

Zero-Dimensional Single-Zone Combustion Modeling of an SI Engine Fueled with Alcohol-Gasoline Blends

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ABSTRACT

This study analysed and compared the combustion characteristics of baseline gasoline (M0) and methanol-gasoline blended with a 10% concentration (M10) in an SI engine using a zero-dimensional, single-zone combustion model. The experiments were conducted in a production 4-cylinder, 4-stroke spark-ignition engine with premixed stoichiometric combustion. In the current work, the engine operated at 2000 rpm and 75Nm of torque. The crank-angle-resolved pressure in the intake, exhaust, and in-cylinder was measured during all experiments. An in-house algorithm was developed in MATLAB to derive the rate of heat release (RoHR) and evaluate mass fraction burned (MFB) curves from the measured in-cylinder pressure data of the two fuels. The effect of various heat transfer correlations such as Annand, Hohenberg, and Woschni on the heat release rate was also studied to identify the most suitable model for the specific SI engine. The results indicated that the methanol-blended fuel (M10) provided a higher RoHR compared to conventional gasoline fuel (M0). The MFB profile curves revealed the combustion progress of the fuels as a function of crank angle. The Hohenberg heat transfer model was better suited for heat transfer to the wall of the engine compared to the other models, as it accorded the best fit in the compression phase. The addition of methanol to gasoline results in an increase in the combustion rate at CA10 (10% MFB) due to its higher flame speed propagation and octane rating.

Keywords: Zero-dimensional model; combustion; heat release rate; SI engine; alternative fuel

INTRODUCTION

The inception of internal combustion engines (ICEs) marked a significant revolution in engineering. Most internal combustion vehicles still rely on conventional fuels derived from fossil sources such as gasoline or diesel (Köten et al. 2020). Conventional fuels emit harmful substances like carbon dioxide (CO₂) and nitrous oxide (N₂O), which contribute to air pollution and global warming. The increasing demand for petroleum has led to concerns not only about the depletion of reserves but also about the environmental pollution caused by conventional fuels (Sarkar et al. 2012). As a result, optimizing internal

combustion engines (ICEs) has emerged as a primary research objective, with recent efforts aimed at improving engine efficiency and stability while minimizing emissions (Dabbaghi et al. 2021).

The primary criticisms of internal combustion engines (ICEs) relate to their environmental impact and the challenges they present. Various approaches have been explored to address these issues, including the pursuit of alternative fuels derived from renewable resources. The United Nations has mandated that countries reduce their CO₂ emissions, prompting researchers to explore biofuels such as ethanol and methanol (Sharudin et al. 2018). The practice of blending low concentrations of methanol and ethanol with conventional fuels is not new; it was used

during the oil supply shortages of the 1970s (Iliev, 2015). One of the appealing aspects of blending ethanol and methanol with conventional fuels is that it can be done without significant modifications to the engine structure (Chansauria & Mandloi 2018).

Methanol, the simplest alcohol with the chemical formula CH_3OH , can be derived from a range of renewable and waste resources, such as agricultural by-products, including fossil feedstocks like coal and natural gas, with natural gas being the main source (Li et al. 2017).

fermentation and distillation of biomass, including agricultural crops (Evidogan et al. 2010). Both methanol and ethanol exhibit substantially higher octane numbers and latent heat of vaporization compared to conventional fuels than conventional fuels, allowing for higher compression ratios and increased thermal efficiency, which is desirable for spark-ignited (SI) engines. This is because methanol and ethanol contain hydrogen, carbon, and oxygen, whereas conventional fuels primarily consist of hydrogen and carbon (Black 1991).

TABLE 1. Characteristics of baseline, methanol and ethanol fuels (Black 1991; Eyidogan et al. 2010)

Characteristics	Baseline (RON 95)	Methanol	Ethanol
Typical formula	C_8H_{18}	CH_3OH	$\text{C}_2\text{H}_5\text{OH}$
Research Octane Number (RON)	91-100	112	111
Heat of vaporisation (kJ/kg)	400	1,110	900
Stoichiometric A/F [9]	14.5	6.4	9
Heating value (kJ/kg) [8]	42.6	26.7	19.85

Table 1 presents that methanol and ethanol exhibit comparable Research Octane Numbers (RON) and possess elevated latent heat of vaporization. Additionally, they exhibit lower stoichiometric air-fuel ratios and heating values relative to the baseline fuel, although there are slight variations between the two. These characteristics are due to the fact that methanol and ethanol belong to the alcohol family, which includes compounds with higher oxygen content in their chemical structures, enhancing fuel performance. Consequently, these attributes are maintained when methanol and ethanol are blended with baseline fuel. Blending high-octane, oxygenated ethanol and methanol with conventional fuel is crucial as it results in an increase in the RON. Every 10% addition of alcohol fuel can raise the octane number by up to five units. This increase is attributed to their ability to reduce engine knocking, which is a result of their higher RON. Moreover, alcohol-blended fuels have been found to decrease CO_2 emissions. Research suggests that using alcohol fuels with higher RON values enables SI engines to run at higher compression ratios, contributing to greater combustion pressure and torque output.

The inclusion of oxygen in the fuel has been observed to affect ignition and early flame propagation, which in turn modifies the in-cylinder pressure profile (Shawal et al. 2016). The optical studies revealed that turbulent flame propagation was influenced by the ignition plasma and the formation of the early kernel during the initial phase of combustion (Shawal et al. 2020). The in-cylinder pressure can be analyzed thermodynamically using the rate of heat release (RoHR), derived from the first law of

thermodynamics (Heywood 1988). RoHR analysis is important because it establishes a correlation between pressure variations and the chemical energy released during combustion. The pressure-derived mass fraction burned (MBF) shows a good correlation with the optically determined flame speed obtained from high-speed imaging technique (Shawal et al. 2020). The RoHR, short for rate of heat release during combustion, quantifies the amount of chemical energy released when fuel is burned (Rosdi et al. 2022). RoHR analysis models can be categorized into single-zone and two-zone models. The single-zone model simplifies the analysis by treating the cylinder contents as a homogeneous mixture, whereas the two-zone model differentiates between burnt and unburnt gases. The RoHR integral provides the mass fraction burned (MFB), an S-shaped curve representing fuel consumption in relation to crank angle. It characterizes the spark ignition combustion process. Despite the simplicity of the single-zone model, the combustion analysis results are nearly identical to those of the two-zone model, as suggested by Cooney et al. (2008) in their study. They concluded that the MFB results from both models were approximately the same when accounting for heat transfer.

Numerous studies have examined the combustion characteristics of various alcohol-blended fuels in comparison to conventional baseline fuels in both spark ignition (SI) and diesel engines. Qi & Lee (2016) found that ethanol blends affect in-cylinder pressure and RoHR, with higher ethanol content (e.g., E20) resulting in higher peaks at higher loads, similar to the MFB due to increased laminar flame speed. Alexander & Porpatham (2019)

concluded that a 10% ethanol blend with LPG showed better RoHR and in-cylinder pressure compared to baseline fuels and other blends. Köten et al. (2020) found that ethanol-enriched gasoline led to higher cylinder pressure, RoHR, power, and torque. This is because ethanol has a lower stoichiometric ratio, allowing for greater fuel combustion. Chen et al. (2019) observed that methanol-gasoline blends exhibited elevated peak cylinder pressure, temperature, and RoHR with increased methanol content, attributing to faster flame development and propagation. Li et al. (2017) showed that alcohol-blended fuels promote advanced combustion phasing due to their higher laminar flame speed and reduced ignition delay, caused by their high latent heat of vaporization. Huang et al. (2020) observed that methanol-blended fuels in diesel engines increased in-cylinder pressure and RoHR, particularly at higher loads, with shorter ignition delays due to methanol's low autoignition temperature.

Numerous studies have examined the effects of varying alcohol-blended fuel concentrations on baseline fuels, including the impact of various heat transfer correlations on spark ignition engine performance. According to Heywood (1988), heat transfer is particularly crucial when temperatures peak towards the end of combustion, as it plays an essential role in the overall energy transfer and thermal management of the system, even though it constitutes only a small fraction of the fuel's energy. Heat transfer to the cylinder wall is key to the overall energy management system (Heywood 1988). Dabbaghi et al. (2021) investigated the impact of heat transfer correlations (Annand, Hohenberg, Woschni, and Sitkei's) on an SI crank-rocker engine. A study by Hairuddin et al. (2013) compared the effects of heat transfer methods on HCCI engine performance using Hohenberg, Woschni, and Modified Woschni models. In the study conducted Meena et al. (2022), the net heat release rate was used instead of the total RoHR for evaluating combustion characteristics. For simplicity, the influence of heat transfer was not considered in the net RoHR usage.

Therefore, the objective of this project is to analyze the combustion characteristics of baseline gasoline (M0) and a methanol-gasoline blend with a 10% concentration (M10) in a production spark-ignited engine using a zero-dimensional, single-zone combustion model, neglecting the effect of crevice volume. Three distinct models of heat loss to the cylinder wall (Woschni, Annand, and Hohenberg) were subjected to testing and comparison in accordance with the total rate of heat release and the performance during the compression phase. The developed algorithm visualizes in-cylinder pressure traces, combustion profiles of various fuels, and the heat release rate derivative over the crank angle for hundreds of combustion cycles.

METHODOLOGY

GEOMETRICAL PROPERTIES OF THE ENGINE

The experiments were conducted utilizing a production four-cylinder, four-stroke spark-ignition engine that operated with premixed stoichiometric combustion. In this study, the engine was operated at 2000 revolutions per minute (rpm) with a torque of 75 Newton-meters (Nm). To maintain this load, corresponding to 4.4 bar IMEP, under an intake pressure of 1000 mbar, the ECU adjusted the intake valve lift to a maximum of 2.2 mm. The ignition timing was automatically adjusted by the Engine Control Unit (ECU) at 34.5° CA bTDC. In all experiments, the crank-angle resolved pressure in the intake, exhaust, and in-cylinder were recorded. Table 2 shows the engine parameters used to determine the geometrical properties in the analysis.

CYLINDRICAL VOLUME, V

The cylinder volume at any crank angle, θ is expressed as:

$$V = V_c + \frac{\pi D^2}{4} (R + a - s) \quad (1)$$

TABLE 2. Engine parameters

Compression ratio, -	10
Connecting rod length to crank radius, mm	22.2
Stroke, mm	90
Crank radius, mm	45
Bore, mm	84
Speed, RPM	2000
Torque, Nm	75
Fuel	M0/M10
Intake valve close, IVC	-175.7°
Exhaust valve open, EVO	179.5°
Exhaust valve close, EVC	-404.5°
Intake pressure, mbar	1000

Where V_c represents clearance volume, V_d denotes the displacement volume, and s denotes the distance between the crank axis and the piston pin, defined as

$$s = a \cos \theta + (R^2 - a^2 \sin^2 \theta)^{1/2} \quad (2)$$

Substituting s into (1) yields

$$V = V_c * [1 + (\frac{r_c - 1}{2}) * (R + 1 - \cos \theta - (R^2 - \sin^2 \theta)^{1/2})] \quad (3)$$

TEMPERATURE MODEL

The temperature model used in the project was based on the ideal gas state equation, which is described in crank angles basis. The equation is stated below:

$$PV = mR_{gas}T \quad (4)$$

Where P represents the in-cylinder pressure measured by a piezoelectric transducer, V is the volume of the engine at any crank angle, m is the residual mass, R_{gas} is the gas constant, and T is the average gas temperature. Rearrange the equation into

$$T = \frac{PV}{mR_{gas}} \quad (5)$$

The instantaneous cylinder volume was pre-determined during the in-cylinder pressure data collection, where the volume was determined with a resolution of 0.5° crank angle.

POLYTROPIC INDEX, N

Polytropic index was used to describe the adiabatic process of A/F mixture during the compression cycle, where the temperature of the mixture increased without any heat lost the cylinder wall. The polytropic index, n of the A/F mixture, was defined based on the pressure-volume relationship during compression.

$$PV^n = \text{constant} \quad (6)$$

$$n = \text{constant} \left(\frac{P_1 V_1}{P_2 V_2} \right) \quad (7)$$

SPECIFIC HEAT RATIO, γ

The specific heat ratio, gamma was computed based on a linear model (Brunt et al. 1998). The γ model is modelled as a linear function of temperature (Kelvin).

$$\gamma(T) = 1.345 - (4.78 \times 10^{-5}) * T \quad (8)$$

HEAT TRANSFER MODEL

The heat transfer to the cylinder wall is governed by Newton's law of cooling or convective heat transfer, as it significantly influences in-cylinder heat transfer, which is provided as

$$Q_w = hA(T - T_w) \quad (9)$$

Where h is the heat transfer coefficient, T is the in-cylinder gas mixture temperature, T_w is the cylinder wall temperature, and A is the cylinder chamber surface area (Chansauria & Mandloi, 2018). Forced convection is a more practical heat transfer model because the external mechanical actions within the chamber generate fluid flows, whereas, radiation is only applicable to a few engine types (Heywood 1988).

Three models of convective heat transfer coefficient, h were applied in the analysis: Annand, Hohenberg, and Woschni (Hohenberg, 1979; Annand, 1963; Woschni, 1967).

WOSCHNI'S CORRELATION

Woschni correlation is based on the convection heat transfer, which is the basic effect derived from the laws of similarity for turbulent flow in terms of Reynolds number and correlates it with Nusselt number. The correlation is as follows (Woschni, 1967):

$$h_c = 3.26B^{-0.2} \times P^{0.8} \times w^{0.8} \times T^{-0.55} \quad (10)$$

Where B is the bore diameter of engine, p is the instantaneous pressure, w is the average cylinder gas velocity, and T is the instantaneous temperature. Woschni stated that $w(\theta)$ increases in the cylinder during combustion, compared to others, where it is constant. It is represented as below:

$$Q_w = hA(T - T_w) \quad (11)$$

$$p_m = p_r \left(\frac{V_r}{V(\theta)} \right) \quad (12)$$

Where C_1 and C_2 are constants and taken as 2.28 and 3.24×10^{-3} , for the combustion and compression respectively. $\bar{S}_p$ is the mean piston speed, V_d is the displacement volume, p_r is the reference gas pressure, V_r is the reference gas volume, p is the instantaneous in-cylinder pressure, P_m is the motored pressure, and V is the instantaneous volume.

HOHENBERG'S CORRELATION

Woschni correlation exhibits certain limitations as it tends to underestimate the compression process while overestimating the combustion process in relation to the heat transfer coefficient. This results in an excessive average heat flux throughout the entire cycle (Hairuddin et al. 2016). Hohenberg developed an alternative method to improve the accuracy of predicting time-averaged heat fluxes. This approach uses the length-based instantaneous cylinder volume and mean piston speed instead of the cylinder bore, and also changes the temperature term exponent. The changes in the heat transfer coefficient as shown below:

$$h_c = C_1 \times V_c^{-0.06} \times P^{0.8} \times T^{-0.4} (C_2 + \bar{S}_p)^{0.8} \quad (13)$$

C_1 and C_2 values are denoted with h values of 130 and 1.4, obtained using precise and detailed measurements of different engines' heat balance, heat flux, and component temperature. V_c is cylinder volume, p is the instantaneous pressure, T is the instantaneous temperature, and $\bar{S}_p$ is the mean piston speed.

ANNAND'S CORRELATION

Annand developed a heat transfer prediction model based on Newtonian principles, incorporating Nusselt and Reynolds numbers. It effectively decomposes the heat transfer coefficient into convective and radiative components. The correlation is expressed as follows:

$$h_g = a \times \frac{k_{gas} Re^{0.7}}{B} + c \cdot \sigma \frac{(T^4 - T_w^4)}{(T - T_w)} \quad (14)$$

Where

$$k_{gas} = 6.1944 \times 10^{-3} + 7.3814 \times 10^{-5} T - 1.2491 \times 10^{-8} \quad (15)$$

B is bore of the cylinder, ρ is the density of the fuel that is calculated using ideal gas law, μ is dynamic viscosity with an approximate value of 7×10^5 kg/ms, and $\bar{S}_p$ is the mean piston speed. k_{gas} refers to the gas thermal conductivity modelled using a polynomial curve-fitting of the measured data. The constant a varies within the range of 0.35 – 0.8 for standard operating conditions, where it relies on the intensity of charge motion. The value of c is equivalent to 0.075 for SI engines during combustion and expansion phase. Stefan-Boltzmann constant, σ is given as 5.67×10^{-8} W/m². K⁴. T is the instantaneous in-cylinder temperature and T_w is the wall temperature of the cylinder. All three models are tabulated in the Table 3.

RATE OF HEAT RELEASE (ROHR)

A zero-dimensional engine model treats the in-cylinder contents as a single zone and applies the first law of thermodynamics with the ideal gas equation of state. This approach incorporates the heat transfer effect to the cylinder wall (Yıldız & Çeper, 2017). The first law of thermodynamics states as follows:

$$\Delta U_{sys} = Q_{net} - W_{net} \quad (16)$$

$$Q_{net} = Q_b - Q_w \quad (17)$$

Where ΔU_{sys} is the change in internal energy, Q_{net} is the net heat transfer to the system, and W_{net} is the net work done by the system.

$$Q_b = \Delta U + Q_w + W_{net} \quad (18)$$

The derivation of the RoHR equation is rooted in the first law of thermodynamics within the combustion chamber, as depicted in Figure 1. ΔU and W_{net} , which are defined as:

$$\Delta U = mc_v dT \quad (19)$$

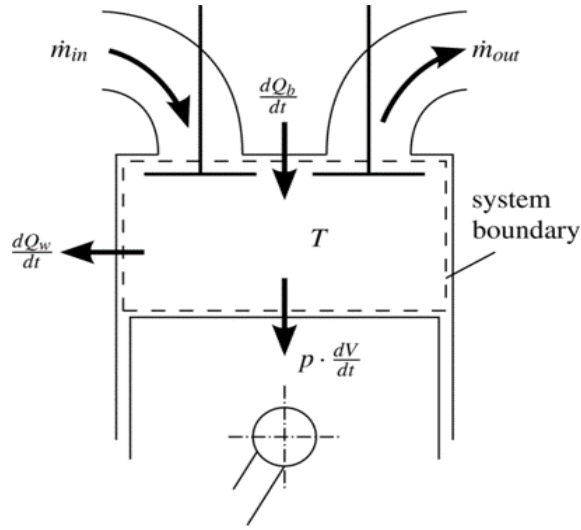


FIGURE 1. First law of thermodynamics assumptions (Heywood, 1988)

Substituting the thermodynamic expression into the first law of the open system:

$$Q_b = mC_v dT + Q_{ht} + PdV \quad (21)$$

$$dT = \frac{VdP}{mR} \quad (22)$$

TABLE 3. Heat transfer correlations

Woschni	$3.26B^{-0.2} \times P^{0.8} \times w^{0.8} \times T^{-0.55}$
Hohenberg	$C_1 \times V_c^{-0.06} \times P^{0.8} \times T^{-0.4} (C_2 + \bar{S}_p)^{0.8}$
Annand	$a \times \frac{k_{gas} Re^b}{B} + c \cdot \sigma \frac{(T^4 - T_w^4)}{T - T_w}$

$$W_{net} = PdV \quad (20)$$

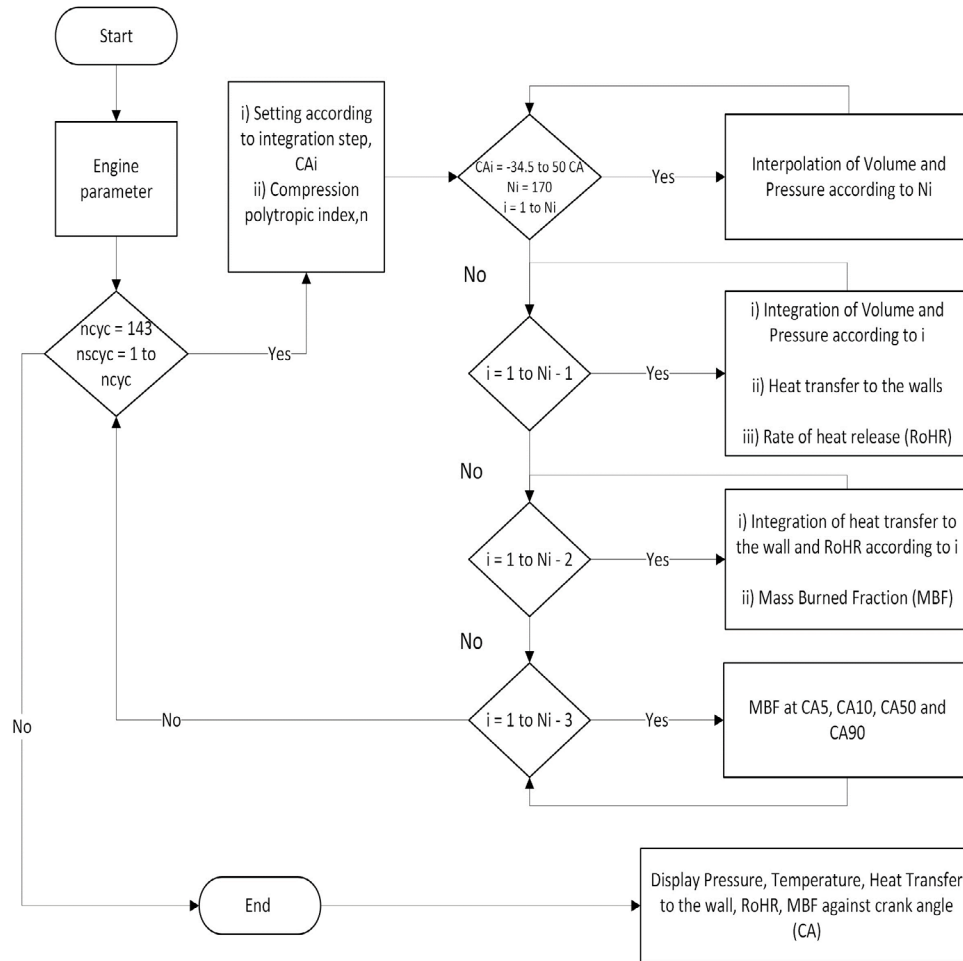


FIGURE 2. Algorithm flow chart of thermodynamic analysis

$$dT = \frac{VdP}{mR} \quad (23)$$

Substituting Eq. 22 and Eq. 23 into Eq.21 yield:

$$dT = \frac{VdP}{mR} \quad (24)$$

Where Q_w is the heat transfer to the cylinder wall. Specific gas constant equation, Eq.25 and specific heat ratio, γ , equation, Eq.26 were substituted into Eq.24. Thus, the derivation of the RoHR is shown as Eq.27.

$$C_p - C_v = R \quad (25)$$

$$\frac{C_p}{C_v} = \gamma \quad (26)$$

$$Q_b = \frac{1}{\gamma - 1} V dP + \frac{\gamma}{\gamma - 1} P dV + Q_w \quad (27)$$

Eq.27 was differentiated with respect to time (crank angle), as shown in Eq.28.

$$\frac{dQ_b}{d\theta} = \frac{1}{\gamma - 1} V \frac{dP}{d\theta} + \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{dQ_w}{d\theta} \quad (28)$$

MASS FRACTION BURNED (MFB)

The mass fraction burned (MFB) quantifies the proportion of energy released as a during each combustion cycle, exhibiting an S-shaped pattern. (Cooney et al. 2008; Heywood, 1988) The combustion of a fuel-air mixture exhibits a distinctive pattern characterized by an initial surge in combustion rate following the ignition of the mixture, followed by a peak during the midpoint of the combustion process. Subsequently, the combustion rate descends to near-zero levels upon the complete combustion of the fuel. The utilization of mass fraction burned or energy-release fraction curves is a viable approach to visually represent the temporal characteristics of different phases of the in spark-ignition engines combustion process. It enables the precise determination of the engine cycle duration occupied by each phase.

The acquisition of MFB involves calculating the cumulative heat release by integrating the RoHR curve. RoHR integration is normalised by dividing with the value of $(m_f \times Q_{LHV})$.

$$CHR = \int_{\theta = IV0}^{\theta = IVc} (RoHR) d\theta \quad (29)$$

$$MFB = \frac{CHR}{m_f Q_{LHV}} \quad (30)$$

ALGORITHM PROCEDURES

The flowchart of the numerical algorithm is presented in Figure 2. The process started with the engine parameters defined, including the pressure and volume derived from the engine data. The simulation computed over 143 consecutive combustion cycles. The heat release integration period was calculated during -34.5 to 50 degrees at top dead centre (aTDC) with an increment of 0.5 degree.

RESULTS AND DISCUSSION

IN-CYLINDER PRESSURE

Figures 3 and 4 present the in-cylinder pressure in relation to the crank angle for both the M0 and M10 fuels. The black lines denote the pressure data for each fuel over 143 cycles. Figure 5 presents a comparative analysis of the averaged pressure at top dead center (aTDC) for the two fuels. The results show that M10 fuel reached a higher average peak pressure at a lower crank angle than the M0 fuel. Based on the results, variations in in-cylinder pressure were not distinctly observable at the start of ignition (SOI), approximately -30° to -10° aTDC, but became more dispersed during the combustion process.

The M0 fuel generated a peak average pressure of 3200 kPa at 7.5° aTDC, whereas the M10 fuel produced a peak average pressure of 3300 kPa at 5.5° aTDC. The in-cylinder pressure for the M10 fuel peaked 100 kPa higher and occurred 2° earlier aTDC than the M0 fuel. This phenomenon can be attributed to the characteristics of alcohol, which possesses a higher-octane number, greater latent heat of vaporization, and a higher laminar flame speed compared to pure gasoline (Paluri & Patel 2022).

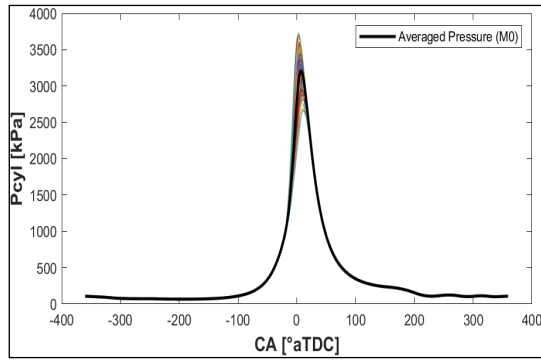


FIGURE 3. In-cylinder pressure traces of M0 fuel over 143 cycles

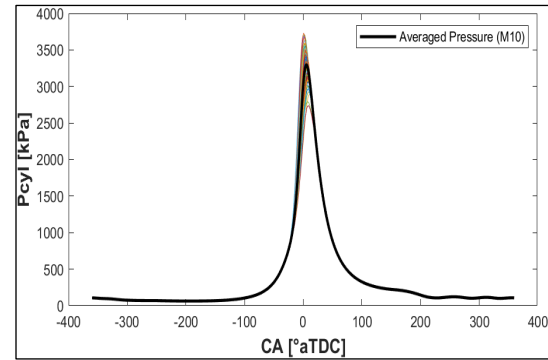


FIGURE 4. In-cylinder pressure traces of M10 fuel over 143 cycles

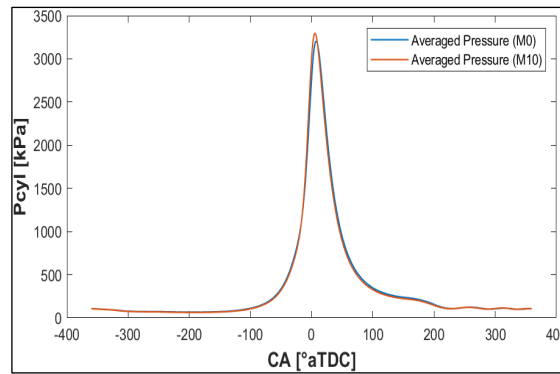


FIGURE 5. Averaged of in-cylinder pressure trace for both fuel over all cycles

The high laminar flame speed of methanol promotes faster flame propagation, and its high latent heat of vaporization cools the air-fuel mixture, leading to more efficient combustion. Furthermore, the air-fuel ratio influences in-cylinder pressure. Methanol, with its lower stoichiometric air-fuel ratio due to its chemical structure, contains oxygen. Consequently, it requires less air for complete combustion. (Rosdi et al. 2022; Black 1991).

To assess the relative variability or dispersion of the data, the peak pressure data from all cycles were statistically analyzed employing the coefficient of variation (CoV) methodology. The CoV is determined by calculating the ratio of the standard deviation to the mean of the peak pressure data set. This is illustrated below:

$$CoV = \frac{\sigma \text{ of peak pressure}}{\text{mean of peak pressure}} \times 100\% \quad (31)$$

The analysis results showed that the peak pressure data for the baseline fuel, M0, displayed greater variability or fluctuation compared to the M10 fuel, with CoV values of 7.58% and 5.49%, respectively. These findings indicate that the M10 fuel exhibited more consistent and precise peak pressure during the combustion process compared to the M0 fuel, aligning with the results obtained by Alexander and Porpatham (2019). This consistency can be attributed

to the higher oxygen content in alcohol fuels. Methanol, with its faster laminar flame speed and higher-octane number, provides more efficient combustion (Alexander & Porpatham 2019; Chansauria & Mandloi 2018). The development and propagation of the flame pattern significantly impact combustion variations. Thus, higher combustion efficiency reduces the peak in-cylinder pressure variability.

HEAT TRANSFER TO THE WALL

Figures 6 and 7 illustrate the heat transfer to the cylinder wall for both M0 and M10 fuels using the heat transfer correlations developed by Woschni et al. 1967. According to Figure 8, Hohenberg's correlation recorded the highest heat transfer to the wall for both the M0 and M10 fuels, followed by Woschni's and Annand's correlations. The M10 fuel also reached peak heat transfer significantly earlier than the M0 fuel. As shown in Table 4, the heat transfer for M10 fuel peaked at 4.0°, 4.5°, and 4.5° after top dead center (aTDC) according to Hohenberg's, Woschni's, and Annand's correlations, respectively. In contrast, the heat transfer for the M0 fuel peaked at 5.5° aTDC across all correlations. The difference in peak heat transfer between the two fuels was approximately 1 to 1.5° aTDC, meaning that M10 fuel achieved peak heat transfer 28% faster than the M0 fuel.

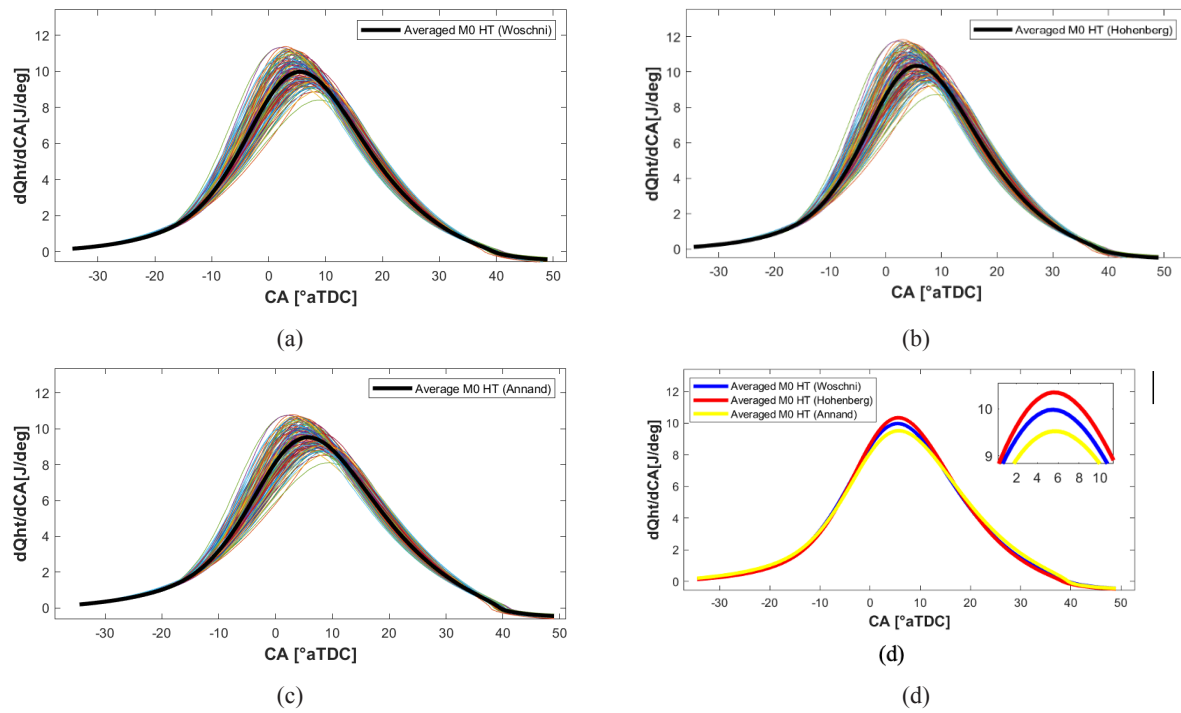


FIGURE 6. RON95 (M0) heat transfer to the wall using (a) Woschni; (b) Hohenberg; (c) Annand and; (d) average of the three correlations

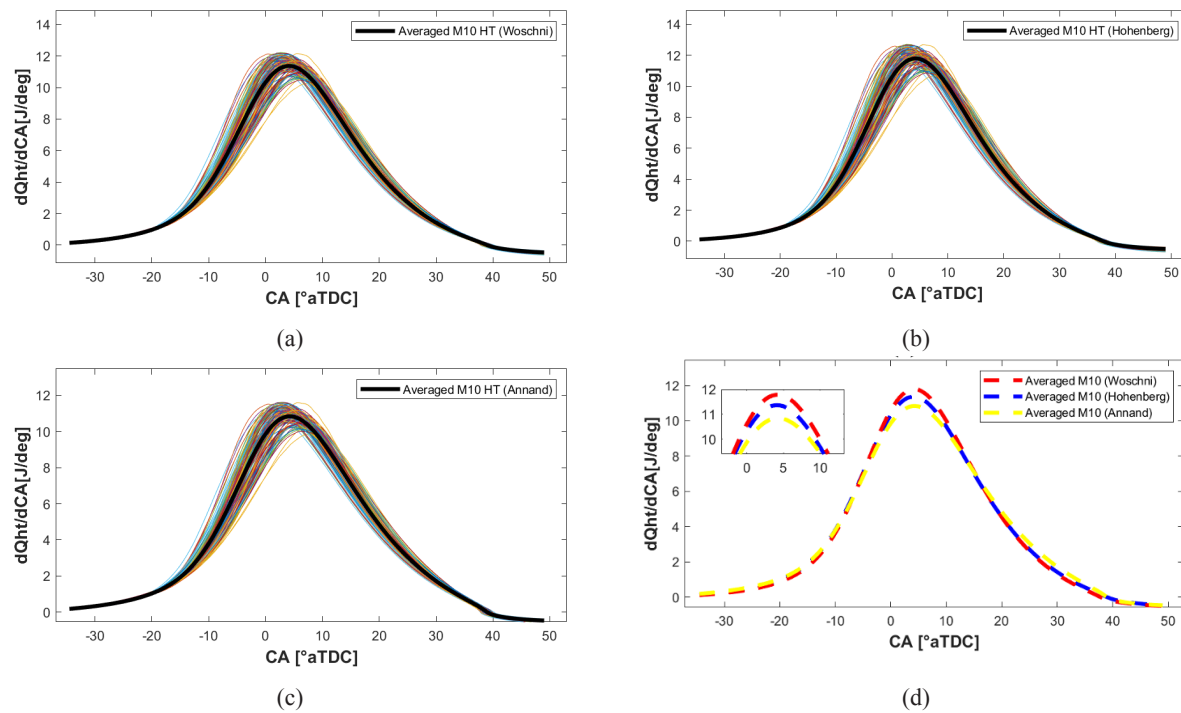


FIGURE 7. Methanol-blended fuel (M10) heat transfer to the wall using (a) Woschni; (b) Hohenberg, (c) Annand; and (d) average of the three correlations over 143 cycles

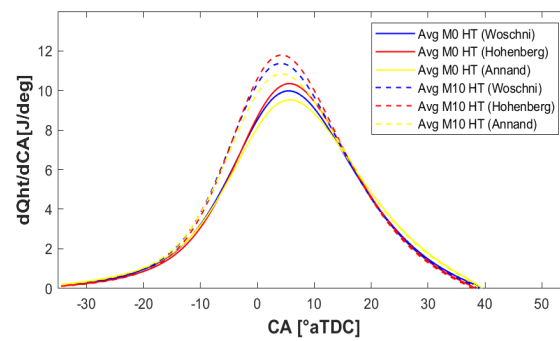


FIGURE 8. Averaged in-cylinder pressure for both fuel over one complete cycle

TABLE 4. Heat transfer correlations

	Baseline (M0)		Methanol-blended (M10)	
	dQ_{HT} (J/deg)	$^{\circ}aTDC$ (deg)	dQ_{HT} (J/deg)	$^{\circ}aTDC$ (deg)
Hohenberg	10.35	5.5	11.79	4.0
Woschni	9.98	5.5	11.37	4.5
Annand	9.52	5.5	10.83	4.5

As per Soyhan et al. (2009), several parameters influence the magnitude of heat transfer coefficients, including length scale, pressure, velocity, and gas temperature. They determined that while temperature only affects the magnitude of correlations, the length scale impacts both the magnitude and the shape of correlations. The influence of the length scale is evident in Hohenberg's modifications to Woschni's correlation. Woschni's correlation initially exhibited inaccuracies that resulted in an underestimation of heat transfer coefficients during the compression process and an overestimation during the combustion process. Hohenberg replaced the bore with a length based on the instantaneous cylinder volume, adjusted for changes in effective gas velocity, and modified the exponent in the temperature term.

Additionally, the heat transfer rate results demonstrated a difference between the two fuels, with M10 fuel showing higher rates. This can be attributed to the fact that each correlation is influenced by pressure, temperature, or both, and M10 fuel improved both variables due to its unique characteristics. Previous research has demonstrated that alcohol-blended fuels, such as methanol, possess a higher oxygen content relative to pure gasoline (RON 95) (Eyidogan et al. 2010; Sharma et al. 2019; Chen et al. 2019; Huang et al. 2020; Li et al. 2017). This elevated oxygen content results in elevated pressure and temperature, which subsequently influences the heat transfer correlations.

RATE OF HEAT RELEASE (ROHR)

Figures 9 and 10 illustrate the rate of heat release (RoHR) for both M0 and M10 fuels over 143 cycles. The figures indicate that Hohenberg's correlation provided the highest RoHR for both fuels among the correlations used. For the M10 fuel, the highest heat release recorded was 33.40 J/degree at -2.5° after top dead center (aTDC) using Hohenberg's correlation. This was followed by 33.84 J/degree at -2.5° aTDC using Woschni's correlation, and 33.11 J/degree at -2.5° aTDC using Annand's correlation. For the M0, the highest RoHR recorded was 31.88 J/degree at -0.5° aTDC using Hohenberg's correlation. This was followed by 31.74 J/degree at -0.5° aTDC using Woschni's correlation, and 30.84 J/degree at -0.5° aTDC using Annand's correlation. The RoHR more closely represents the chemical energy release when considering heat transfer, with the difference being due to combustion inefficiencies. Hairuddin et al. (2013) made a similar observation, noting that a high rate of heat transfer to the wall results in an increased heat release rate. They noted that both Hohenberg's and Woschni's correlations assume that the characteristic velocity remains consistent throughout the strokes, which results in a significant amount of heat being transferred to the wall, thereby increasing the RoHR.

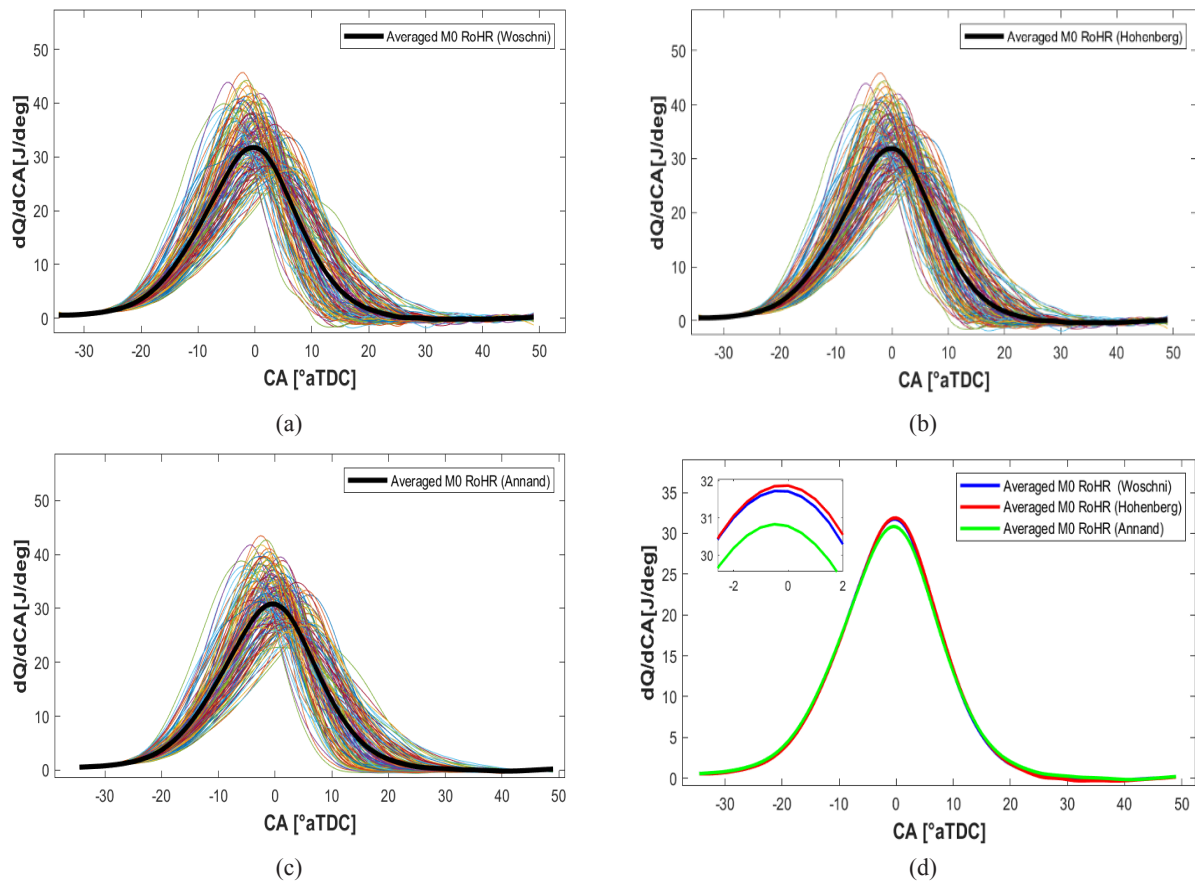


FIGURE 9. RON95 (M0) RoHR using (a) Woschni; (b) Hohenberg; (c) Annand; and (d) average of the three correlations

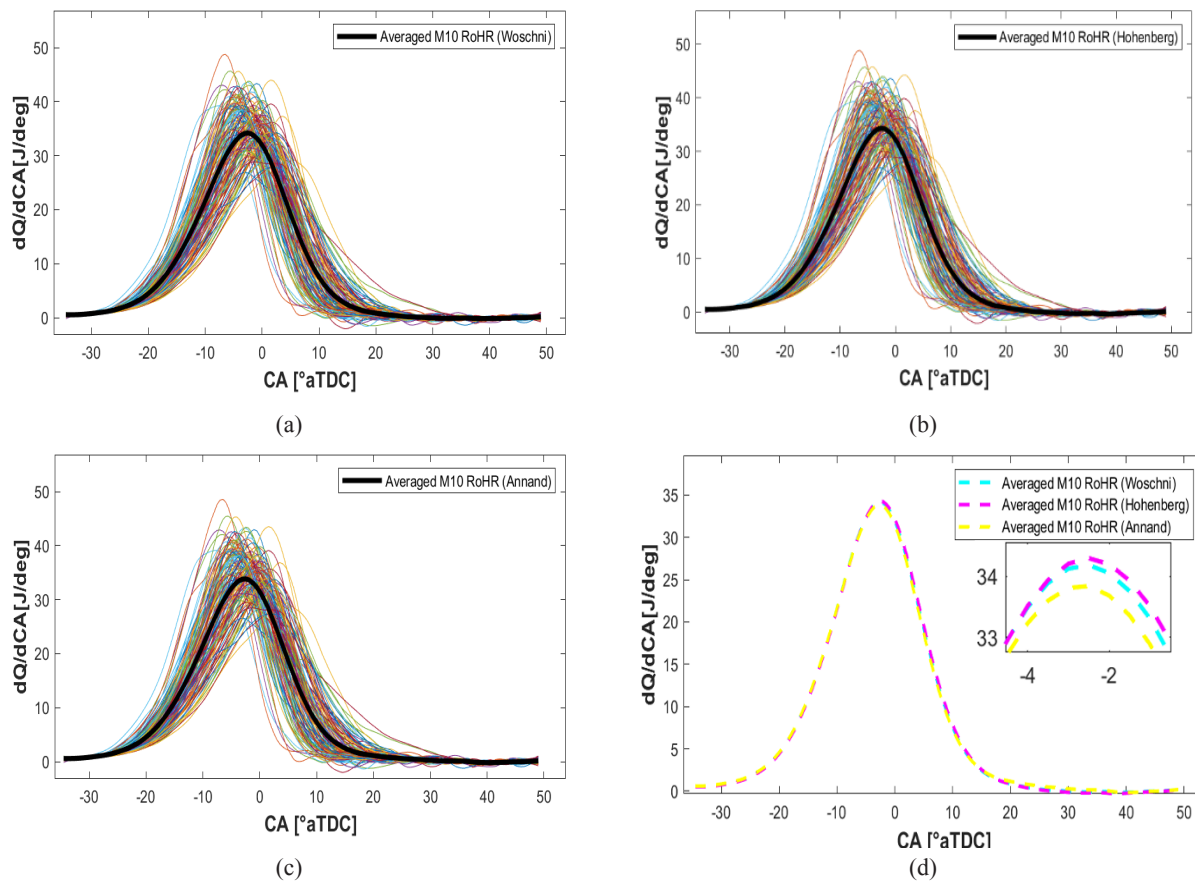


FIGURE 10. Methanol-blended fuel (M10) RoHR using (a) Woschni; (b) Hohenberg; (c) Annand; and (d) average of the three correlations over 143 cycles

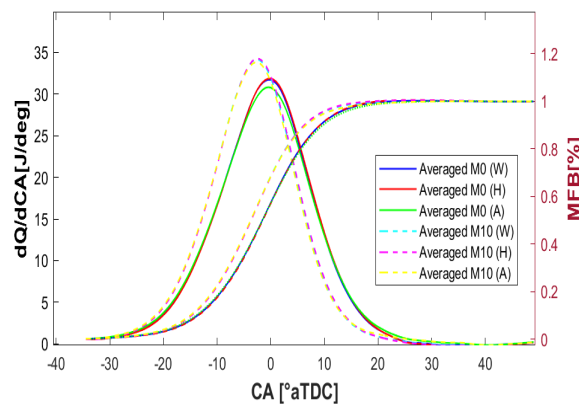


FIGURE 11. Comparison of average RoHR and MFB between two fuels across all correlations

Figure 11 demonstrates that the rate of heat release (RoHR) peak occurred slightly earlier for the methanol-blended fuel. This is evidenced by the shift of its curves to the left, indicating that the peak occurred before the piston reached top dead center (TDC). In contrast, the RoHR peak for the M0 fuel occurred after TDC. According to the data tabulated in Table 5, M10 fuel exhibits a higher RoHR than the M0 fuel. The RoHR for M10 fuel shifted 2.0° aTDC earlier than that of the M0 fuel for all correlations. When comparing the M0 fuel with the M10 fuel, it was observed

that the latter reached its RoHR peak approximately 80% sooner. These findings suggest that the properties of the fuel

and the mechanisms of heat transfer significantly influenced the results. Alcohol fuels exhibit a higher laminar flame speed, which accelerates the combustion process during the initial phase of combustion. Additionally, presence of hydroxyl groups in methanol molecule weakens the molecular bonds of the M0 fuel, leading to a lower temperature at which oxidation occurs. This accelerated oxidation process enhances combustion.

TABLE 5. Rate of heat release (RoHR) for both fuels with different correlations

	Baseline (M0)		Methanol-blended (M10)	
	dQ_{ch} (J/deg)	$^\circ$ aTDC (deg)	dQ_{ch} (J/deg)	$^\circ$ aTDC (deg)
Hohenberg	31.88	-0.5	34.40	-2.5
Woschni	31.74	-0.5	33.84	-2.5
Annand	30.84	-0.5	33.11	-2.5

MASS FRACTION BURNED

Figures 12 and 13 present the mass fraction burned (MFB) for both the M0 and M10 fuels over 143 cycles. The average MFB for each fuel is depicted in black to emphasize the quantity of fuel consumed during combustion. Figure 14 compares the average MFB of both fuels across all correlations over the 143 cycles. The MFB curve effectively represents the flame burning progress of each fuel. Figure 14 illustrates the fuel burn progression for both M0 and M10 fuels, with the latter shifting to the left, consistent with the previously discussed rate of heat release (RoHR) trend. Table 6 presents the mass fraction burned (MFB) determined utilizing the Hohenberg correlation at CA50 for the M0 and M10 fuels. The MFB values are 2.5° aTDC for the M0 fuel and -2.0° aTDC for the M10 fuel. This indicates that the MFB for M10 fuel advanced compared to the M0 fuel. In the early combustion phase, M10 fuel

showed faster progression, with a slight advancement of about 0.5 to 1.0° aTDC. This advancement became more apparent as the combustion process continued, especially at 50% and 90% MFB, where the M10 fuel combusted significantly faster by 4.0 degrees and around 7.0 degrees, respectively.

The elevated latent heat of vaporization and laminar flame speed of methanol in the blended fuel contribute to shorter ignition delays. The elevated latent heat of vaporization facilitates an increase in charge density by more effectively cooling the charge (Sharma et al., 2019). Furthermore, the acceleration of the burn rate is facilitated by the enhanced laminar flame speed of the M10 fuel, particularly during the 50% – 90% MFB phase. The increased oxygen content in alcohol-blended fuels improves the laminar flame speed and promotes more consistent combustion compared to the M0 fuel.

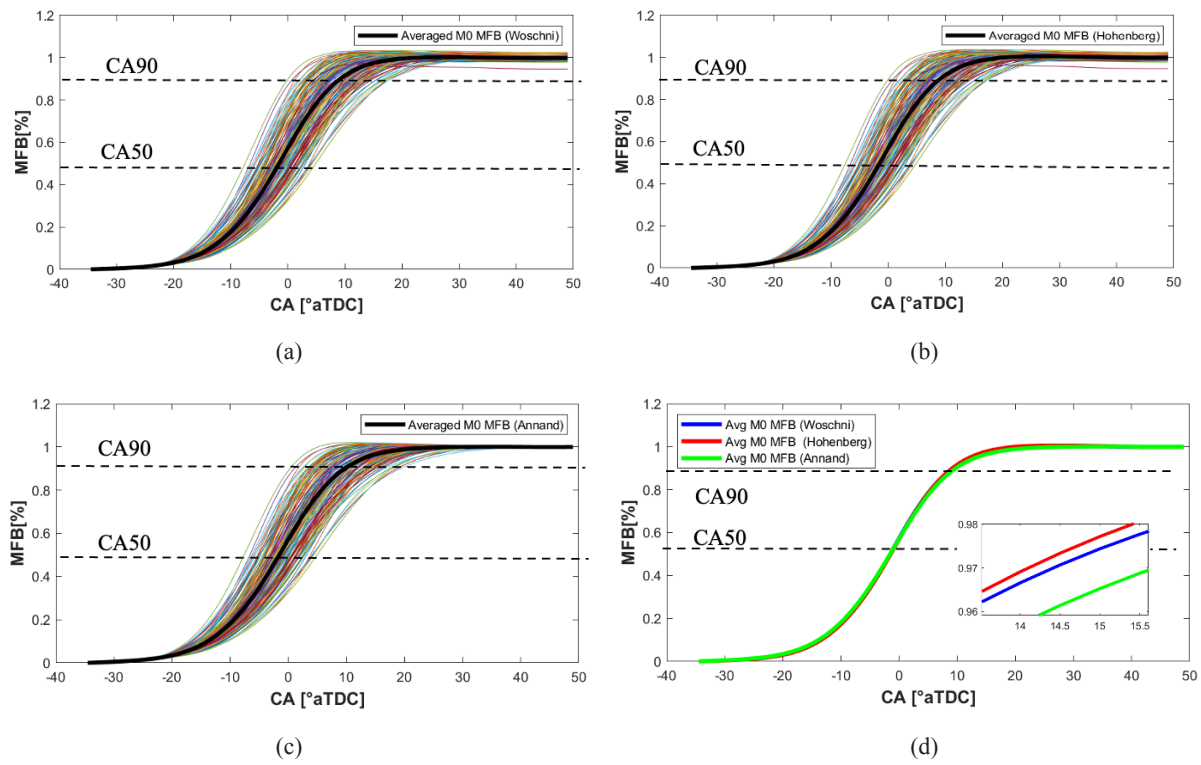


FIGURE 12. RON95 fuel (M0) MFB using (a) Woschni; (b) Hohenberg; (c) Annand and (d) average of the three correlations

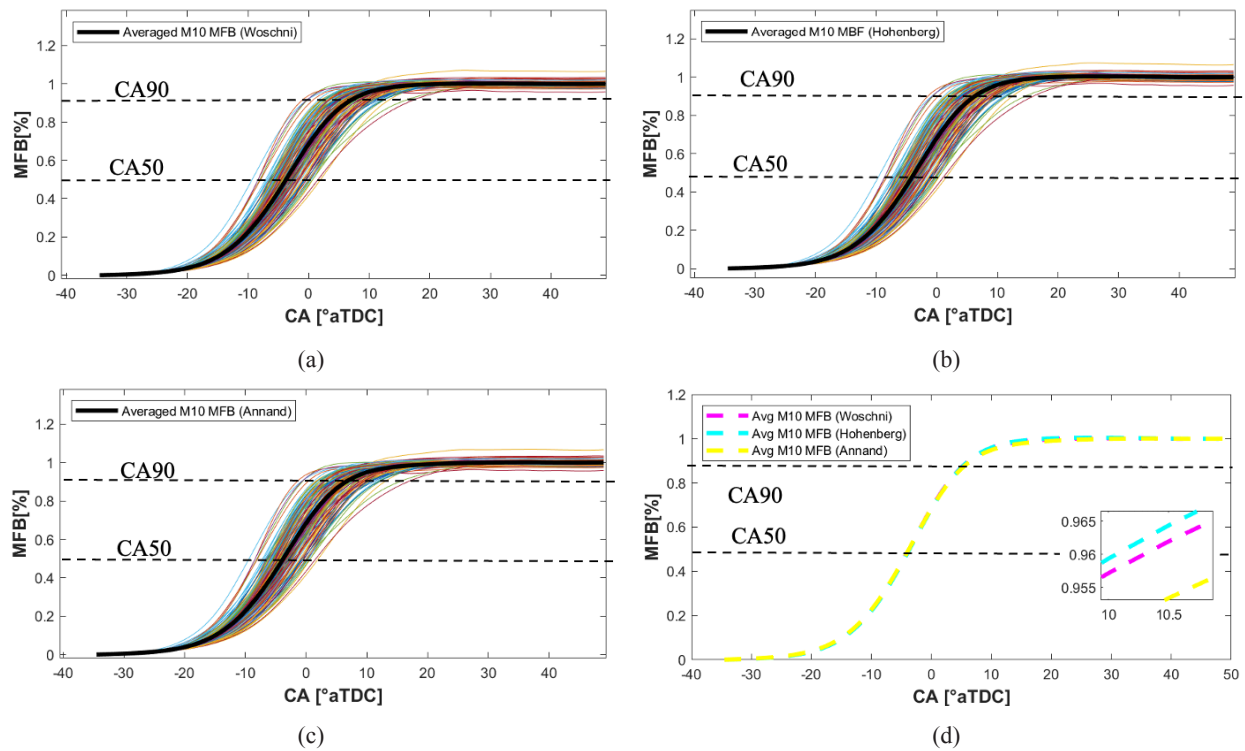


FIGURE 13. Methanol-blended fuel (M10) MFB using (a) Woschni; (b) Hohenberg; (c) Annand and; (d) average of the three correlations over 143 cycles

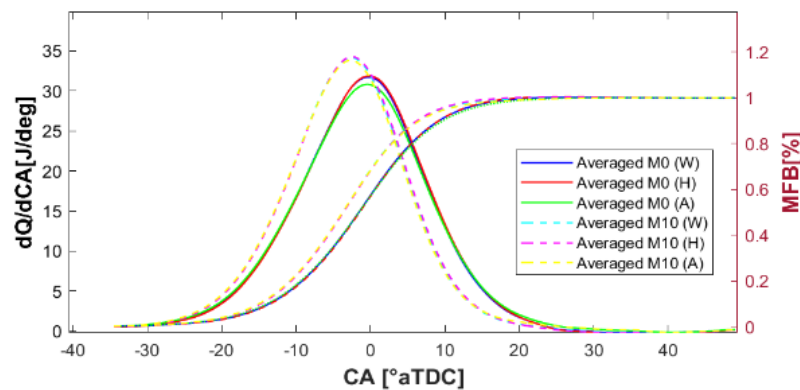


FIGURE 14. Comparison of average RoHR and MFB between two fuels across all correlations

TABLE 6. Heat transfer to the wall for both fuels using different correlations

	Baseline (M0)		Methanol-blended (M10)	
	MFB (%)	°aTDC (deg)	MFB (%)	°aTDC (deg)
Woschni	5	-16.5	5	-17.0
	10	-12.0	10	-13.0
	50	2.5	50	-2.0
	90	13.5	90	6.5
Hohenberg	MFB (%)	°aTDC (deg)	MFB (%)	°aTDC (deg)
	5	-16.0	5	-17.0
	10	-11.5	10	-13.0
	50	2.5	50	-2.0
	90	13.5	90	6.5
Annand	MFB (%)	°aTDC (deg)	MFB (%)	°aTDC (deg)
	5	-16.5	5	-17.5
	10	-12.0	10	-13.0
	50	2.5	50	-2.0
	90	14.0	90	7.0

Figure 14 illustrates the fuel burn progression for both M0 and M10 fuels, with the latter shifting to the left, consistent with the previously discussed rate of heat release (RoHR) trend. Table 6 presents the mass fraction burned (MFB) determined utilizing the Hohenberg correlation at CA50 for the M0 and M10 fuels. The MFB values are 2.5° aTDC for the M0 fuel and -2.0° aTDC for the M10 fuel. This indicates that the MFB for M10 fuel advanced compared to the M0 fuel. In the early combustion phase, M10 fuel showed faster progression, with a slight advancement of about 0.5 to 1.0° aTDC. This advancement became more apparent as the combustion process continued, especially at 50% and 90% MFB, where the M10 fuel combusted significantly faster by 4.0 degrees and around 7.0 degrees, respectively.

The elevated latent heat of vaporization and laminar

flame speed of methanol in the blended fuel contribute to shortened ignition delays. The elevated latent heat of vaporization enhanced charge density by effectively cooling the charge (Sharma et al., 2019). Additionally, the acceleration of the burn rate is facilitated by the elevated laminar flame speed of the M10 fuel, particularly evident during 50% – 90% MFB phase. The increased oxygen content in alcohol-blended fuels improves the laminar flame speed and promotes more consistent combustion compared to the M0 fuel.

CONCLUSION

The results indicated that M10 fuel significantly outperformed the M0 fuel in terms of flame burning

progress. Across all heat transfer correlations, the M10 fuel demonstrated a superior rate of heat release (RoHR). This enhancement was attributed to the inherent characteristics of the M10 fuel, which facilitated the combustion process. This was evident during 50% and 90% of the mass fraction burned (MFB). The differences were notable, with the M10 fuel burning 4.0 and 7.0 degrees faster at 50% and 90% combustion, respectively.

The complete combustion of M10 fuel occurred significantly faster, at 15° aTDC, compared to 21.5° aTDC for the M0 fuel. Additionally, the peak pressure generated by M10 fuel demonstrated lower cyclic variation compared to the peak pressure produced by the M0 fuel. The coefficient of variation (COV) values for M10 fuel were measured at 5.49%, compared to 7.48% for the M0 fuel. This consistency can be attributed to the properties of methanol, specifically its notably higher research octane number (RON) and oxygen content. These characteristics altered the flame pattern development and propagation, resulting in more efficient combustion.

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DECLARATION OF COMPETING INTEREST

None.

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