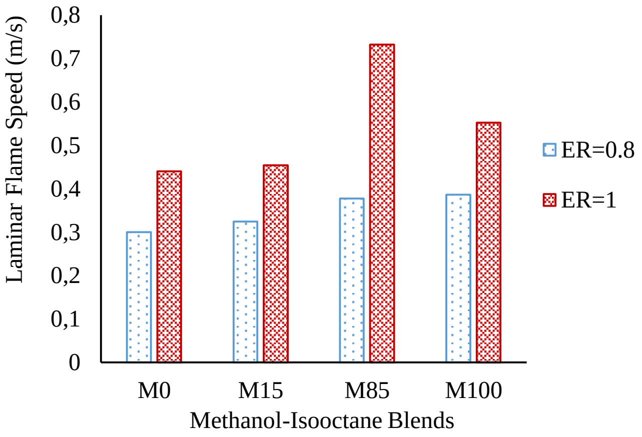

Fig. 2. Laminar flame speed for methanol-isooctane blends

Fig. 3 depicts the in-cylinder pressure with M15, M85, and gasoline. The maximum in-cylinder pressure with M15 is lower than gasoline by 10.6% while with M85 it is almost equal. The maximum in-cylinder pressure increased with the content of methanol in the blend and there is a slight shift of peak towards the top dead center. With the increase of methanol content in the blend, H and OH radicals increased leading to an increase in the reactivity and thus flame propagation [24]. Methanol with high flame speed burns the charge rapidly and therefore peak pressure was higher with M85 than M15.

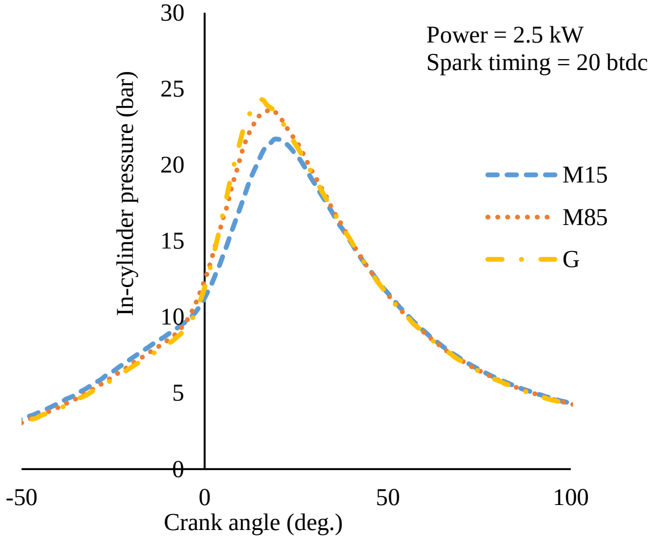

Fig. 3. Pressure-theta variation with different methanol-gasoline blends

Fig. 4 shows the log PV curve with M15, M85, and base gasoline. The degree of constant volume combustion with M85 and gasoline is equal while with M15 is the lowest. An increase in the flame speed with M85 decreased the time span for combustion leading to enhancement of constant volume combustion.

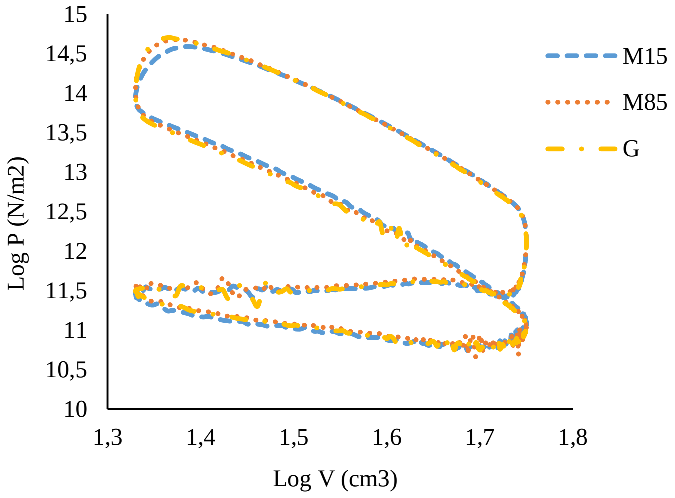

Fig. 4. Log PV curve

Fig. 5 shows the change of heat release rate with M15, M85, and gasoline. The value of the highest rate of heat release with M15 and M85 is lower than gasoline by 18.35% and 6.4% respectively. The heat release is an outcome of the combustion of fuel. As the quantity of methanol increased in the blends, the chemical reactivity increased due to the presence of more H and OH radicals as compared to base fuel. The carbon to hydrogen bond is weaker in methanol due to the presence of OH radical with it. The weaker bonds are easier to break when combustion is initiated. In addition to this, the intermediate species formed during methanol combustion are low molecular weight species than that with gasoline. These factors resulted in an increased heat release rate with methanol content in the blend.

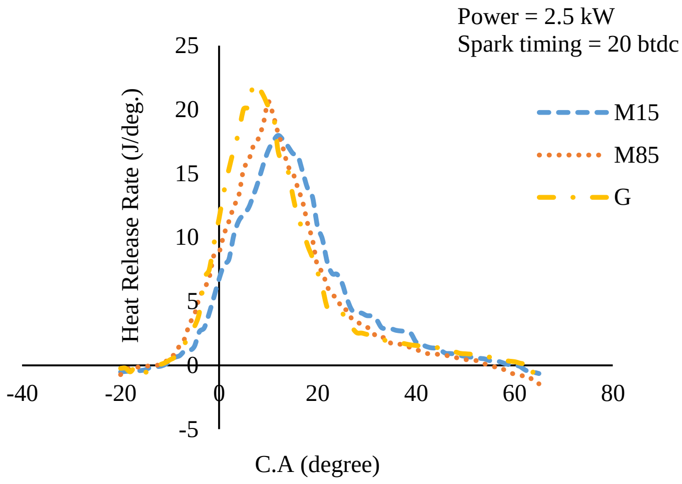

Fig. 5. Heat release rate with methanol-gasoline blends

Fig. 6 shows the change of cumulative heat release with M15 and M85 at a load of 2.5 kW. The maximum cumulative heat release with M15 and M85 is 1.5% and 0.18% respectively which are lower than gasoline. The fuel consumption increased with blends due to the lower calorific value of methanol and the total heat energy produced through combustion is almost equal to all the cases.

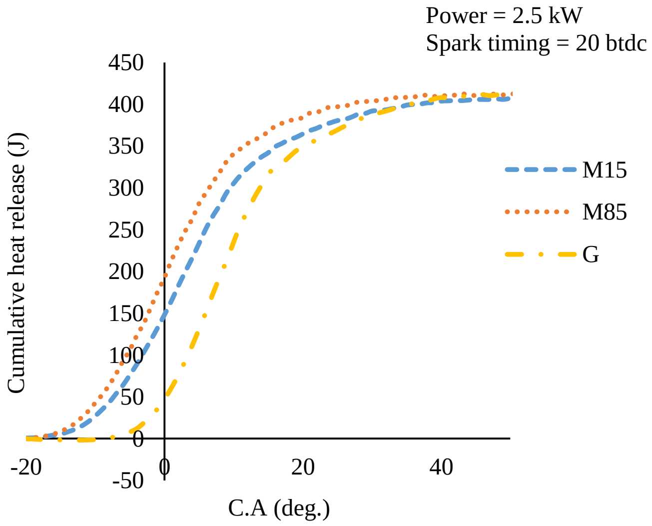

Fig. 6. Cumulative heat release with M15 and M85 blends

Fig. 7 depicts the variation of flame development angle with M15 and M85 at 0.5 kW and 2.5 kW. At 0.5 kW, the flame development angle with M15 and gasoline is the same while with M85 it is shorter than gasoline by 30.5%. At 2.5 kW, with M15, it is 11.1% more than gasoline while with M85 it is 5.5% shorter than gasoline. It was observed from the results that M15, is almost equal to or slightly more than gasoline. In the case of M85, the flame development period was slightly shorter than gasoline at both loads. At a higher load, more fuel is inducted inside the cylinder. The calorific value of methanol is lower than half of gasoline. Therefore, its consumption was more resulting in an increase in oxygen intake. The embedded oxygen in the methanol enhanced the combustion rate and the minimum ignition energy of methanol is lower (0.14 mJ) than gasoline (0.25 mJ) (refer Table 1). These factors were responsible for the initiation of combustion rapidly and thus causing a shorter flame development period at a higher load and with M85. However, in the case of M15, these factors might not be stronger on account of the lower methanol content in the blend.

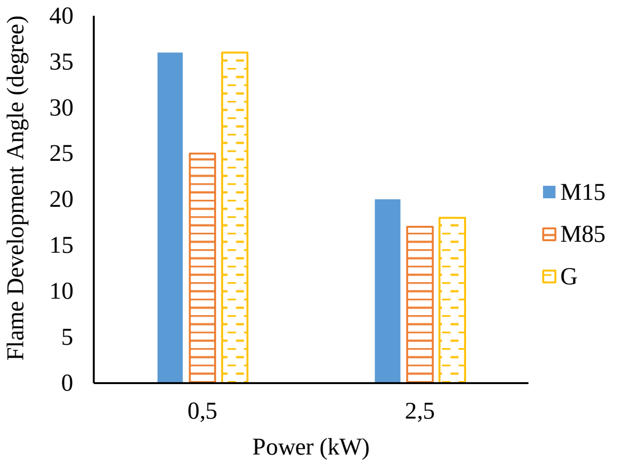

Fig. 7. Flame development angle with M15 and M85 blends at different load

Fig. 8 shows the variation of flame propagation angle at different loads with M15, M85, and gasoline. At 0.5 kW load, with M15 and M85, the flame propagation angle was 3% and 37% lower respectively than gasoline. At 2.5 kW, it was 3.5% and 14.28% lower with M15 and M85 respectively as compared to gasoline. It was observed that at higher loads, the duration of combustion was shorter than at lower loads. At higher load, the fuel consumption was more resulting in high in-cylinder temperature. High in-cylinder temperature accelerated the combustion reactions. The duration of flame propagation decreased with M15 and M85. With the increase in alcohol percentage, oxygen content inside the charge increased. More oxygen content of alcohols (50% by mass in methanol) increases the flame speed of their blends [24]. High flame speed decreased the flame propagation angle with M85. However, with M15, the propagation was marginally lower than gasoline as it is evident from its in-cylinder pressure- curve and laminar burning velocity.

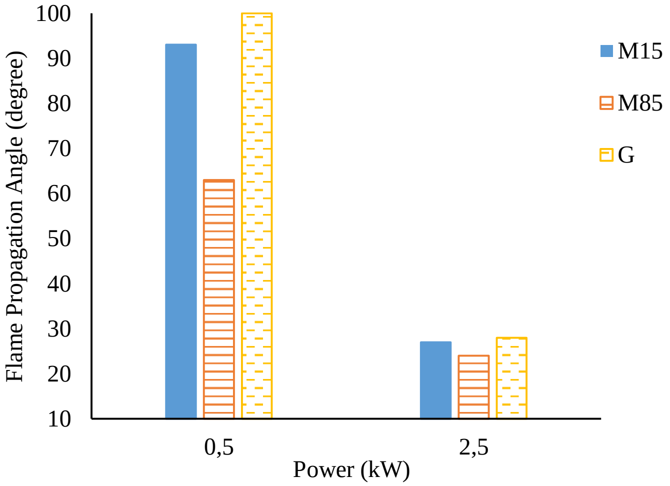

Fig. 8. Flame propagation angle with methanol-gasoline blends at different loads

Fig. 9 shows the (C.A.) crank angle for 50% mass burnt fraction. At 0.5 kW, with M15 and M85, it is 13.4% and 47.7% lower than gasoline. At 2.5 kW, with M15 and M85, it is 41.6% and 58.3% which are lower than gasoline. With the increase in methanol content in the blend, the C.A. 50% is shifted towards the top dead center. With all the fuel blends, the spark timing was constant. The fifty percent mass burnt fraction was near to TDC in the case of blends as compared to gasoline. It is observed from the figure that considerable (50%) heat energy is available in the engine in the early phase of expansion stroke with blends. It implies that conversion to useful work would be higher with blends. It would enhance the brake thermal energy of the engine with M15 and M85 as compared to gasoline. In the case of gasoline, the heat-to-work conversion would take place in the latter part of the expansion stroke where the in-cylinder pressure would decrease. Therefore, a decrease in the thermal efficiency of the engine could be expected.

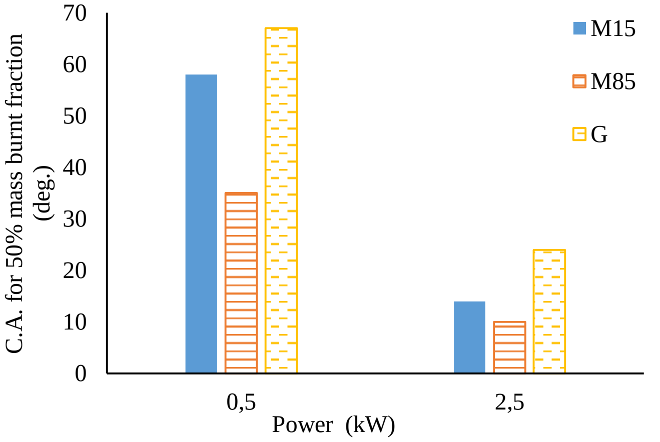

Fig. 9. C.A. for 50% mass burnt fraction with methanol-gasoline blends at different loads

4. Summary

M85 blend resembles gasoline in terms of various combustion characteristics. The peak in-cylinder pressure and maximum rate of heat release are almost equal in both fuels. The overall burning angle was shorter with M85 than gasoline. The added advantage with M85 is the release of fifty percent energy earlier than gasoline. It indicates that the thermal efficiency of an engine could be higher with it than M15. The laminar flame speed of M85 blends (methanol-isooctane) was higher than M15 and isooctane. Therefore, M85 could be regarded as a better substitute for gasoline in a SI engine.

5. Conclusions

The experimental tests were conducted on a spark ignition Genset engine to study the effect of methanol-gasoline blends M15 and M85 on combustion characteristics and compared with gasoline. The main conclusions that emerged from the study are given below.

- The maximum in-cylinder pressure with M15 was lower than gasoline by 10.6% while with M85 it is almost the same.
- The heat release rate increased with methanol content in the blend. The maximum rate was only 6.4% lower with M85 while with M15, it was 18.35% lower as compared to gasoline.
- At a lower load, the flame development angle with M15 and gasoline is the same. It was shorter with M85 than gasoline by 30.5%. At a higher load with M15, it was 11.1% more than gasoline. In the case of M85, it is 5.5% shorter than gasoline. The presence of more oxygen with M85 and lower minimum ignition energy are the main reasons for a short flame development angle.
- At lower load, with M15 and M85, the flame propagation angles were 3% and 37% shorter respectively than gasoline. At higher load, it was 3.5% and 14.28% shorter with M15 and M85 respectively as compared to gasoline. The high flame speed of methanol decreased the flame propagation angle with M85. With M15, the propagation duration was marginally shorter than gasoline.
- At lower load, with M15 and M85, C.A. for 50% mass burnt fraction was 13.4% and 47.7% less than gasoline. At a higher load, with M15 and M85, it was 41.6% and 58.3% less than gasoline. The fifty percent mass burnt fraction was near to TDC in the case of blends as compared to gasoline.
- The effect of methanol fraction in the methanol-isooctane blend was studied using a correlation. The laminar flame speed of M15 and M85 blends increased by 8% and 25.6% at an equivalence ratio of 0.8. It increased by 3.1% and 20.2% with M15 and M85 respectively at an equivalence ratio of 1.

Nomenclature

$Q_{cum.}$	Cumulative Heat Release
T_w	Mean Cylinder Wall Temperature (K)
γ	Ratio of C_p and C_v
$f_{v,\,alc}$	Volume Fraction of Alcohol in the Blend
$S^0_{u,fva}$	Laminar Flame Speed

Conflict of Interest Statement

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

CRediT Author Statement

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

References

[1] Defilippo A, Chin G T, Chen J Y. Development and Validation of Reaction Mechanisms for Alcohol-Blended Fuels for IC Engine Applications. Combust. Sci. Technol. 2013; 185 (8):1202–1226.

[2] Serras-Pereira J, Aleiferis P G, Richardson D. An Analysis of the Combustion Behavior of Ethanol, Butanol, İso-Octane, Gasoline, and Methane in a Direct-İnjection Spark-İgnition Research Engine. Combust. Sci. Technol. 2013; 185(3):484–513.

[3] Nidhi, Subramanian K A. Experimental investigation on Effects Of Oxygen Enriched Air on Performance, Combustion and Emission Characteristics of a Methanol Fuelled Spark İgnition Engine. Appl. Therm. Eng. 2019; 147:501–508.

[4] Rubin M B, McLeaN W J. Performance and NOx Emissions of Spark Ignited Combustion Engines using Alternative Fuels—Quasi One-Dimensional Modeling II. Methanol Fueled Engines. Combust. Sci. Technol. 1978; 18(5–6):199–206..

[5] Pettersson L, Sjöström K. Decomposed Methanol as a Fuel—A review, Combust. Sci. Technol.1991; 80(4–6): 265–303.

[6] Verhelst S, Turner J W, Sileghem L, Vancoillie J. Methanol as a Fuel for İnternal Combustion Engines. Prog. Energy Combust. Sci.2019; 70:43–88.

[7] Sarathy S M, Oßwald P, Hansen N, Höinghaus K Kohse. Alcohol Combustion Chemistry. Prog. Energy Combust. Sci.2014; 44:40–102.

[8] Dillon T J, Hölscher D, Sivakumaran V, Horowitz A, Crowley J N. Kinetics of the Reactions of HO with Methanol (210–351 K) and with Ethanol (216–368 K). Phys. Chem. Chem. Phys. 2005; 7(2):349–355.

[9] Jiménez E, Gilles M K, Ravishankara A R. Kinetics of the Reactions of the Hydroxyl Radical with CH3OH and C2H5OH between 235 and 360 K. J. Photochem. Photobiol. A Chem., 2003; 157 (2):237–245.

[10] Lee P-F, Matsui H, Xu D-W, Wang N- S. Thermal Decomposition and Oxidation of CH3OH. J. Phys. Chem. 2013; 117:525–534.

[11] Aranda V, Christensen J M, Alzueta M U, Glarborg P, Gersen S, Marshall P. Experimental and Kinetic Modeling Study of Methanol Ignition and Oxidation at High Pressure. Int. J. Chem. Kinet. 2013; 45:283–294.

[12] Geng P, Zhang H,Yang S. Experimental Investigation on the Combustion and Particulate Matter (PM) Emissions from a Port-Fuel Injection (PFI) Gasoline Engine Fueled with Methanol–Ultralow Sulfur Gasoline Blends. Fuel 2015; 145:221–227.

[13] Zhao H, Ge Y, Hao C, Han X, Fu M, Yu L, Shah A N. Carbonyl Compound Emissions from Passenger Cars Fueled with Methanol/Gasoline Blends, Sci. Total Environ. 2010; 408(17):3607–3613.

[14] Zaid M A, Badran O, Yamin. Effect of Methanol Addition on the Performance of Spark Ignition Engines. Energy & Fuels 2004; 18(2):312–315.

[15] Ozsezen A N. The Investigation of Thermodynamics and Combustion Properties of Alcohol-Gasoline Blends in an SI Engine Int. J. Green Energy. 2015; 12(11):1107–1112.

[16] Yanju W, Shenghua L, Hongsong L, Rui Y, Jie L, Ying W. Effects of Methanol/Gasoline Blends on a Spark Ignition Engine Performance and Emissions. Energy & Fuels 2008; 22(2):1254–1259.

[17] Zhang B, Ji C, Wang S, Zhou X. Idling Performance of a Hydrogen-Blended Methanol Engine at Lean Conditions. Energy Procedia 2014; 61:331–334.

[18] Wang, Xin & Ge, Yunshan & Liu, Linlin & Peng, Zihang & Hao, Lijun & Yin, Hang & Ding, Yan & Wang, Junfang, Evaluation on Toxic Reduction and Fuel Economy of a Gasoline Direct Injection-(GDI-) Powered Passenger Car Fueled with Methanol-Gasoline Blends with Various Substitution Ratios. Appl. Energy. 2015, vol. 157, pp. 134–143, 2015.

[19] Iliev S. A Comparison of Ethanol and Methanol Blending with Gasoline Using a 1-D Engine Model. Procedia Eng. 2015; 100:1013–1022.

[20] Elfasakhany A. Investigations on the Effects of Ethanol–Methanol–Gasoline Blends in a Spark-Ignition Engine: Performance and Emissions Analysis, Eng. Sci. Technol. an Int. J. 2015; 18(4):713–719.

[21] Prashant G K, Lata D B, Joshi P C. Investigations on the Effect of Methanol Blend on the Combustion Parameters of Dual Fuel Diesel Engine, Appl. Therm. Eng. 2016; 103:187–194..

[22] Soni D K, Gupta R. Numerical Investigation of Emission Reduction Techniques Applied on Methanol Blended Diesel Engine, Alexandria Eng. J. 2016; 55(2): 1867–1879.

[23] Heywood J B. Internal combustion engine fundamentals. New York McGraw-Hill;1988.

[24] Li Q, Jin Wu, Huang Z. Laminar Flame Characteristics of C1–C5 Primary Alcohol-Isooctane Blends at Elevated Temperature. Energies, MDPI, Open Access J. 2016; 9(7):1–17.