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Comparative Analysis of Methanol and Ethanol Fumigation in a V-12 Compression-Ignition Engine

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ABSTRACT

The purpose of this research is to examine how methanol and ethanol fumigation influence combustion behavior and engine performance parameters of a heavy-load V-12 CI engine tested at 2000 rpm under full-load conditions, with different diesel injection timings. The results indicate that: (1) Alcohol fumigation was found to moderate the in-cylinder pressure development and reduce the maximum pressure-rise rate compared with conventional diesel operation, especially at higher substitution percentages (SPs); (2) at the same SP, methanol fumigation results in a more pronounced reduction in in-cylinder temperature compared with ethanol fumigation, and a further retardation of diesel injection timing leads to a greater decrease in peak in-cylinder temperature; (3) a noticeable reduction in in-cylinder pressure occurs between -10 °CA before top dead center (bTDC) and 20 °CA after top dead center (aTDC); (4) in the initial phase of combustion, the dual-fuel engine exhibits a reduced heat release rate relative to the conventional diesel engine, particularly when methanol is used; and (5) a slight improvement in brake torque and brake efficiency is observed when the diesel injection timing is retarded.

1. INTRODUCTION

The utilization of alcohol as a fuel in internal combustion engines presents both advantages and challenges. Due to their relatively high latent heat of vaporization, alcohol causes excessive cooling of the intake charge and combustion chamber surfaces. Furthermore, as the alcohol content increases at a constant equivalence ratio, both the maximum cycle pressure and engine power are reduced. Under low-load conditions, it is necessary to decrease the fumigation level of alcohol, mainly to reduce flame quenching and prevent misfires, while at high loads, to prevent preignition and knock. Compared to the corresponding diesel operation, the rates of pressure rise are higher with alcohol fumigation and increase with the increase in ignition delay. The emissions of unburned hydrocarbons (uHC), particulate matter (PM), and carbon monoxide (CO) rise, whereas nitrogen oxides (NOx) emissions decline [1-3].

Methanol also holds increasing relevance in the context of renewable e-fuels. Beyond its conventional production pathways, methanol can be synthesized sustainably through the combination of green hydrogen, produced via renewable-powered electrolysis, and captured CO_2 from industrial or atmospheric sources. This route enables the production of so-called *e-methanol*, a carbon-neutral fuel that can serve as a storage medium for renewable energy while contributing to long-term decarbonization strategies in the transport and power sectors. The integration of e-fuel technologies - supported by recent developments in hydrogen infrastructures and high-temperature electrolysis systems - highlights methanol's potential as a scalable and environmentally friendly alternative for dual-fuel engine applications. Situating methanol fumigation within this broader e-fuel framework provides additional motivation for its investigation as a promising pathway toward cleaner combustion and reduced lifecycle emissions.

Although alcohol fumigation has often been discussed in relation to pollutant emissions, the present study focuses on combustion and thermal behavior rather than emission formation. Therefore, emissions are mentioned only as part of the general motivation for using oxygenated fuels in compression-ignition engines. The main emphasis of this work is placed on in-cylinder temperature, pressure development, heat-release rate (HRR), combustion phasing, and brake performance under methanol and ethanol fumigation.

Considerable research has been devoted to alcohol-diesel blends, especially those involving methanol and ethanol as alternative approaches to reduce NO_x and PM emissions. These alcohols are typically applied with diesel fuel in the form of blends or via fumigation, while alternative methods like dual-fuel injection have attracted relatively limited interest. Compared with blending, the fumigation method offers greater flexibility, although it requires an additional fuel injection system [4, 5]. Therefore, the fumigation approach is adopted in the present study, in which alcohol fuels are premixed with intake air. This approach contributes to smoother engine operation and allows a substantially higher proportion of alcohol to be introduced into the engine.

Numerous researchers have examined this technology, and key findings from their studies are summarized as follows.

Alcohol fumigation has been widely investigated as a dual-fuel strategy for compression-ignition engines. In this approach, alcohol fuel is introduced into the intake flow, while diesel fuel is directly injected into the cylinder to initiate combustion. Previous studies have shown that alcohol fumigation can strongly influence ignition delay, in-cylinder temperature, pressure development, heat-release characteristics, and brake thermal efficiency (BTE). These effects are mainly associated with the high latent heat of vaporization, oxygenated structure, low cetane number, and lower heating value of alcohol fuels [6, 7].

Methanol and ethanol are among the most commonly investigated alcohol fuels for diesel dual-fuel operation. Although both fuels can reduce the thermal intensity of combustion through charge cooling, their effects are not identical because of differences in latent heat, heating value, oxygen content, stoichiometric air–fuel ratio, and reactivity. Therefore, direct comparison between methanol and ethanol under identical boundary conditions is important for understanding their respective combustion-modifying characteristics.

Abu-Qudais et al. [8] examined how ethanol introduced via intake fumigation and ethanol–diesel blends influence the performance parameters and exhaust emissions of a single-cylinder diesel power unit. Their results showed that the optimum ethanol percentage of fumigation and blended mode is 20% and 15%, respectively. With the optimum fumigation, there is an increase in the BTE, CO, and HC emission levels compared with those of the blended mode.

Ogawa et al. [9] investigated characterization of a single cylinder direct injection diesel engine using two fuel systems, including the Common - Rail (CR) diesel injection system and ethanol injection system on the intake manifold, and at the same time using the exhaust gas recirculation (EGR) method. The results showed that with 20% ethanol and optimum EGR rate, the smoke and NO_x both decreased over the entire working range of the engine. Besides, the results also showed that it is necessary to reduce the compression ratio in order to accelerate the mixing of diesel and ethanol while eliminating misfires and cylinder knocking.

In the studies [3, 7, 10-13], fumigation mode has been widely investigated. The application of ethanol fumigation has been shown to significantly reduce emissions of CO_2, NO_x, and PM. However, increases in CO and uHC emissions have been observed with the application of alcohol fumigation. This technique also leads to higher brake-specific fuel consumption (BSFC) due to the greater heat of vaporization of alcohols. In addition, BTE decreases at low engine loads but improves at higher engine loads. Moreover, this method increased the peak heat release rate and ignition delay.

Previous studies on alcohol fumigation in compression-ignition engines have shown that the introduction of methanol or ethanol into the intake flow can substantially affect combustion phasing and heat-release behavior [14-20]. Because alcohol fuels have high latent heat of vaporization and low cetane numbers, their evaporation reduces the intake-charge temperature and can increase ignition delay. These effects influence the crank-angle location of combustion, the maximum pressure-rise rate, and the early-stage HRR. In-cylinder pressure and temperature are also strongly affected by the fumigated alcohol fraction. A higher substitution percentage (SP) generally strengthens the cooling effect and reduces the thermal intensity of combustion. However, the magnitude of this effect depends on fuel properties, engine operating conditions, and diesel injection timing. Therefore, a direct comparison between methanol and ethanol under identical engine conditions is necessary to clarify their different thermal and combustion responses.

The objective of this research is to examine the influence of methanol and ethanol fumigation on combustion behavior and performance indices of a V-12 high-capacity diesel power unit. To realize the engine operation in dual-fuel mode, a methanol/ethanol supply system was incorporated into the intake passage, while maintaining the conventional diesel direct-injection arrangement. Key combustion metrics - such as cylinder pressure and gas temperature, heat release profile, crank angle at 50% fuel mass burned, and peak pressure rise rate - were evaluated at different substitution levels of methanol or ethanol under various start of injection (SoI) diesel injection timings.

Most previous studies have focused on single-cylinder or small automotive diesel engines. Comparatively fewer studies have examined alcohol fumigation in large-displacement, heavy-duty compression-ignition engines. In addition, direct comparison between methanol and ethanol fumigation in a V-12 engine under the same substitution levels and diesel injection timings remains limited. To address this gap, the present study numerically investigates the effects of methanol and ethanol fumigation on the thermal and combustion behavior of a V-12 compression-ignition engine using a validated GT-Power model.

In the revised literature review, priority is given to peer-reviewed studies related to alcohol fumigation, combustion phasing, heat-release behavior, and thermal characteristics of compression-ignition engines. General reference sources are used only where they provide necessary background or engine-specific technical information.

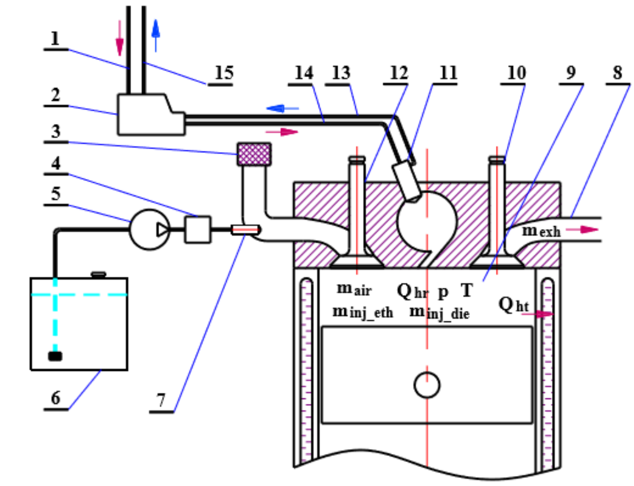
2. MATERIALS AND METHODS

The methanol/ethanol-diesel mixed-fuel supply arrangement is composed of two separate units: the standard diesel injection apparatus and an intake port injection setup dedicated to alcohol fuels, as illustrated in Figure 1.

Table 1. Key characteristics of the fuels [11, 14]

No.	Parameters	Diesel	Ethanol	Methanol
1	Molecular formula	$C_{14}H_{30}$	C_2H_5OH	CH_3OH
2	Molecular weight, [g/mol]	198.4	46.07	32
3	Viscosity at 20 °C, [mPa-s]	2.8	1.2	0.59
4	Density, [kg/m^3]	830	786	790
5	Lower heating value, [MJ/kg]	42.5	28.4	19.7
6	Cetane number	51	≈ 8	4
7	Carbon content [% mass]	87	52.2	37.5%
8	Hydrogen content [% mass]	13	13	12.5%
9	Oxygen content [% mass]	0	34.8	50.0%

An investigation was performed to determine the impact of methanol and ethanol fumigation on the combustion process in a high-capacity V-12 compression-ignition engine, operating at its rated speed of 2000 rpm under full-load conditions. The pertinent fuel properties are provided as shown in Table 1.



1. Diesel supply pipe; 2. High pressure fuel pump; 3. Air cleaner; 4. Secondary fuel filter; 5. Secondary fuel pump; 6. Secondary fuel storage tank; 7. Secondary fuel injector; 8. Exhaust collector; 9. Cylinder combustion chamber; 10. Exhaust outlet valve; 11. Primary fuel injector; 12. Air intake valve; 13. Return fuel pipe; 14. Pressurized fuel line; 15. Return pipe to diesel tank; m_{inj_eth} - Mass of injected ethanol; m_{air} - Mass of intake air; m_{inj_die} - Mass of injected diesel; m_{exh} - Mass of exhaust gas; Q_{hr} - Radiation heat; Q_{ht} - Heat transfer to chamber walls; p - In-cylinder working fluid pressure; T - In-cylinder working fluid temperature.

Figure 1. Schematic diagram of the alcohol fumigation system [7]

The main technical specifications of the investigated V-12 engine are summarized in Table 2. These parameters were obtained from the available technical documentation of the engine and were used to construct the baseline GT-Power model. The technical source is cited only for engine specification purposes and is not used to support the general combustion analysis.

Table 2. Key technical parameters of the tested engine [7]

No.	Parameters	Value	Unit
1	Total cylinders	12	-
2	Engine Configuration	The diesel engine adopts a V configuration, with twelve cylinders organized into two banks of six, separated by a 60° angle.	
3	Ignition order	1^{L}-6^{R}-5^{L}-2^{R}-3^{L}-4^{R}-6^{L}-1^{R}-2^{L}-5^{R}-4^{L}-3^{R}	
4	Ratio of compression	15 ± 0.5	
5	Rated power	387.4/2000	kW/rpm
6	Rated torque	2256.3/1200	Nm/rpm
7	Valve inlet opening	340	°CA aTDC
8	Valve inlet closure	-132	°CA bTDC
9	Valve outlet opening	132	°CA aTDC
10	Valve outlet closure	380	°CA aTDC
11	Fuel consumption rate	265 ± 5	[g.kW.h^{-1}]

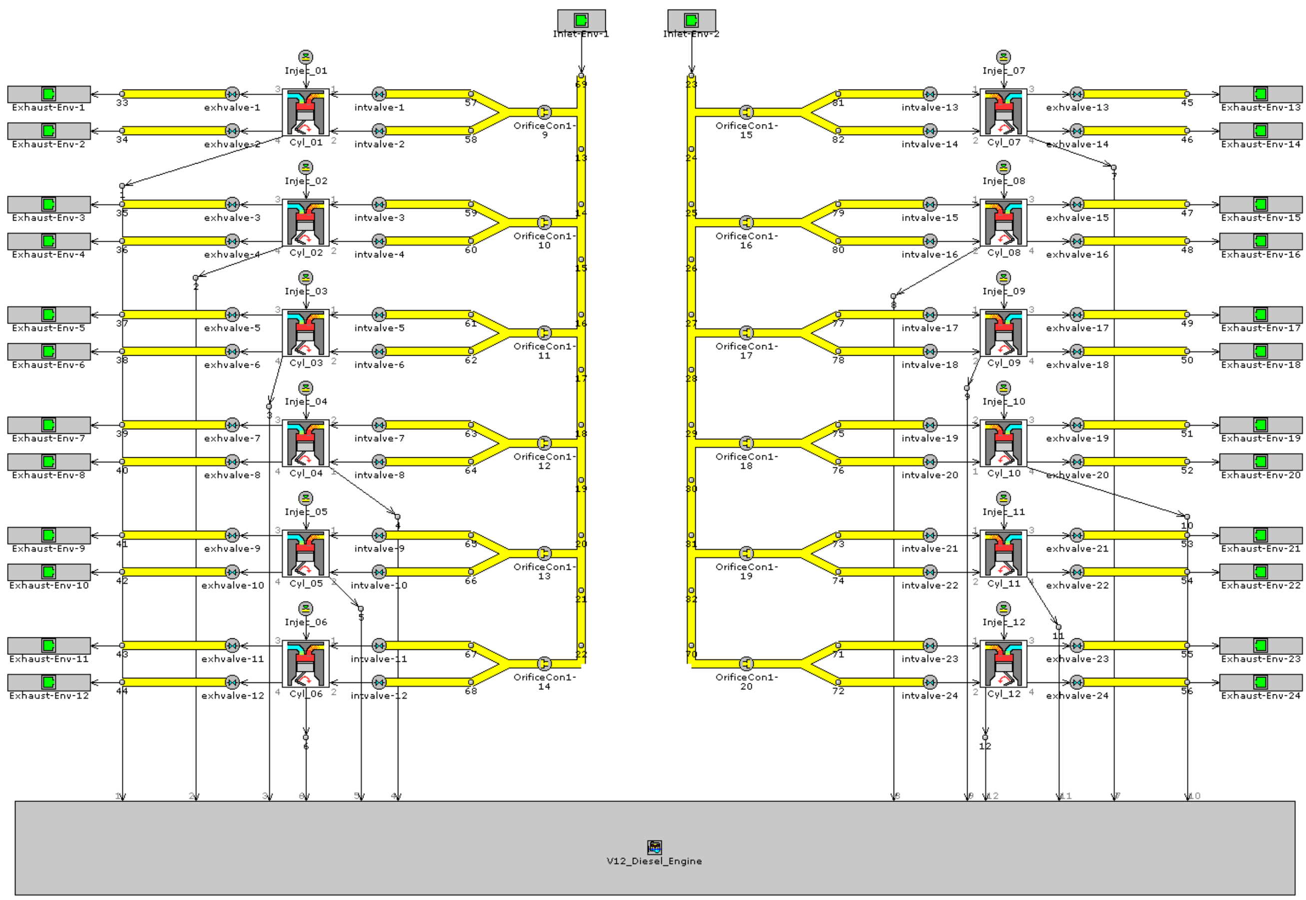


Figure 2. V-12 engine simulation model

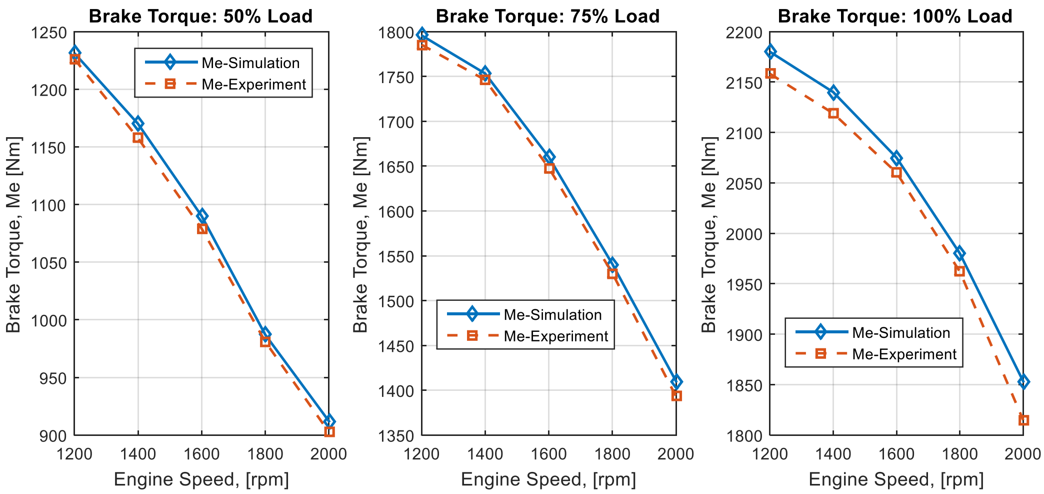


Figure 3. Brake torque results from both simulation and experiment under 50%, 75%, and full load conditions, across engine speeds between 1200 and 2000 rpm

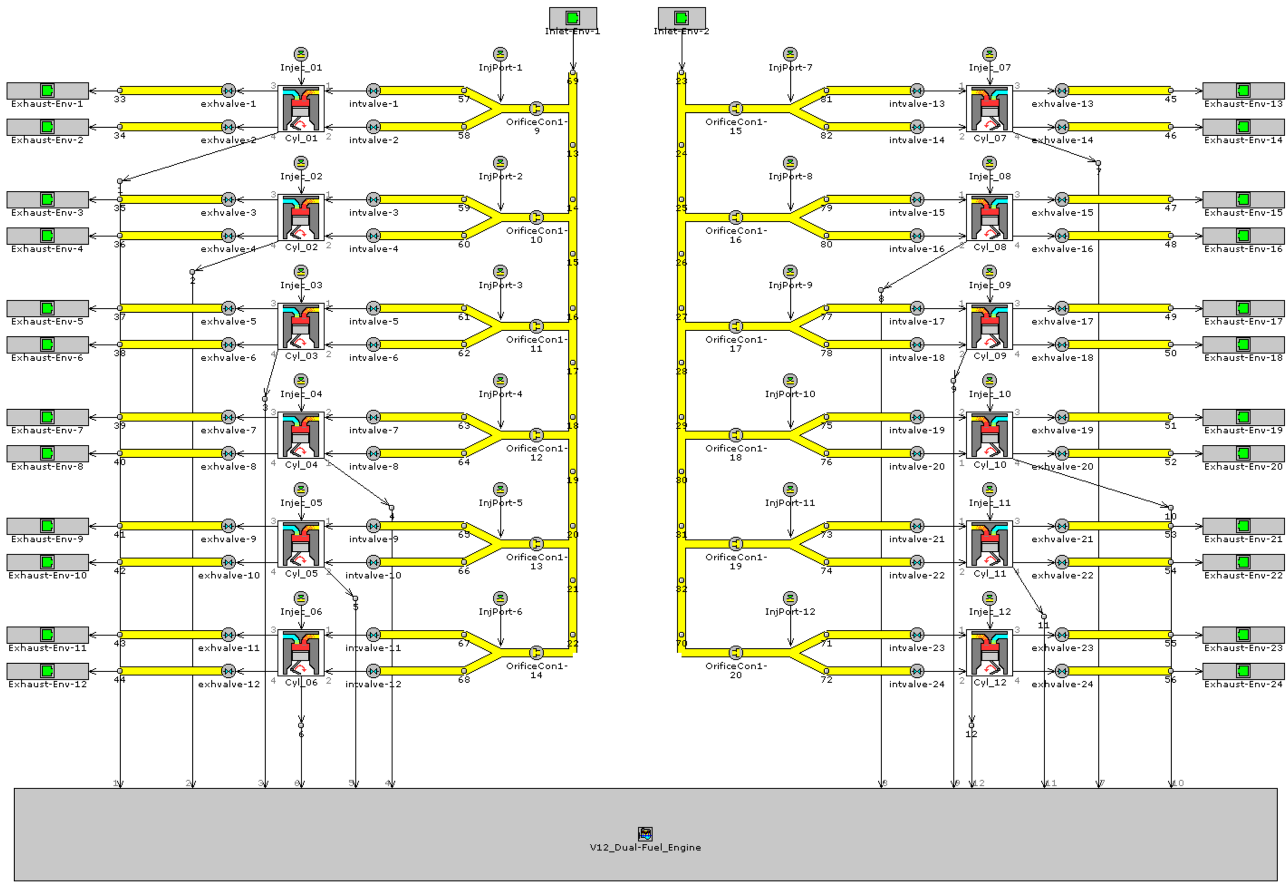


Figure 4. V-12 dual-fuel engine simulation model

Figure 2 depicts the studied engine model developed in GT-Power. Model validation was performed by comparing simulation outcomes with experimental data, as illustrated in Figure 3. As shown in Figure 3, the calculated results for the V-12 engine demonstrate a strong correlation with the experimental data, confirming the precision and dependability of the GT-Power model. The GT-Power model was validated by comparing the simulated brake torque with available experimental data under 50%, 75%, and full-load conditions over the engine-speed range of 1200–2000 rpm. The comparison indicates that the model can reproduce the global performance behavior of the investigated V-12 diesel engine with acceptable agreement. Therefore, the validated model provides a reasonable basis for further parametric analysis of dual-fuel operation under the selected simulation conditions. It should be noted, however, that the validation in the present study is limited to global engine performance because experimental in-cylinder pressure data were not available under the current testing conditions. Consequently, the combustion-related quantities predicted by the model, including in-cylinder pressure, in-cylinder temperature, HRR, pressure-rise rate, and CA50, should be interpreted as simulation-based thermodynamic trends rather than directly validated combustion measurements. This limitation is considered when discussing the results and drawing conclusions.

This validated model can be employed to investigate thermodynamic processes and to configure dual-fuel operation under fumigation conditions. For this purpose, injector assemblies were installed on the intake manifold, and the characteristics of the supplementary fuel were specified, as illustrated in Figure 4.

3. RESULTS AND DISCUSSIONS

The present study focuses on the rated-speed, full-load operating condition of the investigated V-12 compression-ignition engine. This operating point was selected because it represents a thermally demanding condition in which the charge-cooling and combustion-modifying effects of alcohol fumigation can be clearly observed. Within this scope, the main variables are the alcohol type, SP, and diesel SoI. Since only one engine-speed/load condition is considered, the results should not be generalized to the complete engine operating map. The observed trends are therefore discussed specifically for the investigated full-load, 2000 rpm condition. Further simulations and experimental investigations at additional speeds and loads are required to confirm whether the same tendencies remain valid under broader operating conditions.

In this study, simulations were conducted at the maximum brake power speed of 2000 rpm under full-load conditions. The SP of methanol or ethanol was defined as the fraction of diesel fuel mass replaced by the fumigated alcohol, while maintaining the same total fuel quantity. Four substitution levels were considered: 10%, 20%, 25%, and 30%, denoted as M10+D90, M20+D80, M25+D75, M30+D70 for methanol, and E10+D90, E20+D80, E25+D75, E30+D70 for ethanol. The SP was defined on a mass-replacement basis, where a specified fraction of diesel mass (10%, 20%, 25%, or 30%) was replaced by the corresponding mass of methanol or ethanol, while keeping the total injected fuel quantity constant. Because total fuel mass and intake-air parameters were controlled, the global λ remained nearly unchanged across SP cases.

The operating conditions are summarized in Table 3.

Table 3. Simulation scenarios

No.	Variables	Value
1	Pressure	8.59 [bar]
2	Engine speed	2000 [rpm]
3	Load	100%
4	Substitution percentage (SP)	10%, 20%, 25% and 30%
5	Diesel injection timings	-30 °CA, -28 °CA, -26 °CA and -24 °CA (bTDC)
6	Methanol injection timing into intake port	-360 °CA (bTDC)
7	Ethanol injection timing into intake port	-360 °CA (bTDC)
8	Temperature of the injected methanol/ethanol	300 [K]
9	Temperature of the injected diesel fuel	300 [K]
10	Methanol (or ethanol) mass flow rate	10.8 kg/h

Crank angle: °CA; Before top dead center: bTDC; After top dead center: aTDC; Start of injection: SoI.

The simulation matrix was designed to isolate the effects of alcohol type, SP, and diesel injection timing on the combustion and thermal behavior of the engine. The engine speed was fixed at 2000 rpm, and the engine was operated under full-load conditions. The injected fuel temperature, alcohol port-injection timing, and main intake boundary conditions were kept constant for all simulation cases.

The variable inputs were the type of fumigated alcohol fuel, namely methanol or ethanol, the SP, and the diesel SoI. The SP was defined on a mass-replacement basis. For example, M20+D80 indicates that 20% of the baseline diesel fuel mass was replaced by methanol, while 80% of the diesel fuel mass was retained. Similarly, E20+D80 indicates that 20% of the baseline diesel fuel mass was replaced by ethanol. This definition was applied consistently for all substitution cases.

Table 2 presents the fixed technical specifications of the baseline V-12 engine, whereas Table 3 defines the operating and control parameters used in the simulation cases. Thus, Table 2 describes the engine configuration, while Table 3 describes the simulation matrix for the dual-fuel fumigation analysis.

The subsequent sections present a detailed analysis of the simulation results, with a focus on the effects of methanol and ethanol fumigation on combustion behavior and engine performance.

This study is subject to several limitations. First, the experimental validation relied on brake torque measurements, as in-cylinder pressure data could not be obtained due to equipment constraints under current testing conditions in Vietnam. As a result, the model validation focuses on global performance behavior rather than detailed combustion phasing. Second, emission formation was not simulated, and the scope of the analysis is therefore restricted to thermodynamic and performance trends. No claims regarding pollutant emissions are made. Finally, the substitution strategy was evaluated at fixed operating points, and transient effects or turbocharger dynamics were not considered. These limitations should be addressed in future work to further enhance the completeness and generality of the findings.

In the dual-fuel cases, the GT-Power simulation predicts a moderate reduction in the intake-charge temperature due to the high latent heat of vaporization of methanol and ethanol during the fumigation process. The predicted temperature decrease becomes slightly more pronounced at higher substitution levels but remains within a range that does not significantly alter the global excess-air ratio (λ), since both the total fuel mass and the intake-air conditions were kept constant. As a result, the cooling effect manifests primarily through changes in ignition delay and heat-release characteristics rather than through substantial variations in mixture stoichiometry. These trends are consistent with the qualitative thermodynamic behavior typically reported for alcohol-fumigated dual-fuel engines.

To make the comparison more explicit, the following discussion emphasizes not only the direction of variation but also the relative magnitude of the changes observed in the simulation results. The effects of alcohol type, SP, and diesel injection timing are compared using peak in-cylinder temperature, peak pressure, maximum pressure-rise rate, CA50, heat-release-rate characteristics, brake torque, and BTE.

The detailed results corresponding to Figures 5–10.

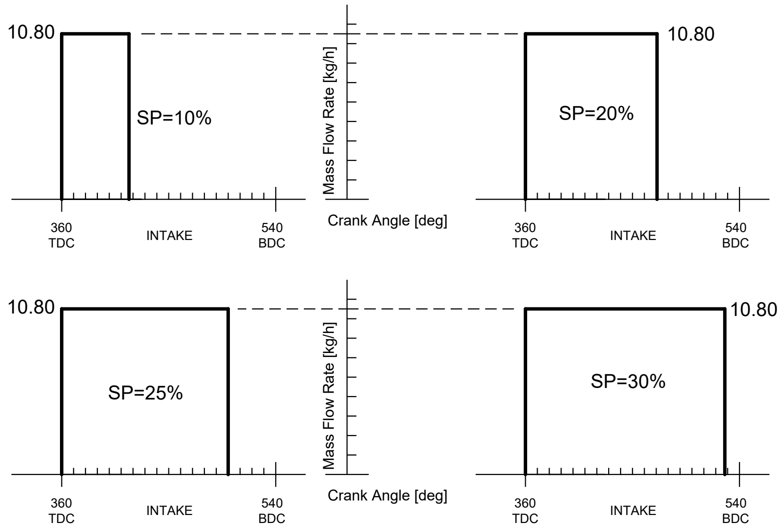


Figure 5. Mass flow rate of the port methanol/or ethanol injector according to the various substitution percentages (SPs)

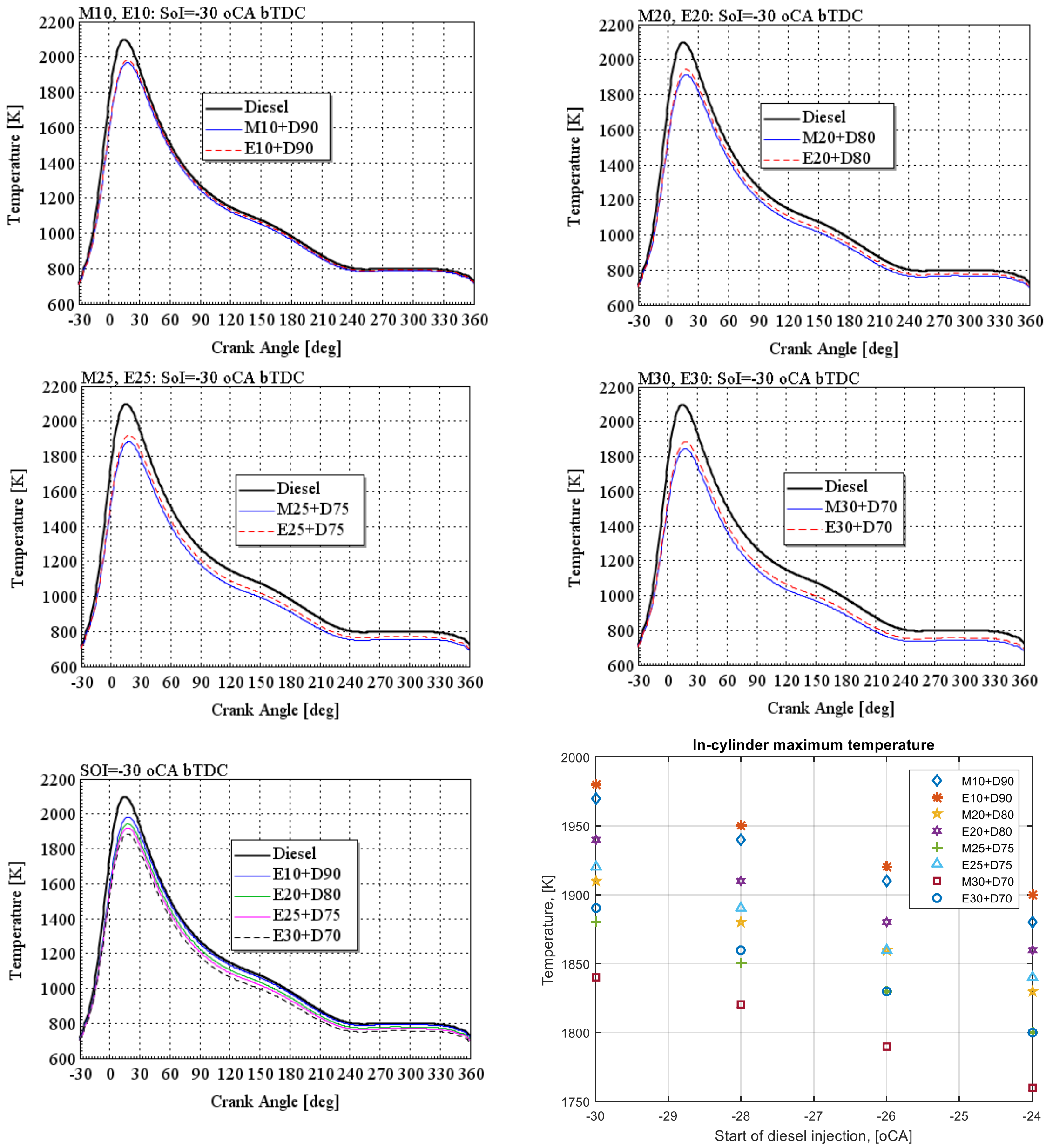


Figure 6. In-cylinder temperature and maximum temperature according to various methanol (or ethanol) substitution percentages (SPs) and start of diesel injections (SoI)

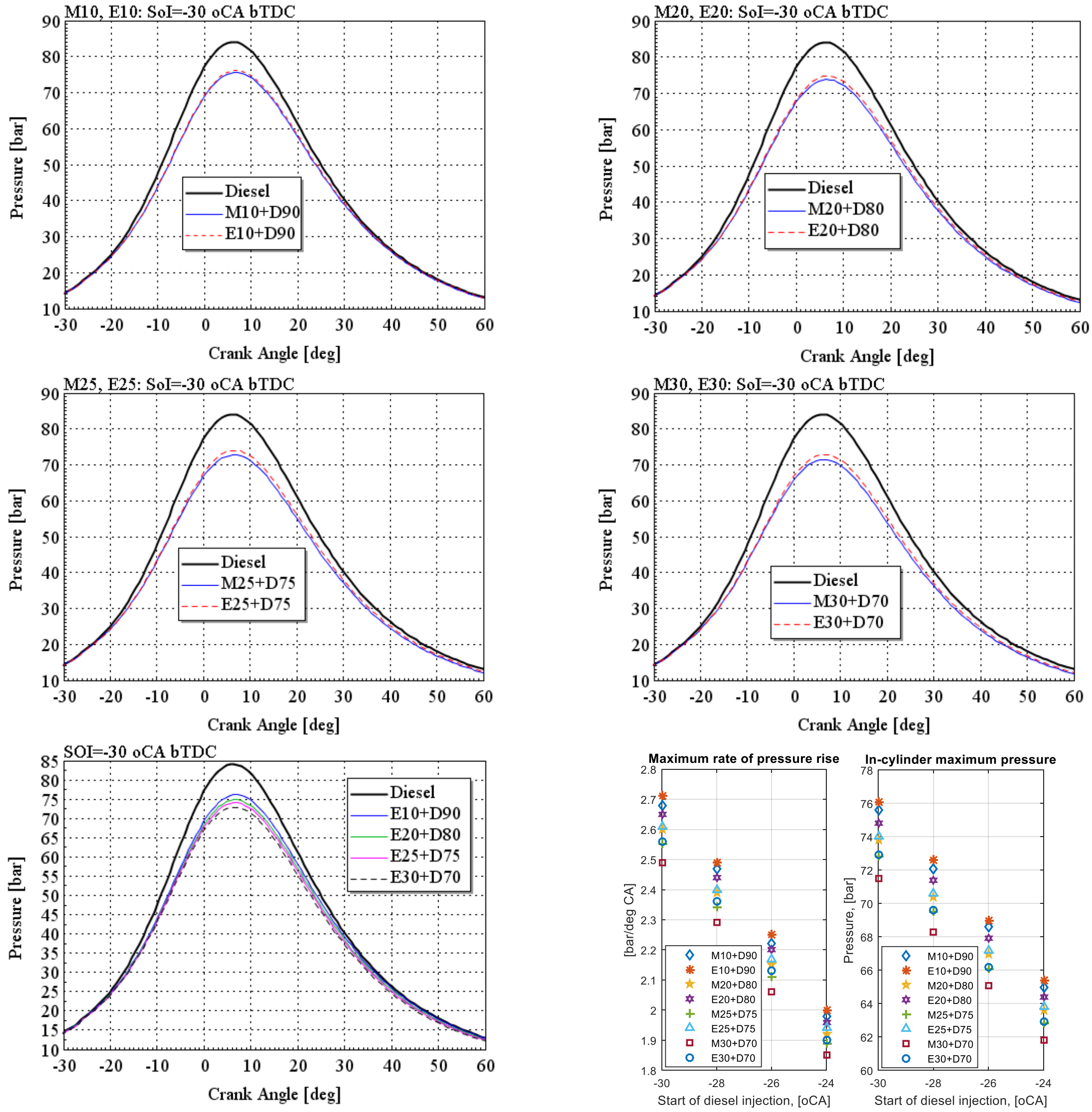
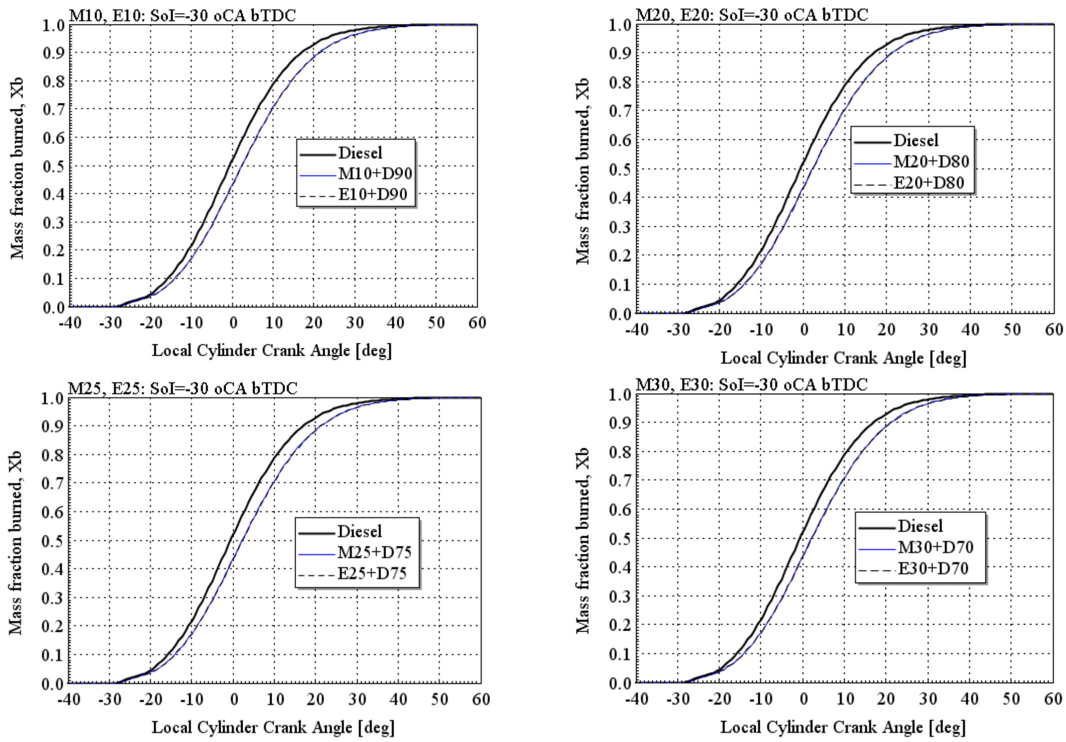


Figure 7. In-cylinder pressure, maximum pressure rise rate, and maximum pressure according to various methanol/or ethanol substitution percentage (SP) and start of diesel injections (SoI)



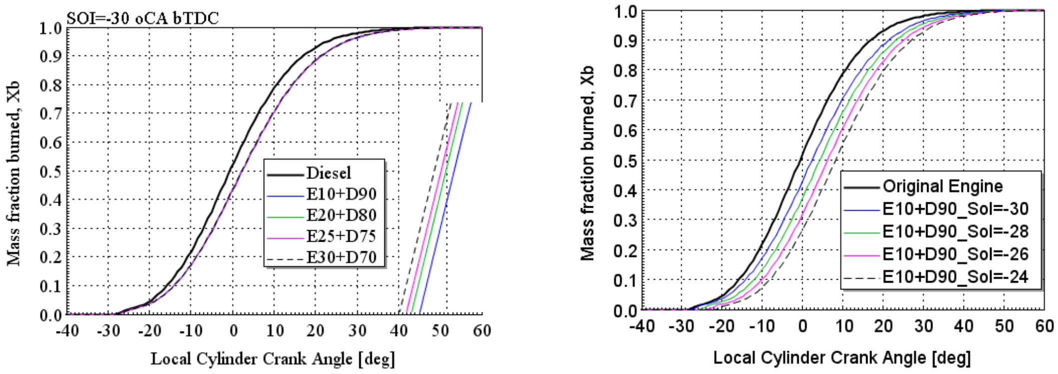


Figure 8. Mass fraction burned as a function of crank angle according to various methanol/or ethanol substitution percentages (SPs) and start of diesel injections (SoI)

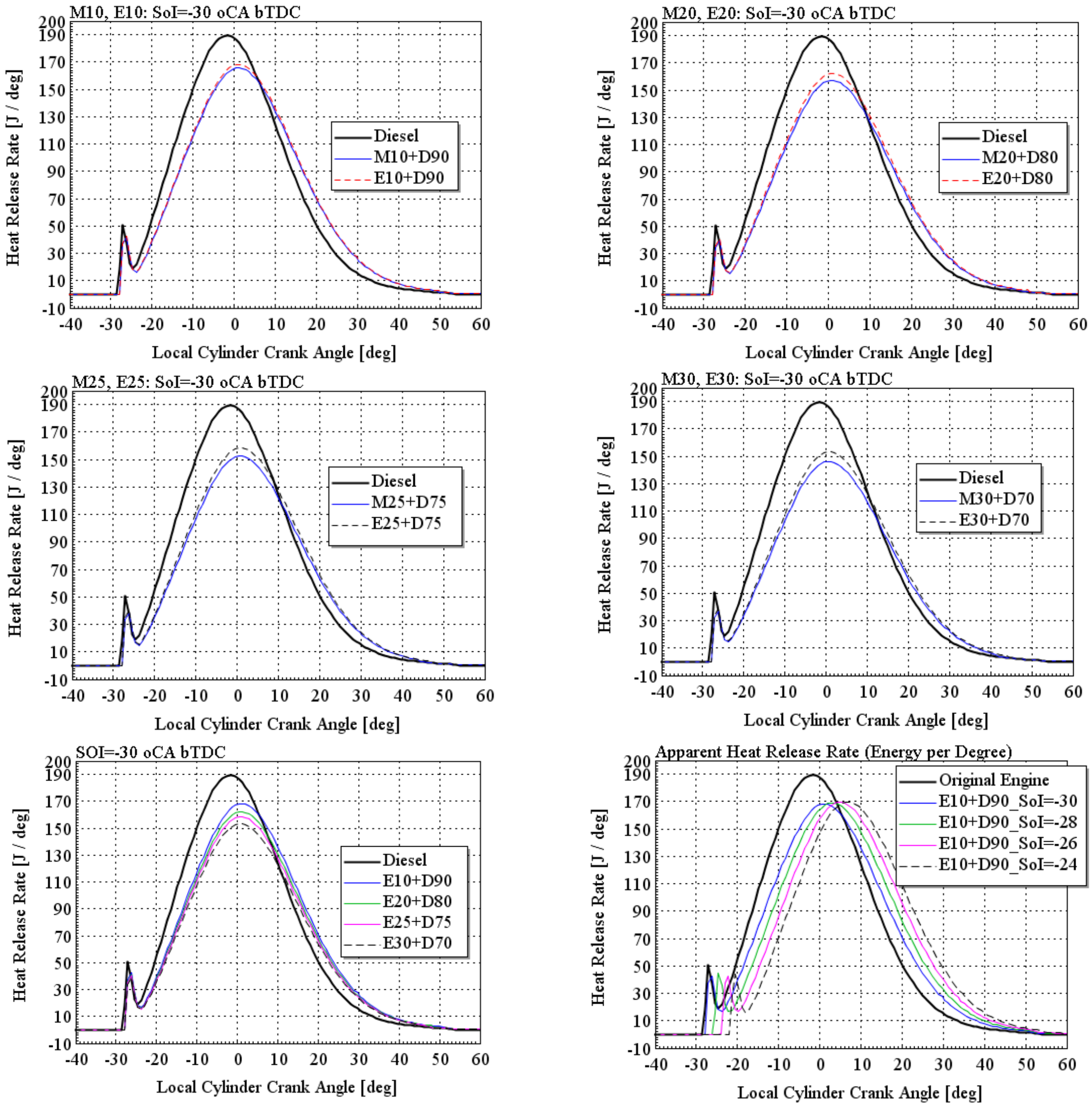


Figure 9. The heat release rate as a function of crank angle according to various methanol/or ethanol substitution percentages (SPs) and start of diesel injections (SoI)

3.1 In-cylinder temperature

Figure 6 shows that increasing the substitution ratio of methanol or ethanol enhances the charge-cooling effect and leads to a clearer reduction in in-cylinder temperature compared with diesel operation. This behavior results from the high latent heat of vaporization of alcohol fuels, which lowers the mixture temperature before combustion. The effect becomes more noticeable at higher substitution levels, while only minor changes occur at small replacement fractions.

The stronger thermal effect of methanol compared with ethanol can be explained by the combined influence of several fuel properties. Methanol has a higher latent heat of vaporization and a lower lower heating value than ethanol. At the same mass-based SP, methanol therefore introduces less chemical energy while absorbing more heat during evaporation. This produces a stronger charge-cooling effect before ignition and reduces the local gas temperature during compression. In addition, methanol has a lower cetane number and higher oxygen content than ethanol. The lower cetane number indicates lower autoignition tendency, while the stronger evaporative cooling further reduces the local mixture reactivity before diesel ignition. Although the oxygen content of methanol may support oxidation during the later combustion stage, the dominant effect under the present fumigation condition is the reduction in pre-combustion temperature and the weakening of the early heat-release intensity. As a result, methanol fumigation produces a larger decrease in peak in-cylinder temperature, maximum pressure-rise rate, and early-stage HRR than ethanol fumigation at the same SP.

For the same SP, methanol produces a greater temperature drop than ethanol due to its higher latent heat and lower heating value, which promotes stronger evaporative cooling. Consequently, the peak in-cylinder temperature decreases more significantly with methanol fumigation.

Compared with the diesel baseline, increasing the alcohol SP progressively reduces the peak in-cylinder temperature. The reduction becomes more visible at higher substitution levels, indicating that the thermal response is more sensitive in the upper substitution range. At the same SP, methanol produces a larger decrease in peak temperature than ethanol because of its stronger evaporative cooling effect and lower energy input.

3.2 In-cylinder pressure

When methanol or ethanol is introduced into the intake manifold in dual-fuel operation, both fuels produce a similar influence on the in-cylinder pressure, as shown in Figure 7. Increasing the substitution ratio consistently lowers the pressure compared with conventional diesel combustion, with the most noticeable reduction occurring between -10 °CA bTDC and +20 °CA aTDC. Outside this region, the differences remain relatively small. The pressure decrease becomes more pronounced at higher substitution levels, particularly for methanol, owing to its stronger charge-cooling effect.

The pressure traces show that the most noticeable differences occur between approximately −10 °CA bTDC and 20 °CA aTDC, corresponding to the transition from late compression to early combustion. Within this crank-angle interval, alcohol fumigation reduces the pressure development rate compared with the diesel baseline. The reduction is more evident for methanol than for ethanol at the same SP.

The maximum rate of pressure rise also declines as the alcohol fraction increases, contributing to smoother engine operation. At the same SP, methanol yields a lower pressure-rise rate than ethanol, again reflecting its greater latent-heat-driven cooling. Furthermore, retarding the start of diesel injection amplifies these trends by delaying heat release and reducing both the peak cylinder pressure and the peak pressure-rise rate.

The reduction in the maximum pressure-rise rate indicates that alcohol fumigation leads to a less abrupt pressure development during the early combustion stage. Therefore, instead of describing the engine operation as "smoother," the present analysis focuses specifically on the calculated pressure-rise-rate behavior.

3.3 Mass fraction burned, X_b

Figure 8 presents the mass fraction burned profiles for different methanol and ethanol substitution levels and diesel injection timings. At a given SoI, dual-fuel operation delays the start of combustion relative to pure diesel operation, leading to a longer ignition delay and a shift of the CA50 (50% fuel mass burned) toward later crank angles. These effects are primarily attributed to the charge-cooling and lower reactivity of the alcohol-air mixture. However, the differences among SPs remain small, indicating that combustion phasing is influenced more strongly by the presence of alcohol fumigation itself than by moderate changes in the substitution ratio.

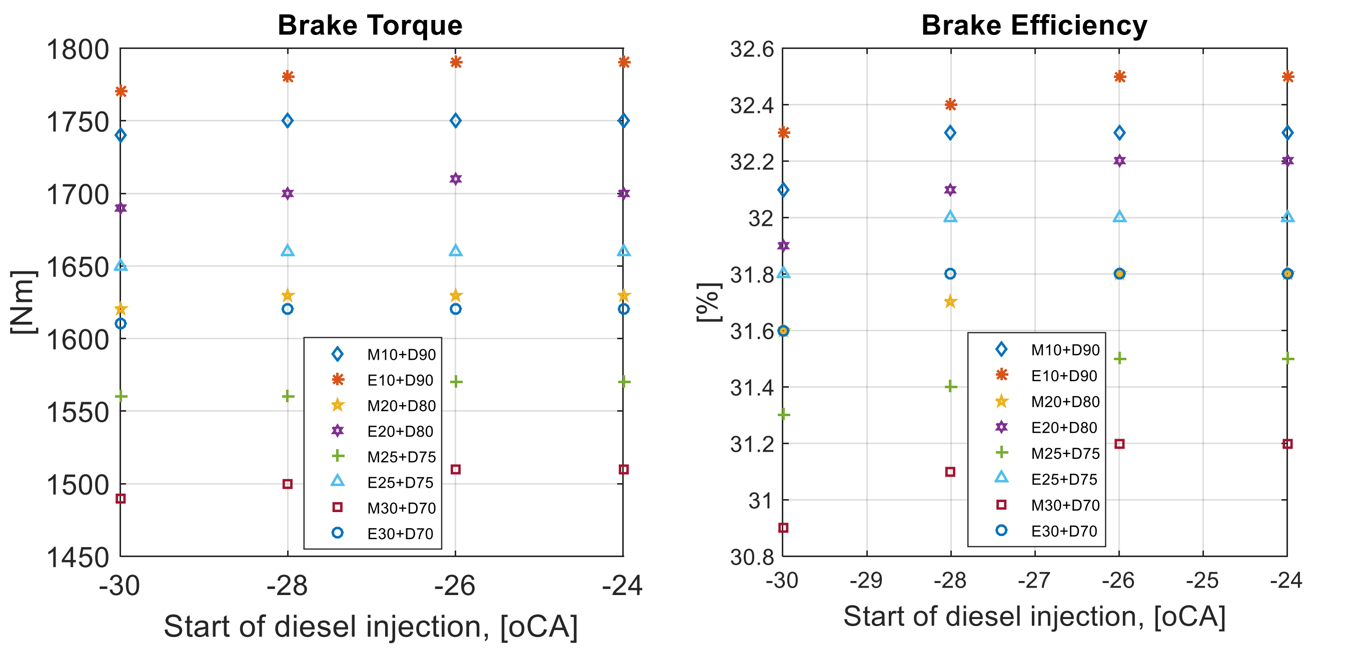


Figure 10. Brake torque and brake efficiency according to various methanol (or ethanol) substitution percentage (SP) and start of diesel injections (SoI)

3.4 Heat release rate

Figure 9 indicates that the HRR in the dual-fuel engine is lower than that of the diesel baseline during the early stage of combustion, with the reduction being more pronounced under methanol fumigation. This behavior is attributed to the lower calorific value and slower energy-release characteristics of methanol and ethanol compared with diesel, as well as the cooling effect introduced by alcohol evaporation, which suppresses the initial premixed burn intensity.

The heat-release-rate curves indicate that alcohol fumigation reduces the early-stage heat-release intensity compared with diesel operation. This reduction is more pronounced at higher SPs and is stronger for methanol than for ethanol. These results suggest that the early combustion process is more sensitive to the combined cooling and reactivity effects of methanol fumigation.

From a heat-release perspective, alcohol fumigation modifies both the intensity and timing of the combustion process. The evaporation of methanol or ethanol in the intake flow absorbs heat from the charge, reducing the temperature of the mixture before ignition. This affects the subsequent diesel-initiated combustion process by changing the ignition delay, early heat-release intensity, and the crank-angle location of the main combustion event.

3.5 Engine performance parameters

Figure 10 shows the variations in brake torque and BTE for different methanol and ethanol substitution ratios and diesel injection timings. A slight improvement in both parameters is observed when the start of diesel injection is retarded. This enhancement results from the modified combustion phasing under dual-fuel operation, where delayed injection reduces peak pressure rise and promotes a more favorable distribution of heat release, thereby improving the overall conversion of fuel energy into useful work.

The results show that alcohol substitution and diesel injection timing affect the combustion process through different mechanisms. The SP mainly controls the strength of the charge-cooling effect and the amount of alcohol-derived chemical energy introduced into the cylinder. Therefore, increasing the SP primarily affects peak in-cylinder temperature, early heat-release intensity, and maximum pressure-rise rate.

In contrast, diesel injection timing mainly governs the temporal location of combustion. Retarding the diesel SoI shifts the main heat-release event toward later crank angles, thereby modifying CA50 and reducing the intensity of pressure and temperature development near top dead center.

The influence of SP becomes more apparent at higher alcohol fractions. Lower substitution levels produce relatively small changes in the combustion traces, whereas higher substitution levels show clearer reductions in peak temperature, pressure-rise rate, and early heat-release intensity. This indicates that the higher substitution range is more sensitive from a thermal-behavior perspective under the investigated operating condition.

Emission formation was not simulated in the present study. Therefore, no conclusions are drawn regarding NOx, CO, HC, or PM emissions. Future work should include emission modeling or experimental emission measurements to evaluate the environmental implications of methanol and ethanol fumigation.

4. CONCLUSIONS

The conclusions of this study are limited to the investigated full-load, rated-speed simulation condition. Broader engine-map investigations are required before the results can be generalized to other engine speeds, loads, or transient operating conditions. The present results demonstrate that methanol and ethanol fumigation can modify the simulated thermal and combustion behavior of the investigated V-12 compression-ignition engine under the selected full-load, rated-speed condition. However, the present study does not provide sufficient evidence to establish general application potential over a wider engine operating range. Therefore, any practical implication of alcohol fumigation should be regarded as prospective and should be further assessed through experimental combustion measurements, emission analysis, and broader operating-condition studies.

Methanol and ethanol fumigation reduced the maximum pressure-rise rate and moderated the early-stage pressure development under the investigated operating condition.

This study numerically investigated the effects of methanol and ethanol fumigation on the thermal and combustion behavior of a V-12 compression-ignition engine using a GT-Power model. Under the investigated full-load, rated-speed condition, alcohol fumigation reduced the peak in-cylinder temperature, moderated the in-cylinder pressure development, lowered the maximum pressure-rise rate, delayed combustion phasing, and reduced the early-stage HRR compared with conventional diesel operation.

At the same mass-based SP, methanol produced a stronger thermal and combustion-modifying effect than ethanol. This behavior is mainly attributed to the higher evaporative cooling effect, lower heating value, lower cetane number, and different mixture reactivity of methanol. Retarding the diesel SoI further shifted the main combustion event toward later crank angles and influenced both thermal behavior and brake performance.

The findings of this study are limited to the investigated simulation conditions. Further experimental validation using in-cylinder pressure measurements, emission analysis, and broader speed/load conditions is required before the practical applicability of methanol and ethanol fumigation in large compression-ignition engines can be fully assessed.

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