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Introduction

The UK government has outlined plans to ban the sale of new petrol and diesel powered vehicles by 2035, eliminating tailpipe emissions. However, the current lack of infrastructure and demand for electric vehicles in the UK poses challenges to their widespread adoption. This report explores alternative solutions, such as synthetic or non-carbon based fuels, to reduce the reliance on non-renewable fossil fuels in internal combustion engines, providing regulations are not altered in 2035. The study aims to propose an alternative fuel strategy and engine architecture that meets a reduced emissions specification. An investigation into alternative fuels and engine technologies supports the choice of an alternative fuel for testing in an ICE. The research recognized the potential benefits of dimethyl ether and ethanol, which were advanced for CFD combustion testing simulations and emissions estimation in Ricardo Wave.

1 Literature Review

1.1 Emissions Regulations

In September 2023, the UK government published an outline detailing target dates and criteria for automotive manufacturers concerning the sale of new automobiles and vans. Key elements include a mandate that by 2030, 70% of new vans and 80% of new cars must be zero emissions, and that by 2035, the sale of new petrol and diesel vans and cars is expected to be banned under a zero emissions regulation (Gov UK, 2023). Assuming these regulations remain unchanged in 2035, alternative non-carbon-based or synthetic fuels for internal combustion engines (ICEs) could serve as effective substitutes for non-renewable petrol and diesel. Emissions, in addition to renewability, are a critical consideration as they are the focus of government targets. For petrol and diesel-powered ICEs, key emissions under scrutiny include carbon dioxide, carbon monoxide, nitrogen oxides (NO_x), and particulate matter (Guzzella & Onder, 2013).

1.2 Internal Combustion Engine Ignition Methods

Various ignition methods for ICEs have been developed and continue to be refined. This section reviews some of the most extensively researched and tested methods. Research regarding engine cycles/stroke is detailed in appendix A.

1.2.1 Compression Ignition

The compression ignition (CI) engine, typically associated with diesel technology, ignites fuel-air mixtures at high pressures without a spark plug. CI engines often use higher compression ratios than spark ignition engines due to diesel's ignition characteristics. Some renewable biofuels and synthetic fuels perform optimally in CI engines; examples include dimethyl ether (DME), synthetic diesel, and biodiesel (Wallner & Miers, 2012). Although CI engines offer improved fuel economy compared to petrol engines, higher combustion temperatures tend to increase NO_x and particulate matter emissions. Nonetheless, the use of exhaust gas recirculation (EGR), diesel particulate filters (DPF), selective catalytic reduction (SCR), and other emission control systems effectively regulates emissions (Mollenhauer & Tschöke, 2010).

1.2.2 Spark Ignition

Spark ignition (SI) engine architecture is predominantly optimised for petrol/iso-octane. Recently, blends of iso-octane and alcohol-based fuels such as ethanol have been introduced to reduce carbon dioxide (CO₂) emissions and fossil fuel use (Gov UK, 2022). While most modern SI vehicles are compatible with ethanol blends, some are not. This is primarily due to differences in engine design, given ethanol's distinct combustion and ignition properties compared to iso-octane, and material compatibility issues with metals, rubbers, and plastics in fuel systems. Previous investigations have shown promising results for alternative fuels in SI engines, including alcohol-based fuels (ethanol, methanol, butanol), hydrogen, liquified petroleum gas (LPG), compressed natural gas (CNG), and ammonia (Ravi, et al., 2017). Each fuel offers distinct emissions benefits but poses unique ignition and combustion challenges that require adaptations in engine design to optimise performance, efficiency, and emissions.

1.2.3 Homogeneous Charge Compression Ignition (HCCI)

Homogeneous charge compression ignition uses principles from both SI and CI engines to optimise fuel efficiency and reduce emissions. SI technology is adopted where fuel and air are pre-mixed before compression within the combustion, this is used instead of CI stratified direct injection technology to achieve a homogenous mixture. However, in contrast to SI engines that rely on spark plugs to ignite the fuel/air mixture, HCCI utilises specific temperatures and pressures to simultaneously auto-ignite the homogeneous mixture (Kumano, et al., 2008). Benefits of this system is reduced fuel consumption, increased thermal efficiency and reduced emissions, in particular NO_x. However, associated problems are limited operating ranges, challenging combustion control (pressures, temperatures and mixture must be exact), robust engine construction requirements, higher carbon monoxide and hydro-carbon emissions (Verma, et al., 2022). A comparison of combustion strategies is demonstrated in figure 1.

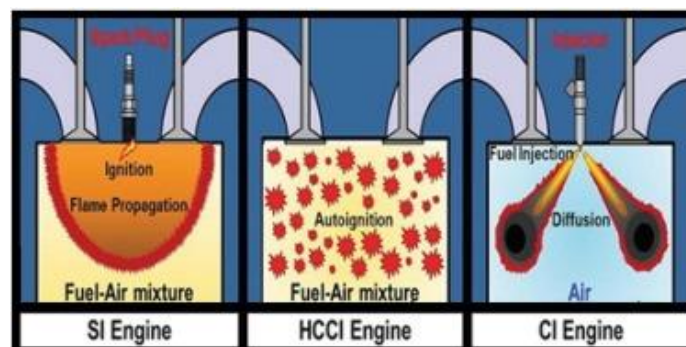


Figure 1: Comparing Combustion Strategies of Spark Ignition, Homogeneous Charge Compression Ignition and Compression Ignition (Verma, et al., 2022).

1.2.4 Reactivity Controlled Compression Ignition (RCCI)

Reactivity controlled compression ignition (RCCI) is a dual-fuel technology under development to reduce fuel consumption and pollutant emissions. It employs two fuels through in-cylinder blending and multiple fuel injections (Singh, et al., 2019). Initially, a low-reactivity fuel (low cetane number) is injected into the intake port, where it mixes with air and recirculated gases before entering the combustion chamber. Subsequently, a highly reactive fuel is directly injected prior to ignition using single or multiple injections, as shown in figure 2 (Reitz & Duraisamy, 2015). Various dual-fuel setups, such as petrol-diesel, ethanol-diesel, and petrol-biodiesel, have been tested in both light and heavy-duty vehicles (Kokjohn, et al., 2011).

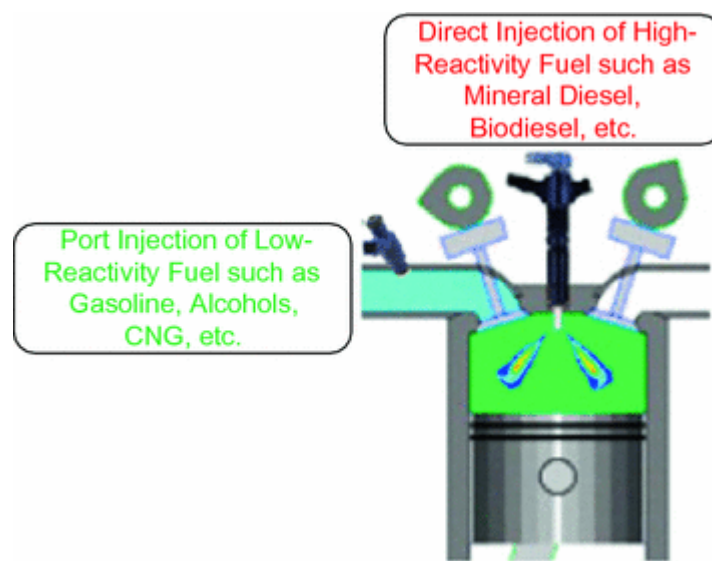


Figure 2: Fuel Injection Points Diagram for RCCI Engine (Singh, et al., 2019).

Reitz & Duraisamy (2015) conduct a comprehensive research and computational simulation study on many aspects of RCCI technology. Investigations into alternative dual fuel strategies, injection techniques and spray targeting all aim to reduce emissions. Testing indicated that direct injection of both fuels results in less unburnt fuel in crevice regions giving similar: performance figures, hydrocarbon and carbon monoxide emissions, but significantly reducing NO_x. The plots in figure 3 demonstrate the primary differences in operating temperatures between conventional diesel and RCCI. The capability of running lower fuel equivalence ratios in RCCI engines also drastically reduces soot and NO_x production due to lower peak combustion temperatures.

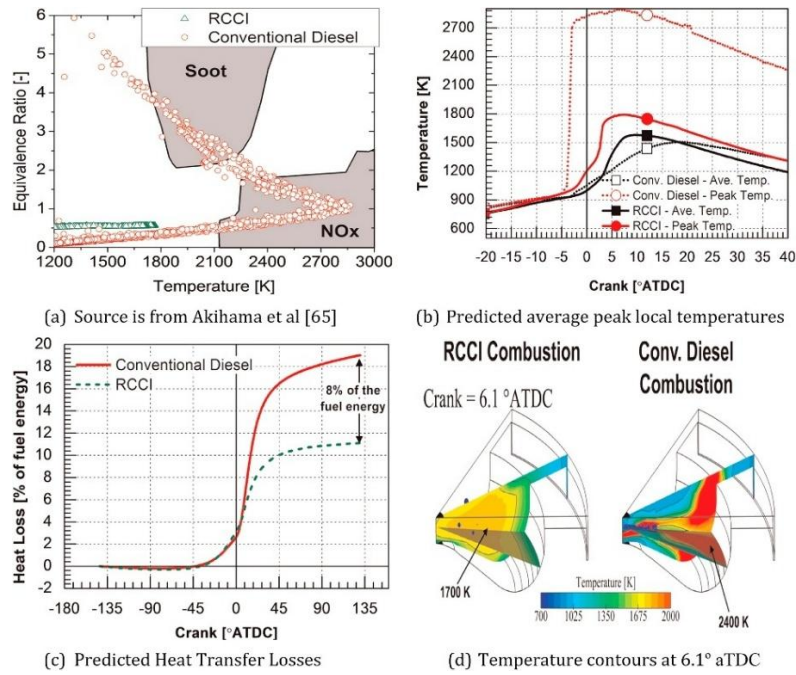


Figure 3: Model Predicted Comparison for Local Temperature, Heat Transfer Loss and Temperature Contours. Comparing Conventional Diesel with RCCI Diesel-Petrol (Reitz & Duraisamy, 2015).

1.3 Alternative Fuels

With the forthcoming end of petrol and diesel vehicle sales, alternative fuels suitable for internal combustion are undergoing extensive investigation and testing. Figure 4 details a variety of fuels available listed under their associated categories. Hydrogen is detailed in appendix B.

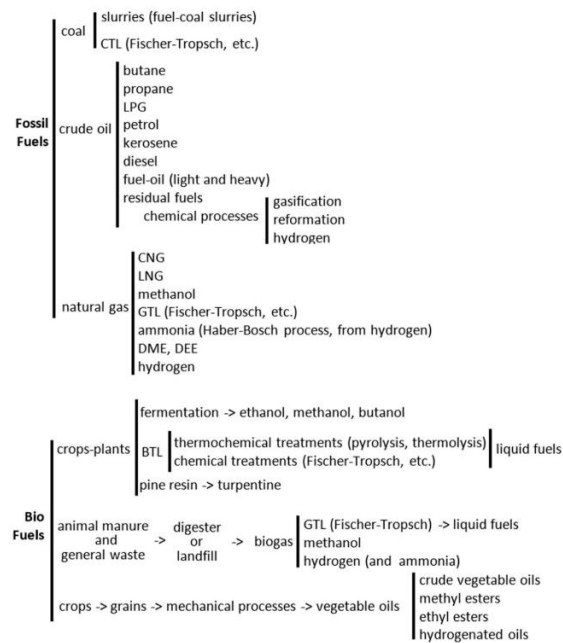


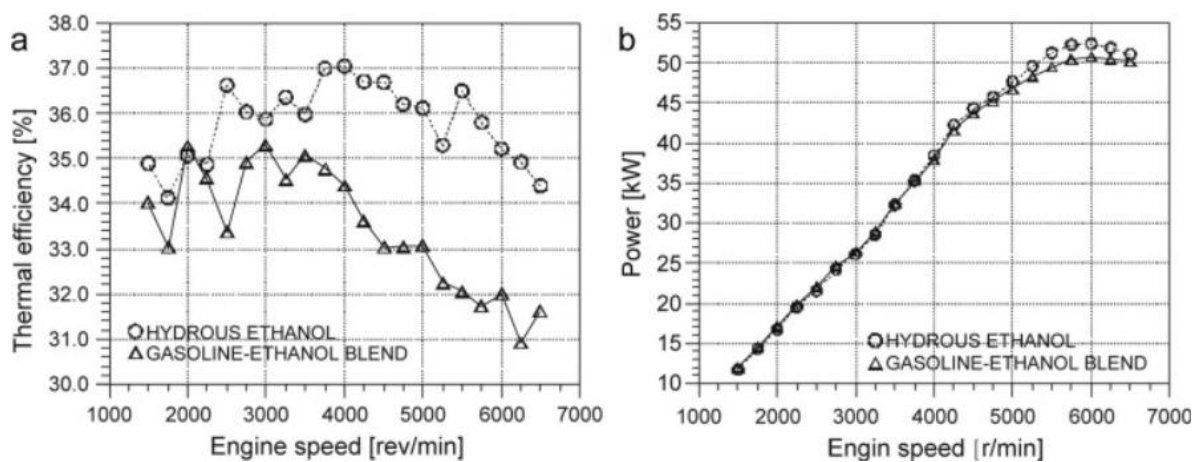
Figure 4: Fossil Fuels and Biofuels Grouped into Associated Categories (Martins & Brito, 2020).

1.3.1 Alcohols

Renewable biofuels such as bio-alcohols are a viable alternative to conventional petrol and diesel. However, alcohols exhibit distinct combustion characteristics that require specific engine design modifications for optimal performance. Ethanol (C_2H_5OH), derived from biomass fermentation, is commonly blended with petrol to reduce emissions and fossil fuel consumption, though it can also be used independently with appropriate adaptations (Wang, et al., 2022). Table 1 details the properties of anhydrous and hydrous ethanol, highlighting significant differences in stoichiometric air/fuel ratios and heating values. Graph 1 shows that hydrous ethanol achieves higher thermal efficiency than a petrol-ethanol blend, resulting in increased engine power due to its reduced burn duration.

Table 1: Comparing Properties of Bioethanol, Biobutanol and Petrol (Hönig, et al., 2014).

Parameter	Ethanol	Butanol	Gasoline
Chemical formula	C_2H_5OH	C_4H_9OH	$CH_{1.87}$
Density at 15 °C(kg dm ⁻³)	0.79	0.81	~ 0.73
Kinematic. viscosity at 20°C (mm ² s ⁻¹)	1.52	3.64	0.4–0.8
Boiling point (°C)	78	118	30–215
Calorific value (MJ kg ⁻¹)	26.8	32.5	42.9
Heat of vaporization (MJ kg ⁻¹)	0.92	0.43	0.36
Vapor pressure (kPa)	19.3	18.6	60–90
Mixed octane number			
RON	106–130	94	95
MON	89–103	80–81	85
Oxygen content (% by weight)	34.7	21.6	< 2.7



Graph 1: Testing Results comparing Hydrous Ethanol (HE100W6.8 - Hydrous Ethanol Containing 6.8 Vol% Water) to E22 (78 Vol% Gasoline And 22 Vol% Anhydrous Ethanol) for a) Thermal Efficiency and b) Power (Wang, et al., 2022).

Hamid, *et al.* (2016) indicate that butanol (properties detailed in Table 1) is a renewable fuel that is less corrosive and less prone to water contamination than ethanol. Butanol's calorific value is lower than petrol's but higher than ethanol's, resulting in slightly lower fuel consumption than ethanol. However, high butanol blends and pure butanol raise concerns over knocking and long-term engine durability. Graph 2 details the mass fraction burn rates of ethanol, butanol, iso-octane, and petrol. Both alcohols behave similarly, though ethanol burns slightly faster (Aleiferis, *et al.*, 2009), as supported by flame growth imaging Figure 5.

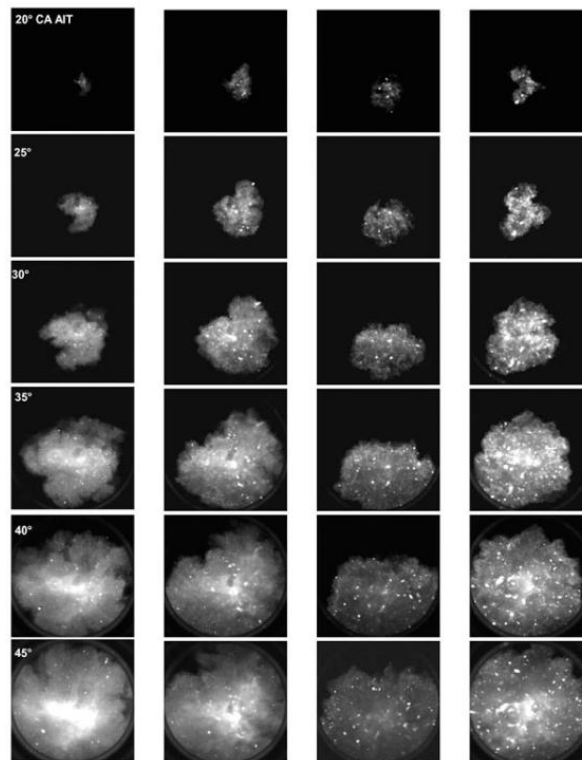
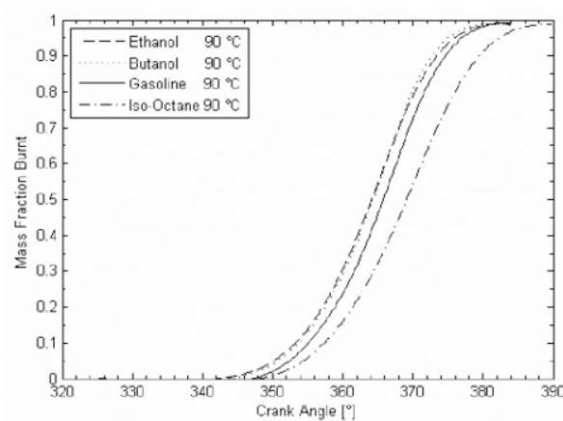


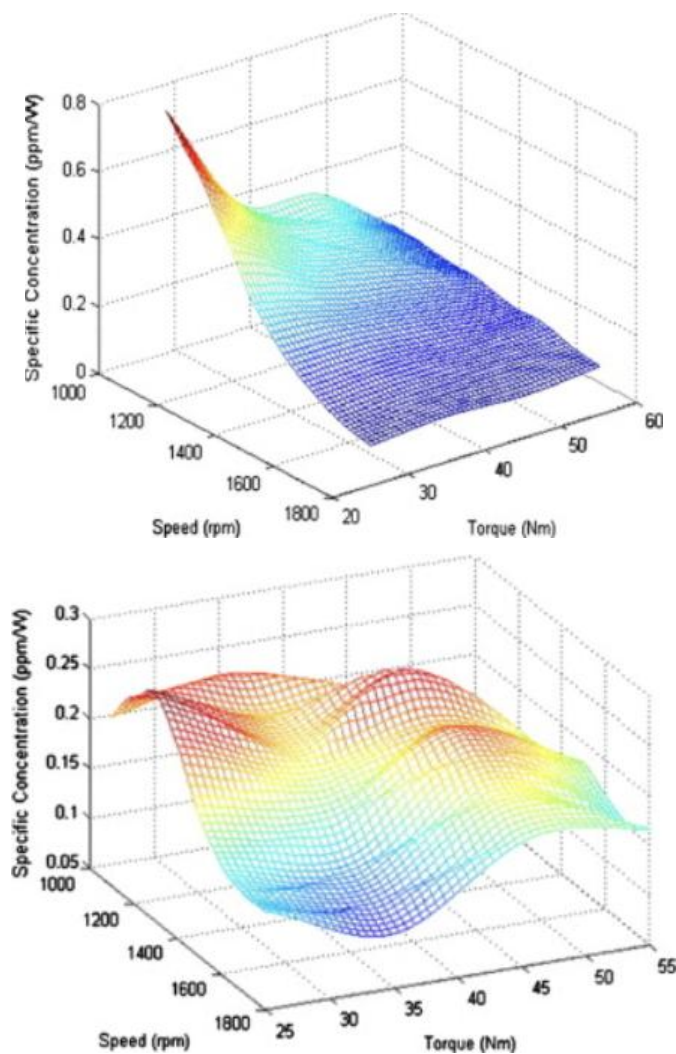
Figure 5: Flame Growth: Ethanol (column 1), Butanol (column 2), Iso-Octane (column 3) and Petrol (column 4) (Aleiferis, *et al.*, 2009).



Graph 2: Mass Fraction Burn Profile Comparing Ethanol, Butanol, Iso-Octane and Petrol (Aleiferis, *et al.*, 2009).

1.3.2 Dimethyl Ether (DME)

Dimethyl ether (DME) is a synthetic fuel with the chemical composition CH_3OCH_3 (or $\text{C}_2\text{H}_6\text{O}$). It can be produced from various feedstocks, making it renewable depending on the manufacturing method (Boudreaux, 2021). DME serves as a viable alternative to diesel in CI engines due to its slightly higher cetane number and absence of carbon-carbon bonds, which reduces particulate matter production and eliminates soot (Singh, et al., 2021). Thomas, *et al.* (2014) found that, without combustion or fuelling adjustments, NO_x emissions from DME decrease significantly with increasing engine speed and load, while diesel NO_x remains relatively constant (Graph 3). With appropriate fuelling and combustion adjustments, DME performance can be further optimized.



Graph 3: Surface Plot for Specific NO_x Emissions, Torque and Engine Speed for (Top) DME and (Bottom) Diesel in a CI Engine with no Modifications to the Combustion or Fuelling System (Thomas, et al., 2014).

2 Methodology Overview

Based on the literature review, this study will investigate DME's potential to reduce harmful emissions. Initial experiments will evaluate DME in SI engines with a lower compression ratio as an alternative to CI. Additional tests will examine the results of ethanol to assist in eradicating knock and pre-ignition issues prone to be caused by DME. The aim is to finish with a low emission DME-ethanol blend for use in SI engines. These fuels are selected for their renewability, supporting solo-fuel experimental data, and proven emission reductions. SI technology is selected for its precise combustion control, lower NO_x production than CI (RAC, 2025), and enhanced lean-burning capabilities. The near-60% thermal efficiency achieved by RCCI engines using low and high reactivity fuels in CI engines inspired this fuel blend strategy and the functional feasibility study in SI engines (Reitz & Duraisamy, 2015). Anticipated challenges with DME auto-ignition in SI engines will be addressed through blending and other means (EGR, ignition timing and more). Therefore, combustion characteristics and results analysis for individual testing of ethanol and DME will heavily influence the understanding behind success or failure of the blend. The testing methodology is shown in Figure 6. Although HCCI merits consideration for current development and emission reductions, its inconsistent operation across typical engine speeds excludes it from this study. While this report primarily discusses well-known alternative fuels, options such as ammonia, methanol, and butanol also merit consideration.

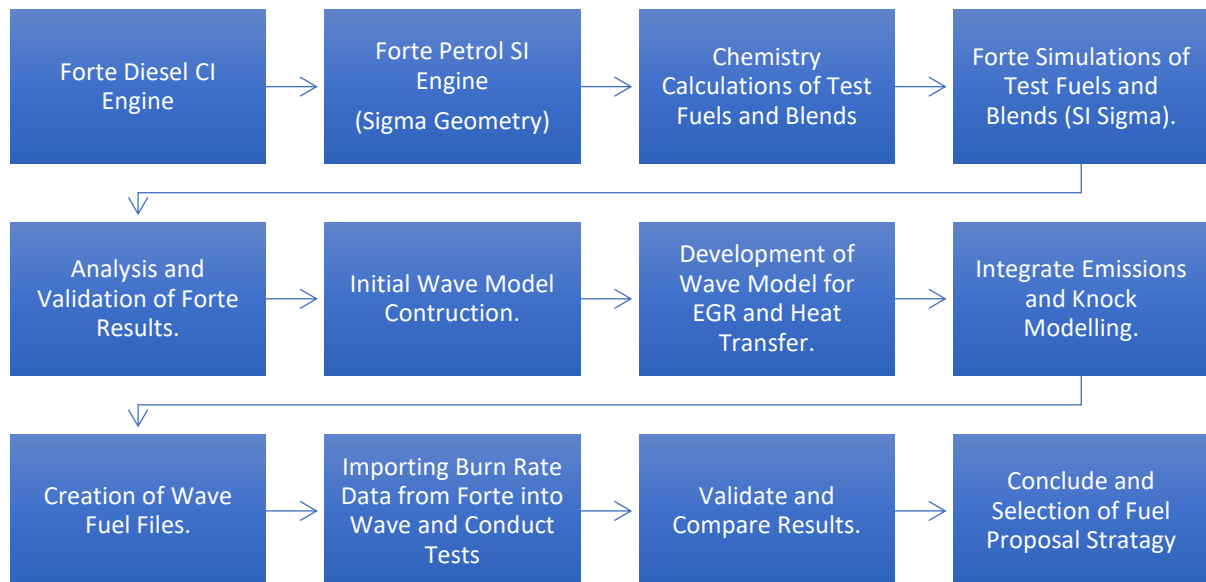


Figure 6: Testing Methodology Flow Chart.

3 ANSYS Forte Combustion Simulation

3.1 Diesel Compression Ignition

Using the ANSYS Forte tutorial manual, a basic sector mesh for a diesel-powered CI engine was set up to obtain baseline data (ANSYS Inc, 2021). The sector geometry (Figure 7) is based on research by the Engine Combustion Network (ECN), an “international collaboration among experimental and computational researchers in engine combustion” (ECN, 2025). Once geometry was established, the mesh and remaining settings were configured per the Forte tutorial, with results presented in Graph 4.

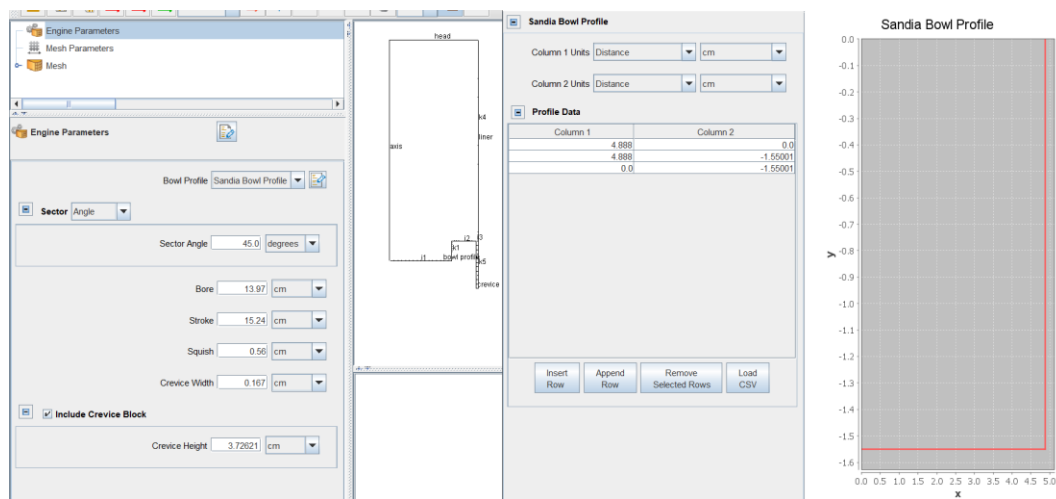
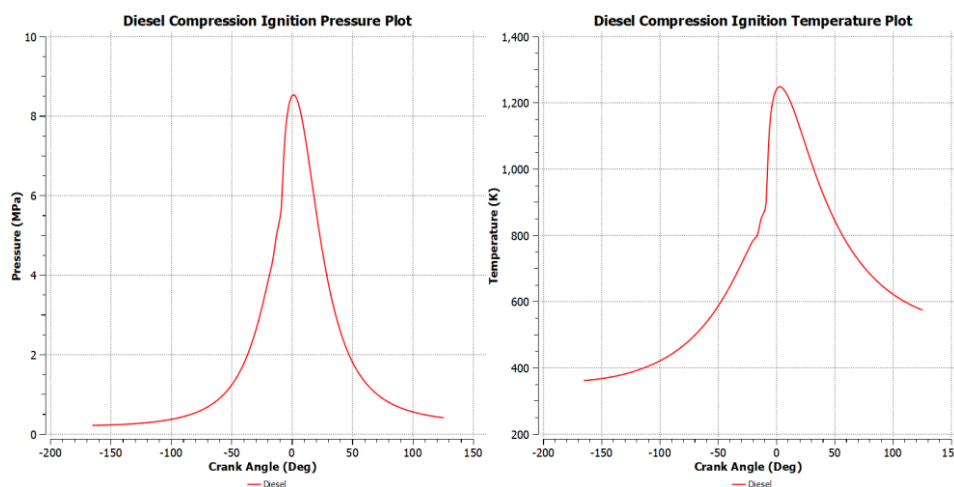


Figure 7: Diesel Sector Setup Geometry.



Graph 4: Diesel Compression Ignition Tutorial Model Results (Left) Pressure Data, (Right) Temperature Data.

3.2 Sigma Petrol Spark Ignition

The 1.6-litre SI Ford Sigma engine was selected for testing the ethanol and DME blend. Its geometry was constructed in ANSYS as a 60-degree cylinder section. Figure 8 details bore and stroke per Sigma engine data and estimations for crevice width, crevice height and squish. Due to a lack of piston data, the bowl profile was assumed flat, potentially affecting accuracy. The mesh was then created following the diesel sector mesh tutorial (ANSYS Inc, 2021).

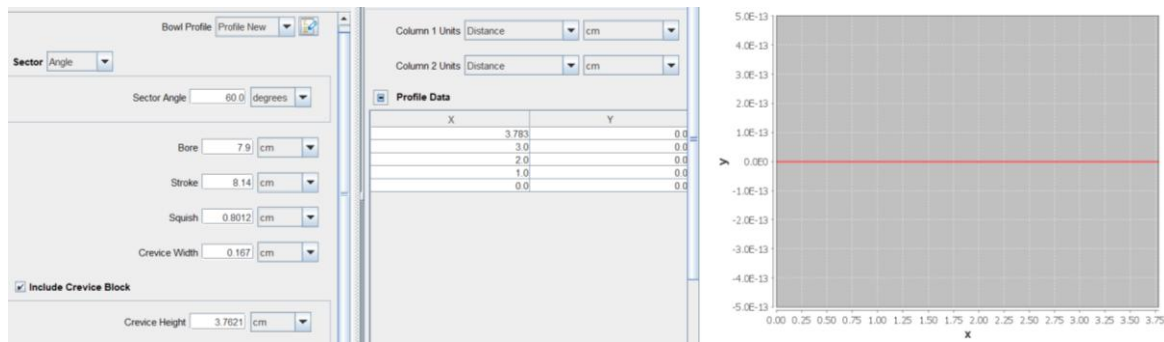
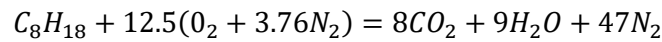
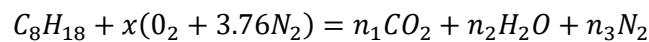


Figure 8: Ford Sigma Geometry Section and Bowl Profile.

3.2.1 Iso-Octane Chemical Composition

Prior to entering mixture composition into the Forte simulation, stoichiometric combustion of iso-octane (C_8H_{18}) is balanced in a MATLAB script and demonstrated in equations below:



The balanced chemical equation of stoichiometric combustion for iso-octane can be used to calculate air-fuel ratio by mass, using table 2. Mixture composition can then be calculated in terms of mass fractions for entry into Ansys Forte, demonstrated figure 9 and equations below.

Table 2: Element Atomic Weights (NIST, n.d.).

Element	Carbon	Hydrogen	Oxygen	Nitrogen
Atomic Weight	12.011	1.00784	15.999	14.0067

$$\begin{aligned}
C_8H_{18} & (8 \times 12.011) + (18 \times 1.00784) = 114.22912 \\
O_2 & (12.5 \times 2 \times 15.999) = 399.975 \\
N_2 & (12.5 \times 2 \times 3.76 \times 14.0067) = 1316.6298 \\
O_2 \text{ \& } N_2 & 399.975 + 1316.6298 = 1716.6048 \\
O_2 \text{ \& } N_2 \text{ \& } C_8H_{18} & 1716.6048 + 114.22912 = 1830.83392 \\
\text{Air-Fuel Ratio} & \frac{1716.6048}{114.22912} = 15.02773373
\end{aligned}$$

Calculations of mass fractions:

$$\begin{aligned}
C_8H_{18} & \frac{114.22912}{1830.83392} = 0.06239185256 \\
O_2 & \frac{399.975}{1830.83392} = 0.2184660201 \\
N_2 & \frac{1316.6298}{1830.83392} = 0.7191421273
\end{aligned}$$

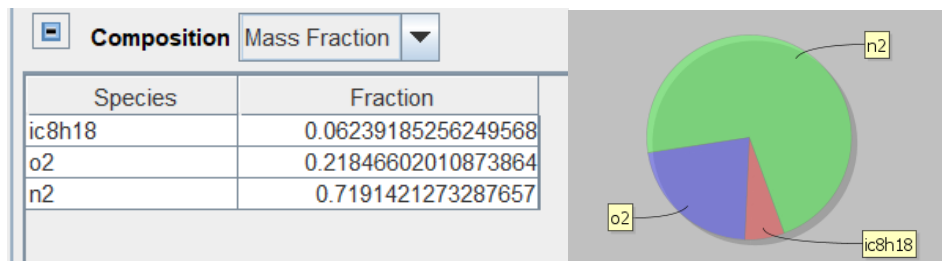


Figure 9: Stoichiometric Mixture Composition for Iso-Octane (Mass Fraction).

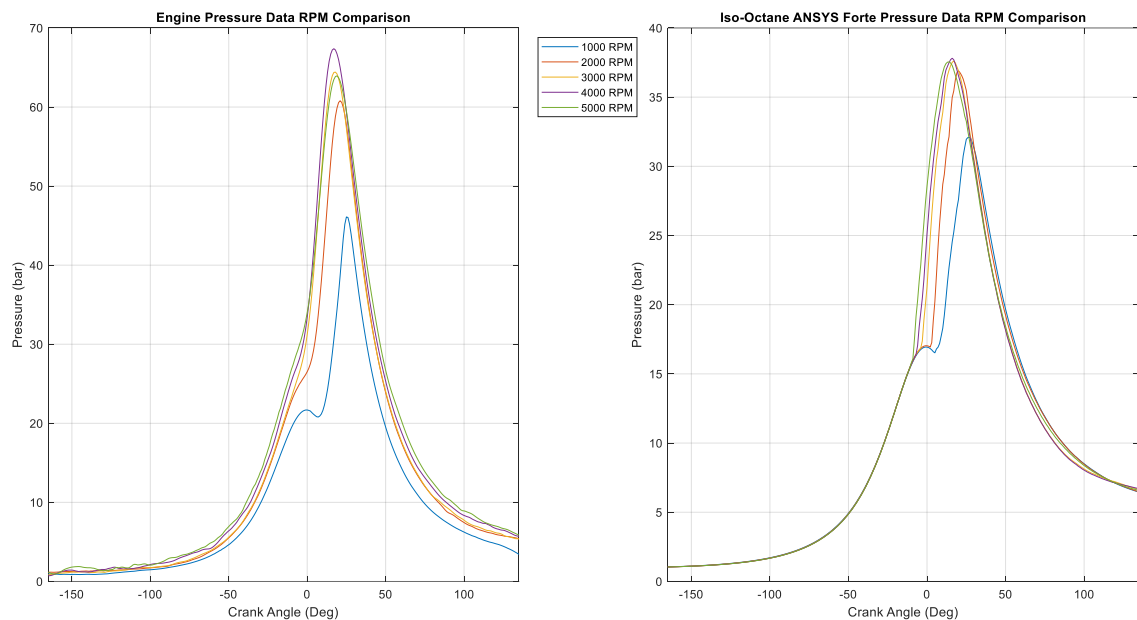
Additional setup data regarding ignition timing, initial temperature, and pressure is detailed in Table 3. These parameters are loosely based on actual Sigma engine data; however, because liner, piston, and cylinder head temperatures are unknown, logical assumptions were made. Consequently, initial pressure and temperature were held constant across all RPMs, even though different RPMs experience variations in temperatures and pressures. The pre-specified fuel file in ANSYS models soot, NO_x, CO, CO₂, hydrocarbons, particulate matter, and more.

Table 3: Iso-Octane Setup Forte Parameters.

Fuel File	Gasoline_1comp_49sp_chem.inp				
Setup Data	RPM				
	1000	2000	3000	4000	5000
Spark Start Angle (deg)	5	3	-3	-6	-10.13
Spark Duration (degs)	5	5	5	5	5
FER	1	1	1	1	1
Conrod Length (cm)	13.3	13.3	13.3	13.3	13.3
Temperature (K)	300	300	300	300	300
Pressure (bar)	1.046	1.046	1.046	1.046	1.046

3.2.2 Sigma Iso-Octane Simulation Validation

Graph 5 (right subplot) presents simulation results for 1000–5000 rpm, illustrating refined ignition timing and enabling comparison with actual engine pressure data (left subplot). Initial observations show significantly lower simulation pressures, suspectedly due to simplistic piston profile, reduced compression ratio (in CFD model) and sector geometry. Notably, physical engine testing was conducted at a lambda of 0.9, whereas the simulation used a value of 1. Although some values differ by nearly double, the trendlines remain relatively consistent.

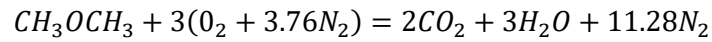
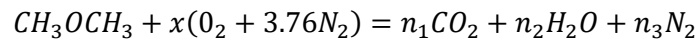


Graph 5: Sigma Engine Pressure Data Comparison to Iso-Octane ANSYS Forte Simulation Pressure Results After Some Ignition Timing Refinement.

3.3 Sigma Dimethyl Ether Spark Ignition

DME was the next stage of testing where engine geometry was unchanged (see figure 8). However, because DME is a different chemical compound, its air-fuel ratio was recalculated.

Prior to entering mixture composition in initial region parameters, stoichiometric combustion of DME (CH_3OCH_3) is balanced in a MATLAB script and demonstrated in equations below:



The balanced chemical equation of stoichiometric combustion for DME can be used to calculate the air-fuel ratio by mass. The mixture composition can then be calculated in terms of mass fractions for entry into Ansys Forte, this is detailed in the equations below and demonstrated in figure 10. The pressure data results for 1000-5000 RPM are detailed in graph 6.

Calculation of compound masses:

$$\text{CH}_3\text{OCH}_3 \quad (2 \times 12.011) + (6 \times 1.00784) + 15.999 = 46.06804$$

$$\text{O}_2 \quad 3 \times (2 \times 15.999) = 95.994$$

$$\text{N}_2 \quad 3 \times (3.76 \times 2 \times 14.0067) = 315.99115$$

$$\text{O}_2 \text{ \& } \text{N}_2 \quad 95.994 + 315.99115 = 411.985152$$

$$\text{O}_2 \text{ \& } \text{N}_2 \text{ \& } \text{CH}_3\text{OCH}_3 \quad 411.985152 + 46.06804 = 458.053192$$

$$\text{Air-Fuel Ratio} \quad \frac{458.053192}{46.06804} = 8.94297$$

Calculations of mass fractions:

$$\text{CH}_3\text{OCH}_3 \quad \frac{46.06804}{458.053192} = 0.1005735596$$

$$\text{O}_2 \quad \frac{95.994}{458.053192} = 0.2095695471$$

$$\text{N}_2 \quad \frac{315.99115}{458.053192} = 0.6898568889$$

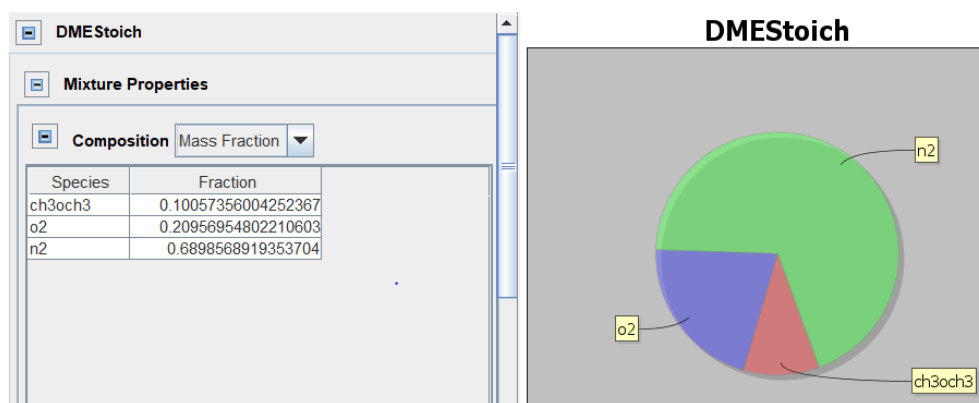
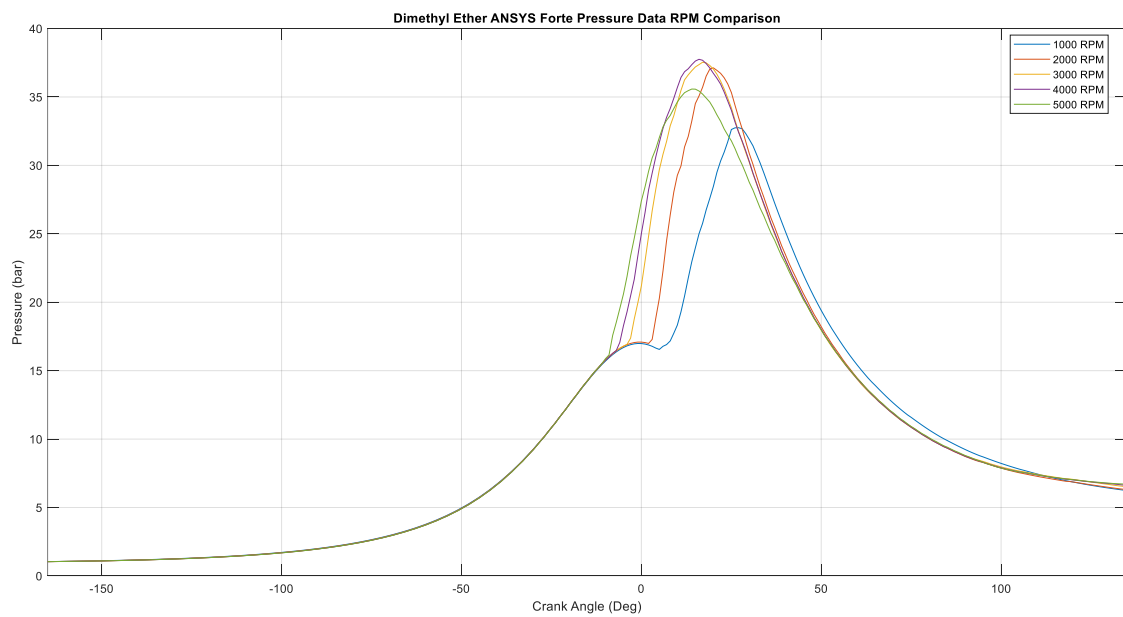


Figure 10: Stoichiometric Mixture Composition for DME (Mass Fraction).

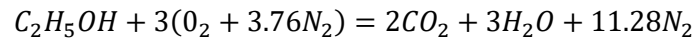
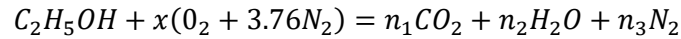


Graph 6: DME ANSYS Forte Simulation Results for 60 Degree Sector Mesh.

3.4 Sigma Ethanol Spark Ignition

The next aspect of testing was conducted for stoichiometric combustion of ethanol using Sigma geometry. However, ethanol and DME have the same elemental composition: 2 carbon atoms, 6 hydrogen and 1 oxygen. Whilst the element quantity and types are the same, there are many differences between these fuels, most significant is that DME is an ether and ethanol an alcohol (Parsons, 2013).

As ethanol (C_2H_5OH) has the same element quantities and types as DME, the balanced equation and air-fuel ratio are calculated to be equal to DME, demonstrated in equations below and figure 11.



Calculation of compound masses:

$$C_2H_5OH \quad (2 \times 12.011) + (6 \times 1.00784) + 15.999 = 46.06804$$

$$O_2 \quad 3 \times (2 \times 15.999) = 95.994$$

$$N_2 \quad 3 \times (3.76 \times 2 \times 14.0067) = 315.99115$$

$$O_2 \text{ \& } N_2 \quad 95.994 + 315.99115 = 411.985152$$

$$O_2 \text{ \& } N_2 \text{ \& } C_2H_5OH \quad 411.985152 + 46.06804 = 458.053192$$

$$\text{Air-Fuel Ratio} \quad \frac{458.053192}{46.06804} = 8.94297$$

Calculations of mass fractions:

$$C_2H_5OH \quad \frac{46.06804}{458.053192} = 0.1005735596$$

$$O_2 \quad \frac{95.994}{458.053192} = 0.2095695471$$

$$N_2 \quad \frac{315.99115}{458.053192} = 0.6898568889$$

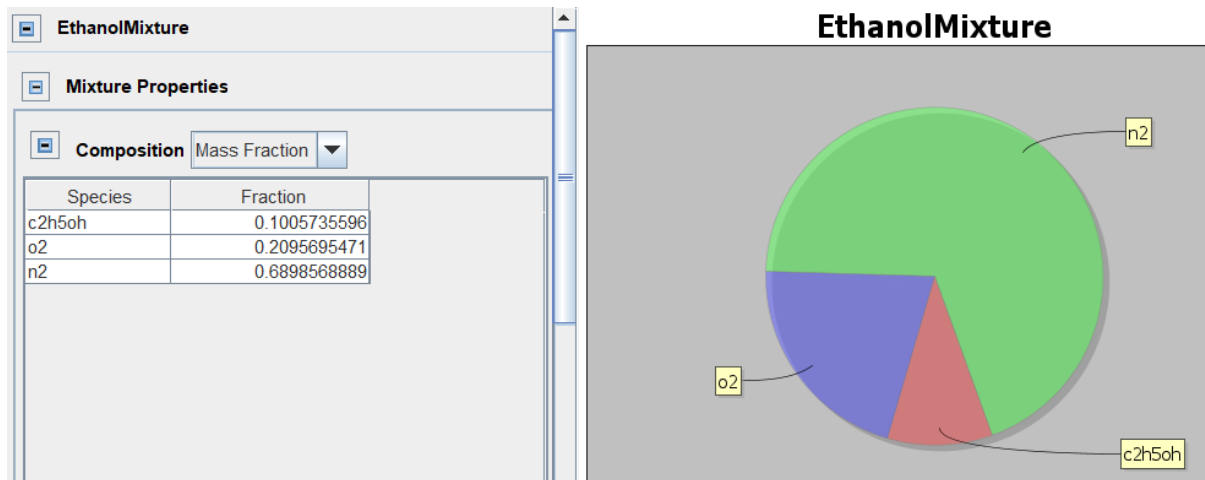
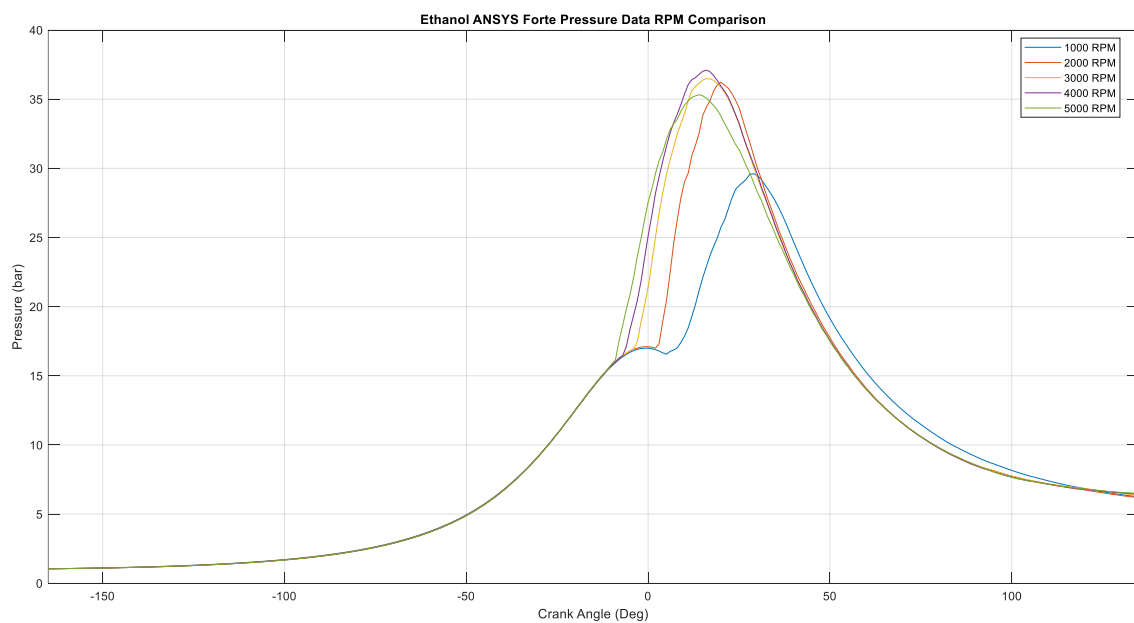


Figure 11: Stoichiometric Mixture Composition for Ethanol (Mass Fraction).

Ethanol simulation results are demonstrated in graph 7, detailing cylinder pressure for 1000-5000 RPM between -165 to 135 degrees in crank angle rotation. The peak cylinder pressure observed for ethanol is at 4000 RPM. Further analysis is detailed in a later section comparing the different fuels and results.

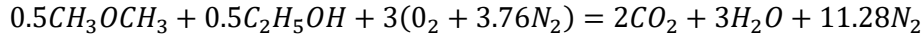
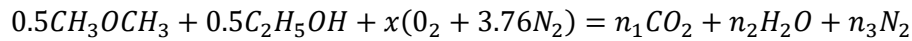


Graph 7: Ethanol ANSYS Forte Simulation Results for 60 Degree Sector Mesh.

3.5 Sigma Dimethyl Ether and Ethanol Blend Spark Ignition

DME as a standalone fuel yields cleaner combustion, reduced particulate matter and significantly less soot than fossil fuels (Singh, et al., 2021). However, its higher averaged combustion temperatures can increase NO_x formation (Leblanc, et al., 2020). In contrast, ethanol combustion produces lower cylinder temperatures than DME and petrol, thereby reducing NO_x. Similar to DME cleaner combustion also reduces CO, HC, and particle counts (Harrington, et al., 2022). These findings justify a DME-ethanol blend, with ethanol reducing NO_x, CO, and HC emissions and DME minimizing soot and particulate matter. Furthermore, Leblanc, *et al.* (2020) report that a lambda value of 1.2 significantly reduces flame temperature and NO_x production for DME.

Stoichiometric combustion of a 50-50 blend between DME (CH₃OCH₃) and ethanol (C₂H₅OH) is balanced in a MATLAB script where air-fuel ratio is determined to calculate compound mixture mass fractions (figure 12). Calculations are demonstrated in equations below with pressure results (graph 8):



Calculation of compound masses:

$$CH_3OCH_3 \text{ \& } C_2H_5OH \quad (2 \times 12.011) + (6 \times 1.00784) + 15.999 = 46.06804$$

$$O_2 \quad 3 \times (2 \times 15.999) = 95.994$$

$$N_2 \quad 3 \times (3.76 \times 2 \times 14.0067) = 315.99115$$

$$O_2 \text{ \& } N_2 \quad 95.994 + 315.99115 = 411.985152$$

$$O_2, N_2, CH_3OCH_3, C_2H_5OH \quad 411.985152 + 46.06804 = 458.053192$$

$$\text{Air-Fuel Ratio} \quad \frac{458.053192}{46.06804} = 8.94297$$

Calculations of mass fractions:

$$\text{CH}_3\text{OCH}_3 \quad \frac{23.03402}{458.053192} = 0.050286779793$$

$$\text{C}_2\text{H}_5\text{OH} \quad \frac{23.03402}{458.053192} = 0.050286779793$$

$$\text{O}_2 \quad \frac{95.994}{458.053192} = 0.2095695471$$

$$\text{N}_2 \quad \frac{315.99115}{458.053192} = 0.6898568889$$

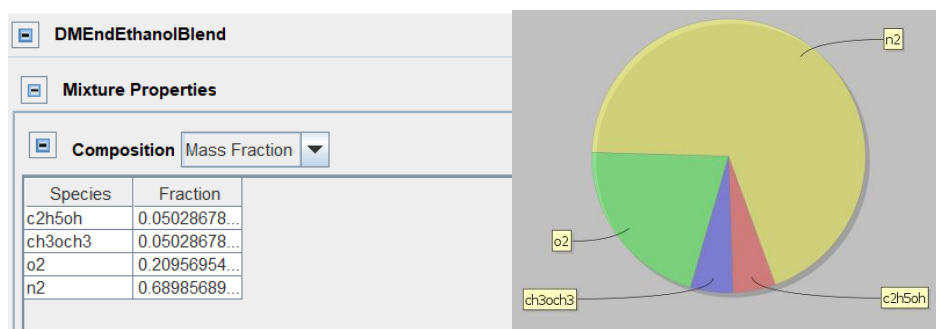
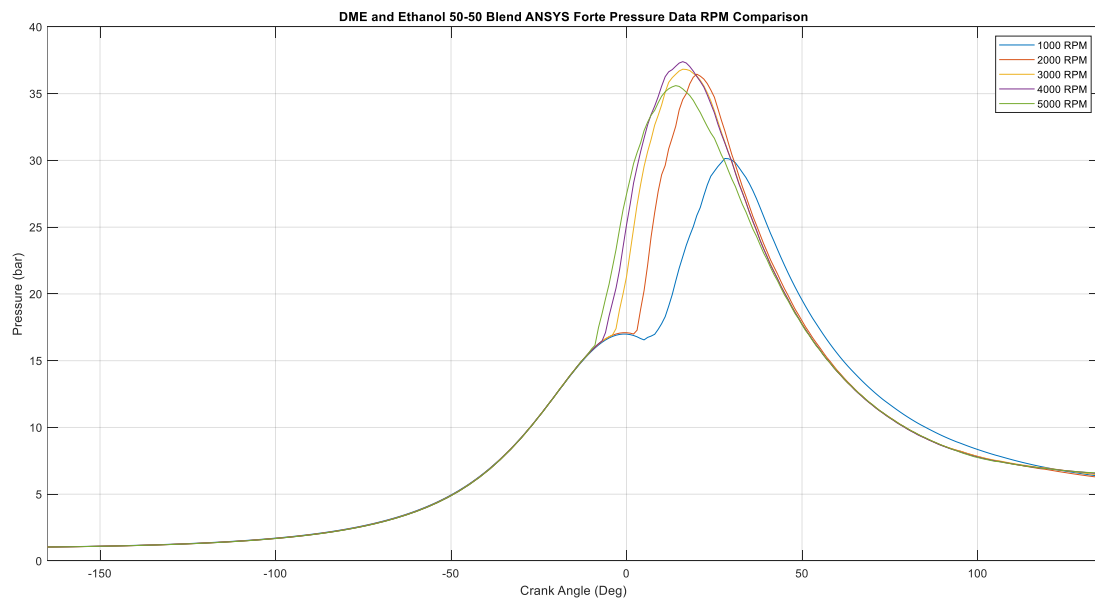


Figure 12: Stoichiometric Mixture Composition for 50-50 DME and Ethanol Blend (Mass Fraction).

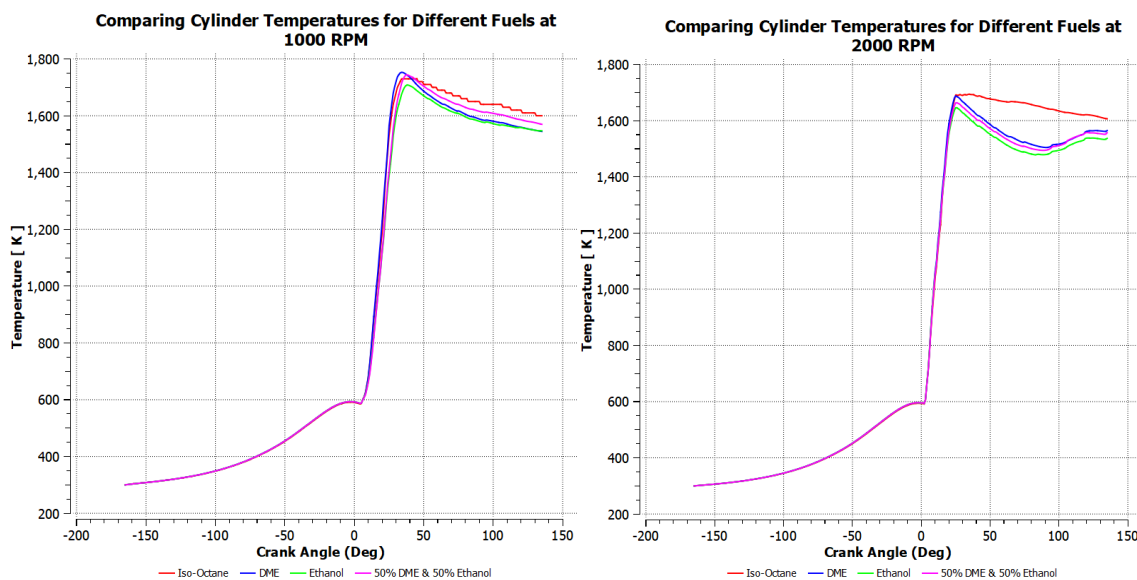


Graph 8: DME and Ethanol Blend (50-50) ANSYS Forte Simulation Results for 60 Degree Sector Mesh.

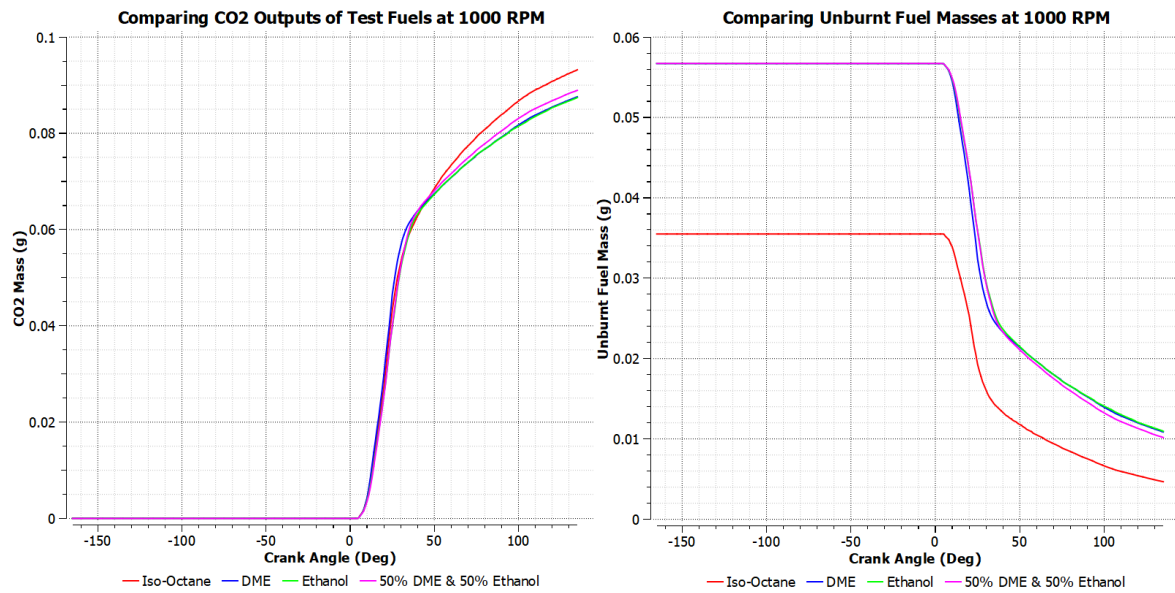
3.6 Forte Results Analysis

When modelling ethanol and DME combustion, a generic chemistry model (“fuel_library.cks”) was used. This model did not simulate soot, particulate matter, NO_x, or CO. Although most aspects of Sigma geometry are utilized, bowl and dome profiles are simplified to lower compression ratio and prevent auto-ignition and knock, particularly for DME.

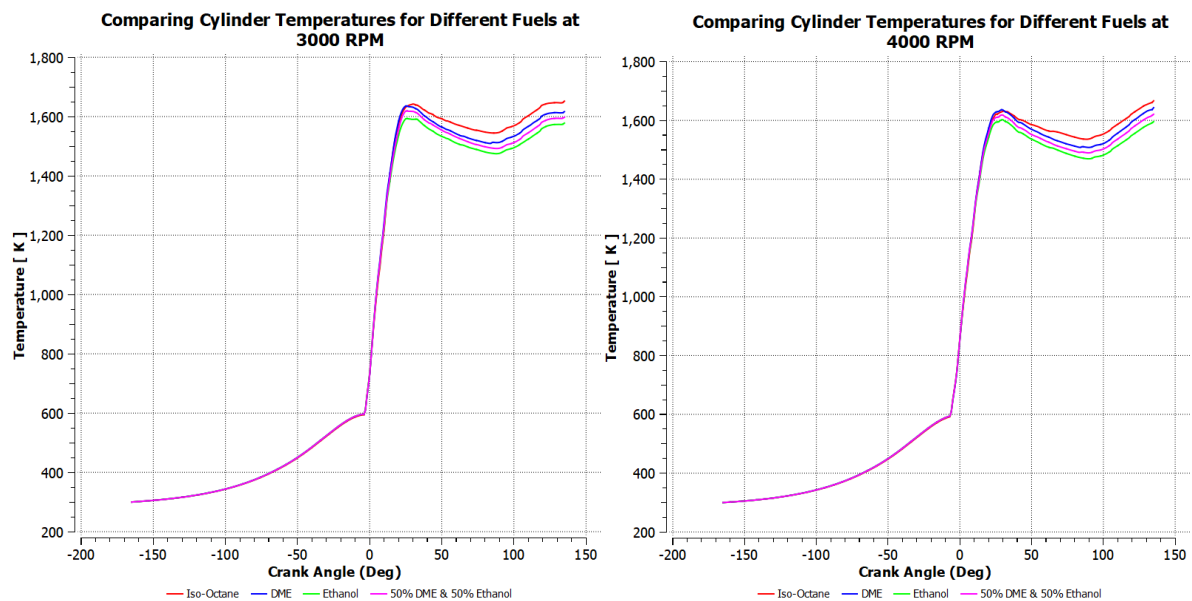
Graph 9 presents temperature results for each fuel tested in ANSYS Forte at 1000 and 2000 RPM. DME exhibits the highest initial flame front temperature, indicated by its steep gradient; this then drops rapidly after combustion of 55-60% of the initial air-fuel mixture. This is attributed to DME accumulating in crevice regions during compression where ignition is inhibited (Agarwal, *et al.*, 2023), thereby reducing emissions (Graph 10). The ethanol–DME blend benefits from a lower peak combustion chamber temperature, which helps reduce NO_x formation. Agarwal, *et al.* (2023) also observed reduced cylinder pressure compared to diesel. These results largely validate the simulation pressure data in Graphs 5–8, except for slightly higher peak pressures at 1000 and 2000 RPM. Additionally, Graphs 9, 11, and 12 indicate a progressive reduction in maximum in-cylinder temperature with increasing engine speed, which Jeon, *et al.* (2014) attribute to incomplete combustion.



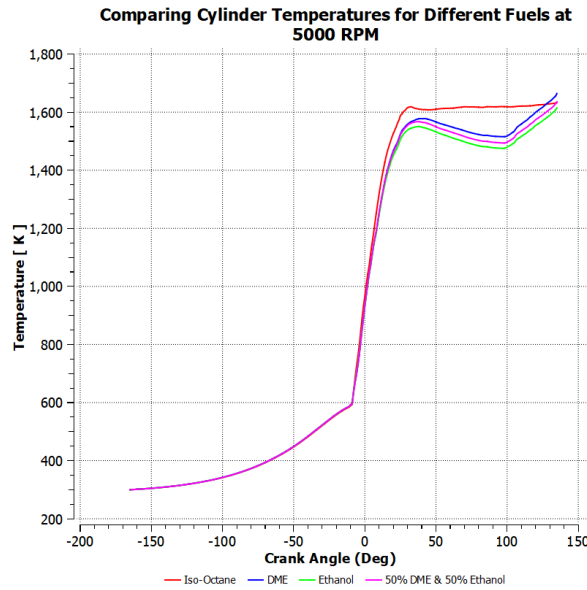
Graph 9: Plots Comparing In-Cylinder Maximum Temperatures for Alternative Fuels at (Left) 1000 RPM and (Right) 2000 RPM.



Graph 10: Comparing Test Fuels (Left) Carbon Dioxide Mass Production at 1000 RPM, and (Right) Mass of Unburnt Fuel for each Test Fuel at 1000RPM.



Graph 11: Plots Comparing In-Cylinder Maximum Temperatures for Alternative Fuels at (Left) 3000 RPM and (Right) 4000 RPM.



Graph 12: Comparing In-Cylinder Maximum Temperatures for Alternative Fuels at 5000 RPM.

Flame propagation at 1000 RPM for each fuel is shown in Figures 13, 14, and 15, highlighting differences in temperature and propagation speed. DME combusts most rapidly to begin with and at higher temperatures than the other fuels, especially between 10° and 25° crank angle. Similar effects are observed in the ethanol-DME blend, where temperatures increase relative to standalone ethanol.

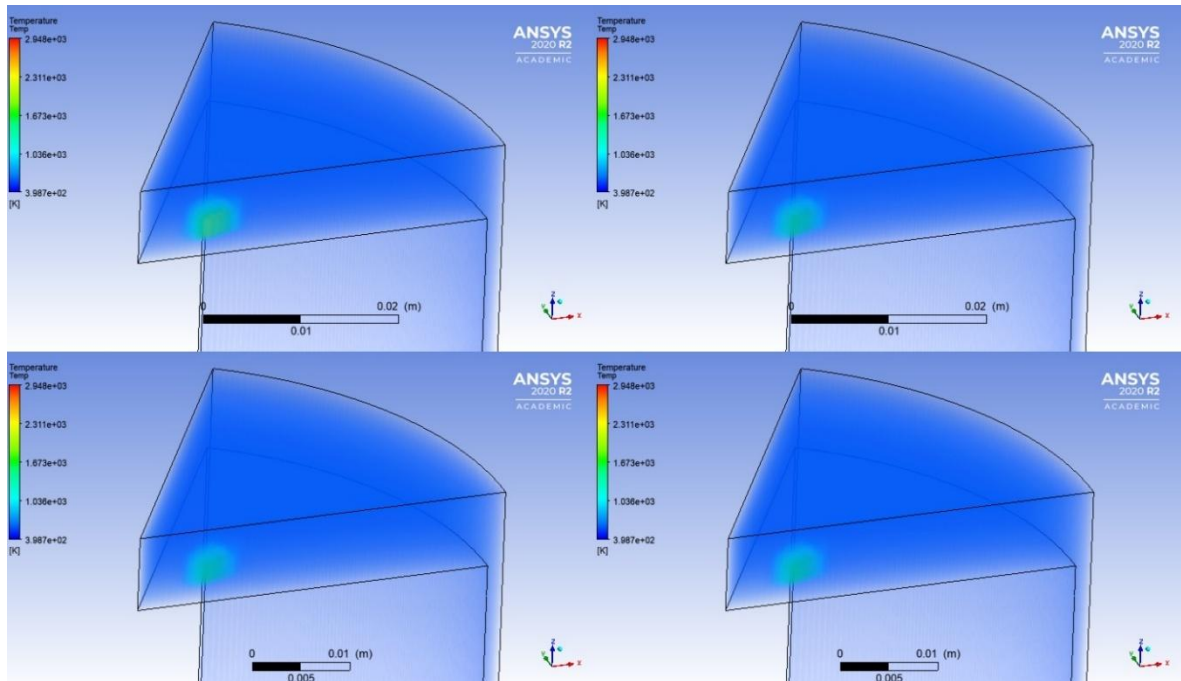


Figure 13: Piston Sector Temperatures at 5 Degrees Crank Angle and 1000 rpm: (Top Left) Iso-Octane, (Top Right) DME, (Bottom Left) Ethanol, (Bottom Right) Ethanol and DME Blend.

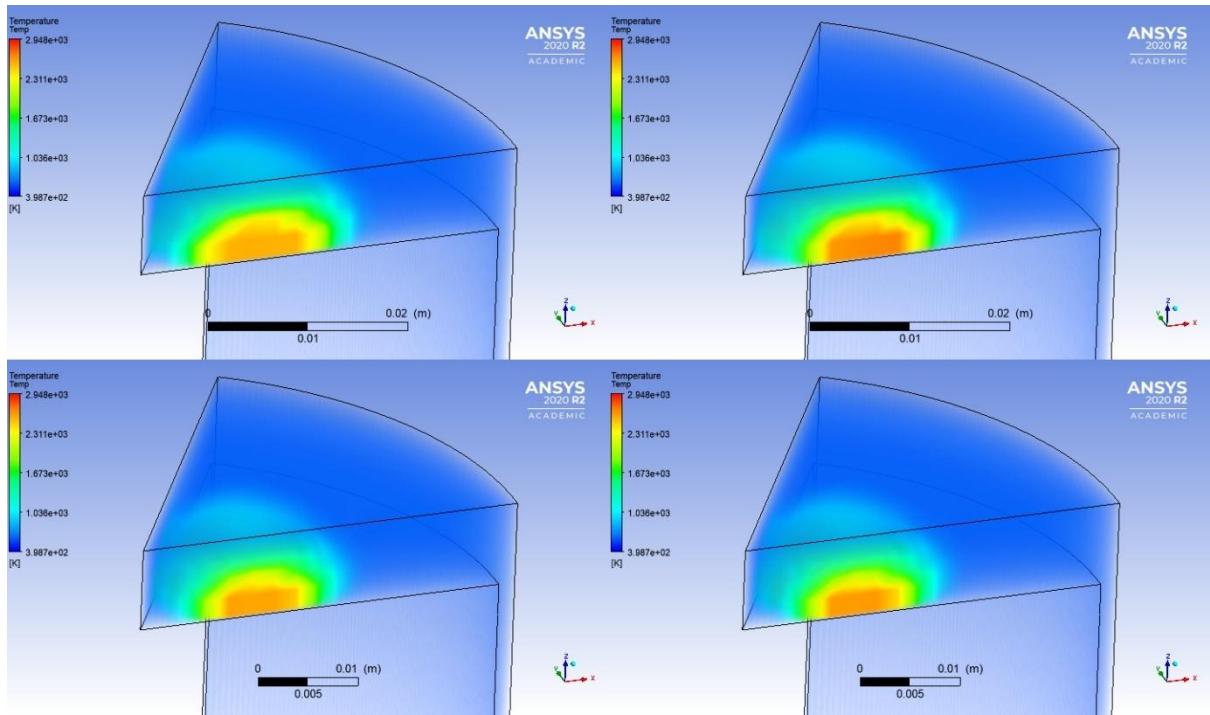


Figure 14: Piston Sector Temperatures at 10 Degrees Crank Angle and 1000 rpm: (Top Left) Iso-Octane, (Top Right) DME, (Bottom Left) Ethanol, (Bottom Right) Ethanol and DME Blend.

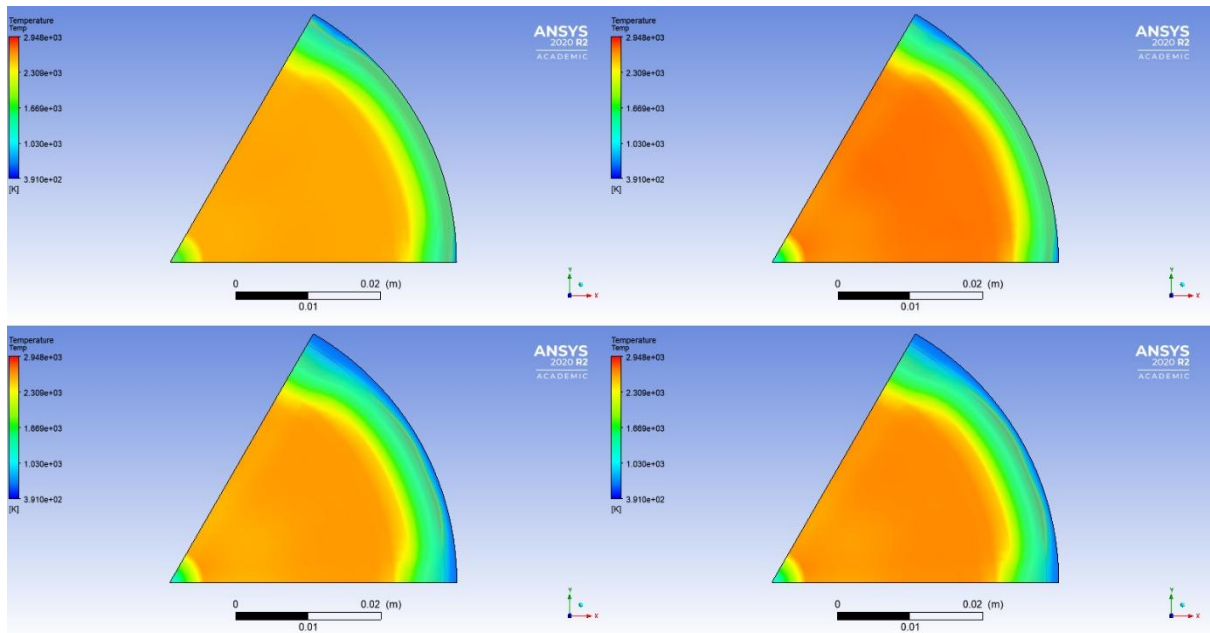


Figure 15: Piston Sector Temperatures at 25.02 Degrees Crank Angle and 1000 rpm: (Top Left) Iso-Octane, (Top Right) DME, (Bottom Left) Ethanol, (Bottom Right) Ethanol and DME Blend.

3.7 Assumptions

During Forte testing, assumptions have been made based on computational abilities and geometry data. This section details some of the key assumptions made within CFD testing.

3.7.1 Chemistry Files

ANSYS Forte uses Chemkin chemistry sets for its combustion modelling. For certain fuels, particularly DME, a generic ‘fuel_library’ set was employed. This limited data does not model CO, NO_x, or soot, reducing the accuracy of emission predictions. However, validation showed that the flame temperature and propagation characteristics aligned with published experimental results, thereby confirming trendline precision of the combustion simulation.

3.7.2 Sector Geometry and Mesh

Due to computer processing limitations, a 60° sector mesh was modelled. The model lacks an accurate cylinder head, valves, and intake ports, so it does not precisely simulate air-fuel intake. This simplification may significantly affect the air-fuel mixture accuracy, potentially leading to unstable combustion, flame propagation issues, or misfiring (Cheng, et al., 2024). However, because the cylinder element composition is predefined in the initialisation, these possible outcomes are not predicted. Additionally, neglecting accurate piston ring and crevice region details reduces combustion accuracy in these areas. Previous evidence indicates that the overall combustion response is modelled correctly in Forte simulations (Agarwal, et al., 2023). Different bowl profiles favour distinct engine designs and fuel types (Abdelrazek, et al., 2022); thus, a simplified bowl profile was employed as a control measure to isolate test fuel effects from engine architecture and to reduce the compression ratio below 10:1, lowering the risk of auto-ignition. Further work could investigate optimal profiles for each test fuel. Other CFD assumptions include the use of Reynolds-Averaged Navier-Stokes (RANS) for turbulence modelling which contains assumptions regarding flow characteristics (ANSYS Inc, 2024).

3.7.3 Fuel Supply and Injection Techniques

Although DME exhibits favourable characteristics in ICEs, significant modifications to fuel storage and delivery systems are required to accommodate its unique properties compared to conventional petrol and diesel. This aspect is not represented in ANSYS Forte but remains crucial when selecting an alternative fuel.

At atmospheric pressure (1.01325 bar), DME exists as a gas. To safely store and transport DME as a liquid, pressurised tanks (approximately 5 bar) must be used (De Ojeda & Wu, 2025). Figure 16 details the storage and injection system, which employs a high-pressure fuel pump connected to a high-pressure common rail and direct injector. The pump maintains DME in a liquid state and prevents wear on seals and components caused by its low viscosity and lubricity. The common rail and injector ensure the fuel is finely atomised into small droplets that vaporise rapidly, achieving a more homogeneous mixture.

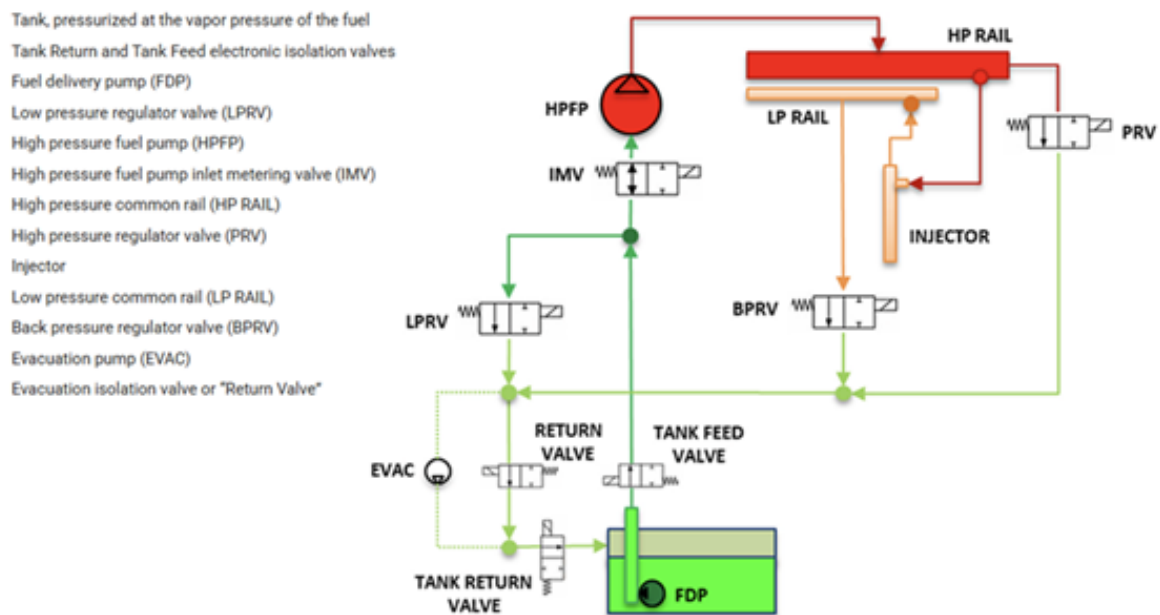


Figure 16: DME Fuel Storage and Injection System Schematic Overview for use in a Direct Injection Compression Ignition Engine (De Ojeda & Wu, 2025).

4 Ricardo Wave

Due to DME's high knock probability in SI engines, testing in Ricardo Wave included petrol (baseline), DME (to assess knock effects), and a 50-50 ethanol-DME blend (to reduce knock/prevent knock). Forte results revealed negligible knock, which is unexpected given DME's high cetane number (Singh, et al., 2021). The Wave model began as a simple single-cylinder SI engine running on indolene, later modified to incorporate heat transfer modelling and a complex EGR system controlled by a proportional-integral-derivative (PID) controller (Niekerk, 2020). The EGR system was evaluated across nine cases ranging from 0 to 24% EGR.

4.1 Model Development

An initial model was developed using the Ricardo Wave tutorial to simulate indolene combustion in a single-cylinder SI engine (Ricardo Software, 2019). Figure 17 demonstrates the single-cylinder geometry as entered into the model (Figure 18). The initial setup omitted heat transfer, EGR, and NO_x modelling. A simple injection profile was configured to inject fuel at -100 degrees (before top dead centre) with an air-fuel ratio of 14.5:1, as defined in the manual. Graph 13 displays in-cylinder pressure data at 1000 RPM, while Graph 14 shows temperature and indicated torque.

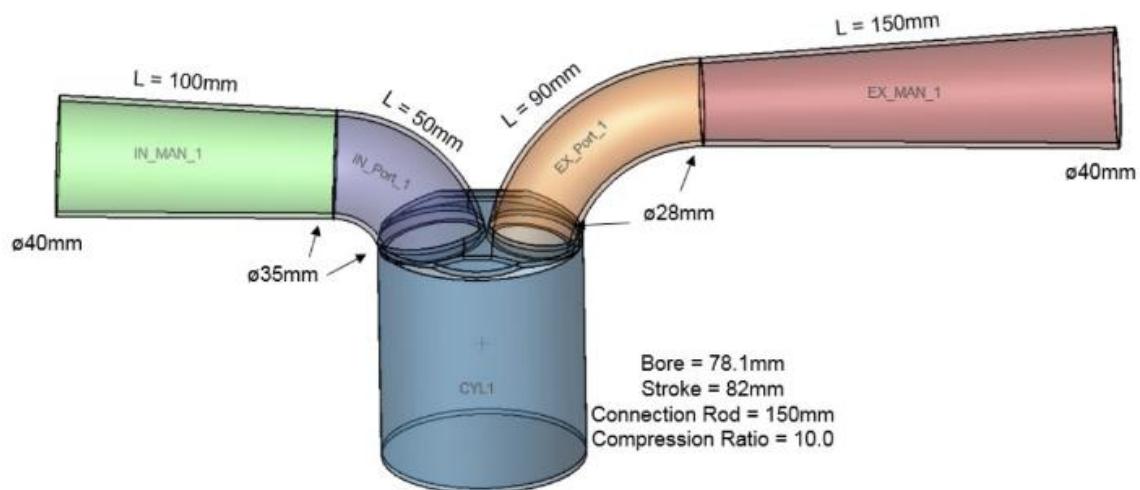


Figure 17: Single Cylinder SI Engine Geometry (Ricardo Software, 2019).

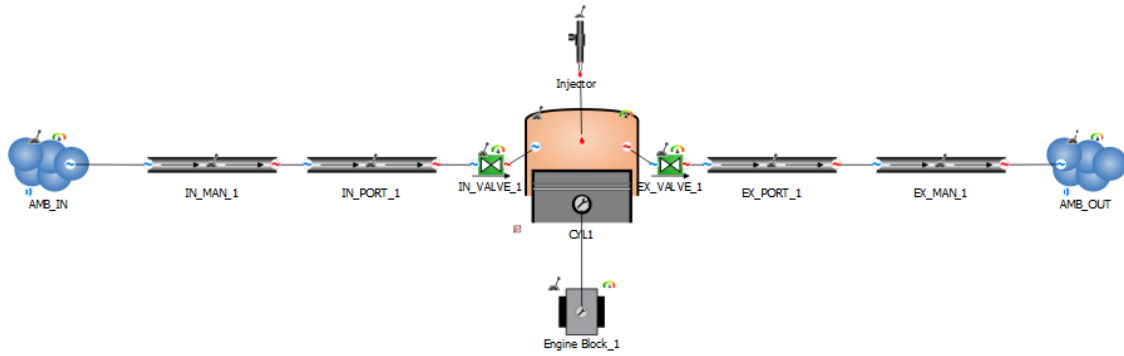
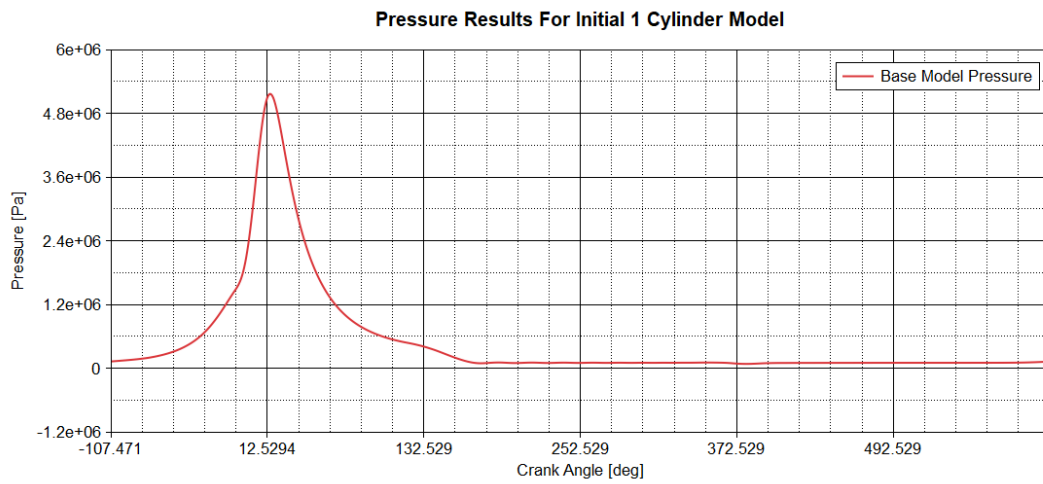
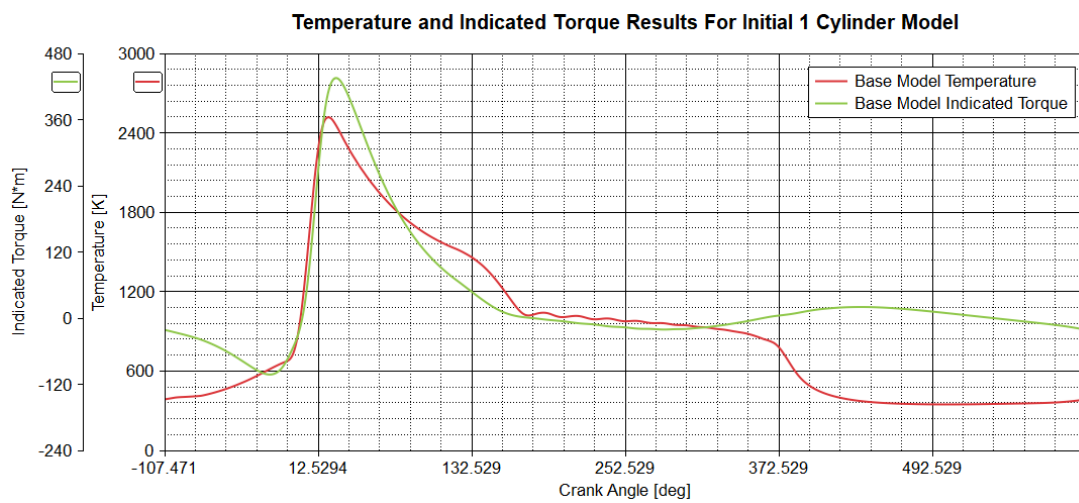


Figure 18: Basic Indolene Spark Ignition Engine Model in Ricardo Wave.



Graph 13: Ricardo Wave Basic Single Cylinder SI Engine Model Pressure Results.



Graph 14: Ricardo Wave Basic Single Cylinder SI Engine Model Temperature and Indicated Torque Results.

4.2 Heat Transfer and EGR

To facilitate complex emission studies and regulation, a PID controller, 30 mm orifice, two flow sensors, and ducts were added (Figure 19) to the previous model for testing varying EGR levels (Niekerk, 2020). To calibrate the PID controller's gain parameters, a sweep test was run in 'Wave Build', with the gain response shown in Figure 20. Heat transfer between the gas and cylinder walls is modelled using the 'Load Transfer Woschni' model; its equations are detailed in the user manual (Ricardo Software, 2019). The manual also specifies coolant and oil properties, with piston oil cooling defined as splash.

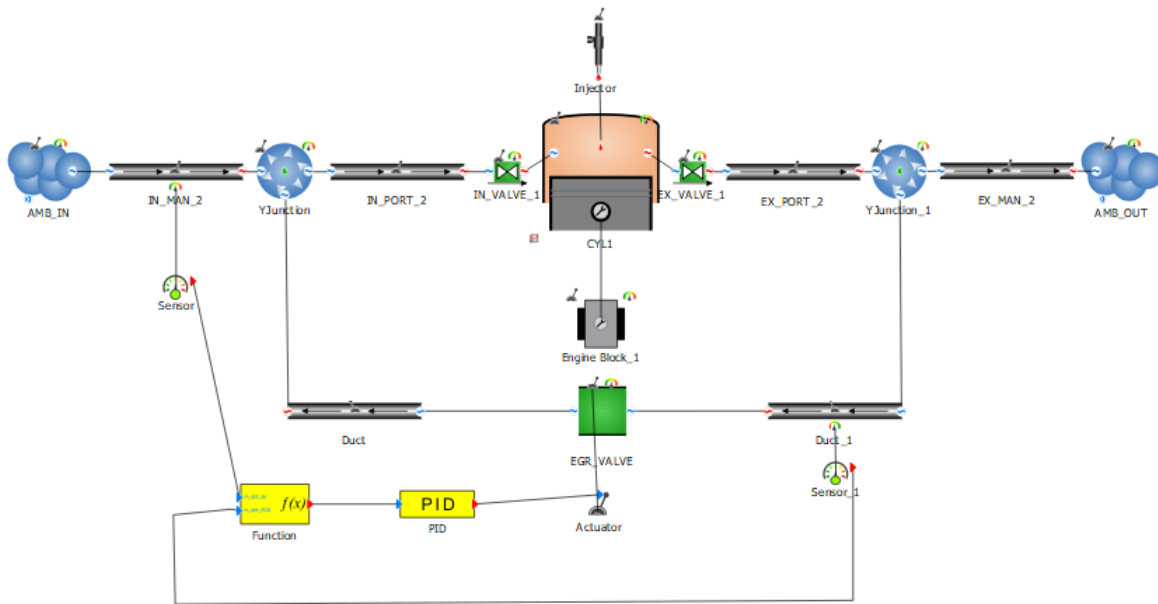


Figure 19: Single Cylinder Ricardo Wave Model with EGR System.

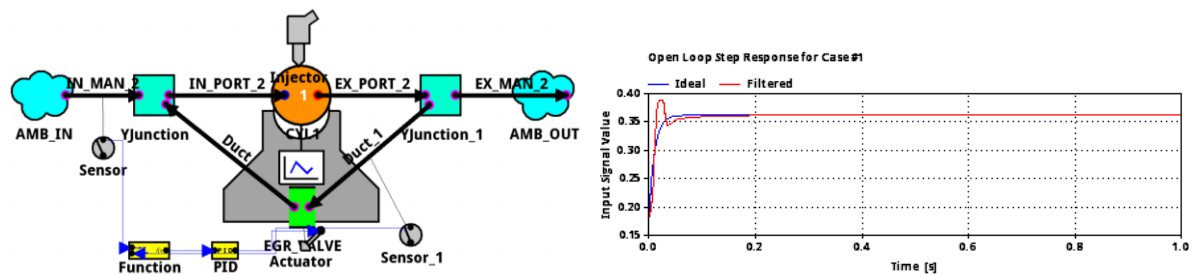
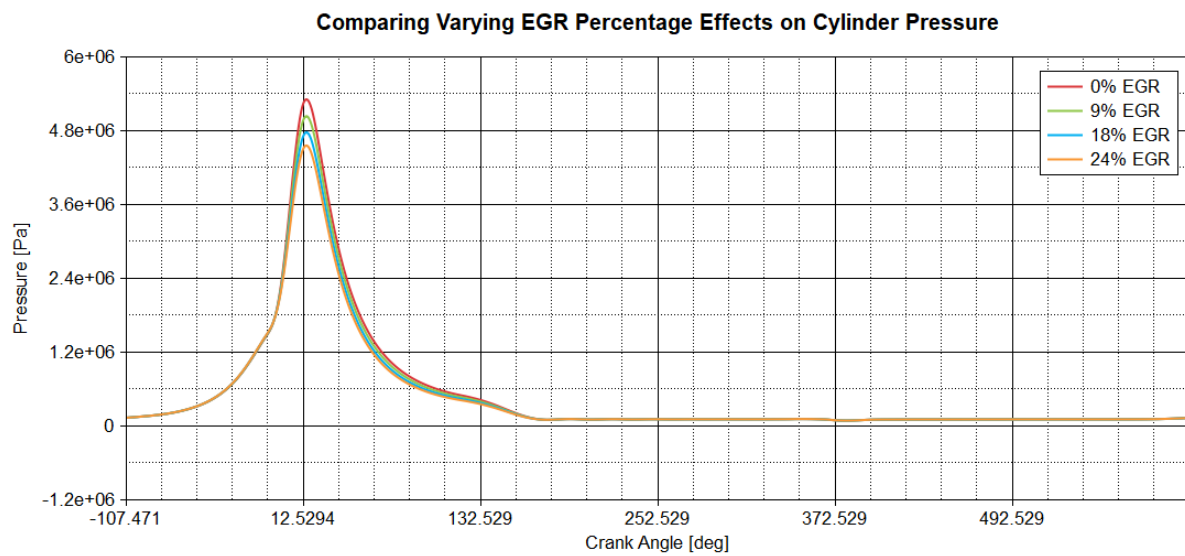
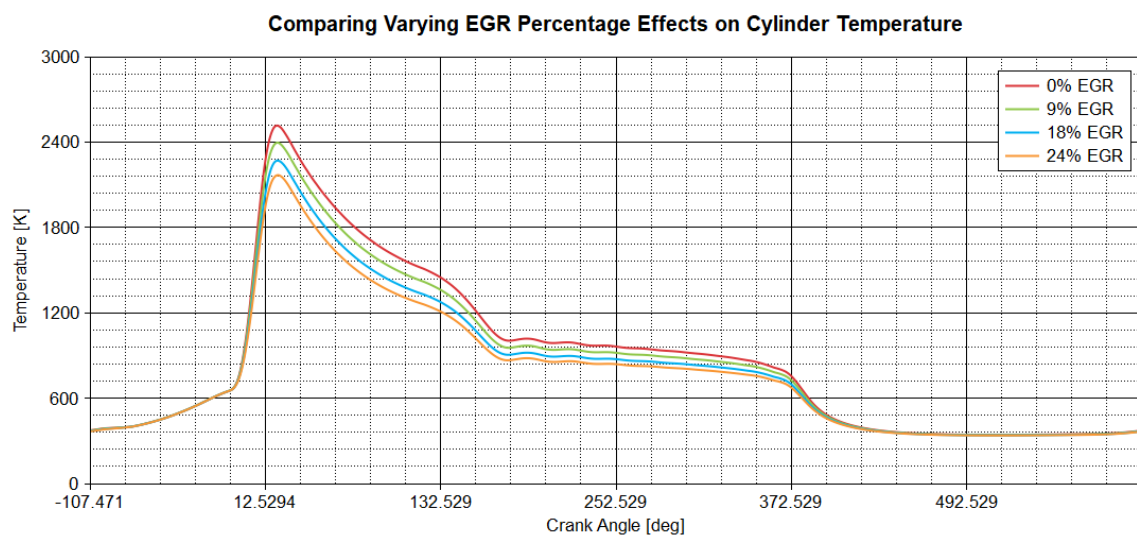


Figure 20: Calibration of PID Gain Parameters for EGR Control System, (Left) Wave Build Model, (Right) Controller Gains Open Loop Step Response Plot.

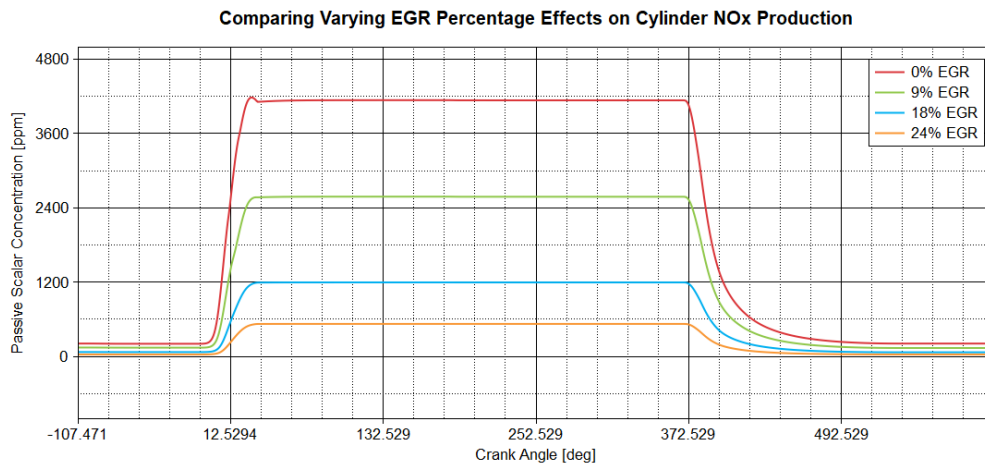
Graphs 15, 16, and 17 present the heat transfer and EGR model results at 1000 RPM. Pressure data indicates that as EGR percentage increases, cylinder pressure drops due to temperature reduction observed in Graph 16. This temperature decrease leads to a significant reduction in NO_x production. Increasing EGR from 0% to 24% resulted in an 87.4% drop in NO_x emissions, which is notably high compared to the approximately 80% reduction observed in diesel engines with 20% EGR at full load (Parashuram & Kumar, 2016). Considering this model is a SI engine, the extent of NO_x reduction appears to be overestimated.



Graph 15: EGR Effects on Cylinder Pressure for EGR and Heat Release Single Cylinder Model.



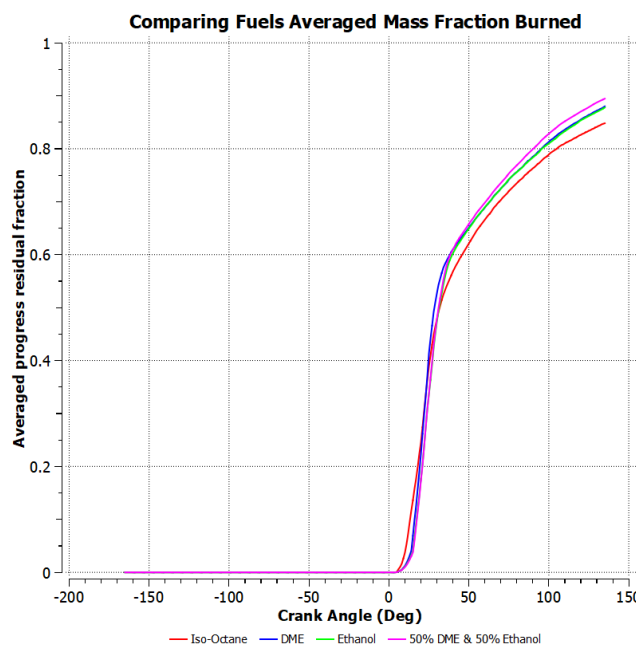
Graph 16: EGR Effects on Cylinder Temperature for EGR and Heat Release Single Cylinder Model.



Graph 17: EGR Effects on Cylinder NOx Production for EGR and Heat Release Single Cylinder Model.

4.3 Alternative Fuels Mass Fraction Burned Rates

Graph 18 presents the mass fraction burned results at 1000 RPM for each test fuel. Initial observations show that iso-octane burns a higher fraction initially but levels off sooner and at a lower overall value than the other fuels. DME and ethanol exhibit similar initial burn rates; however, near 60%, DME burns faster. The DME-ethanol blend burns more slowly during most of the process, yet its burn rate surpasses that of the other fuels in the final stage

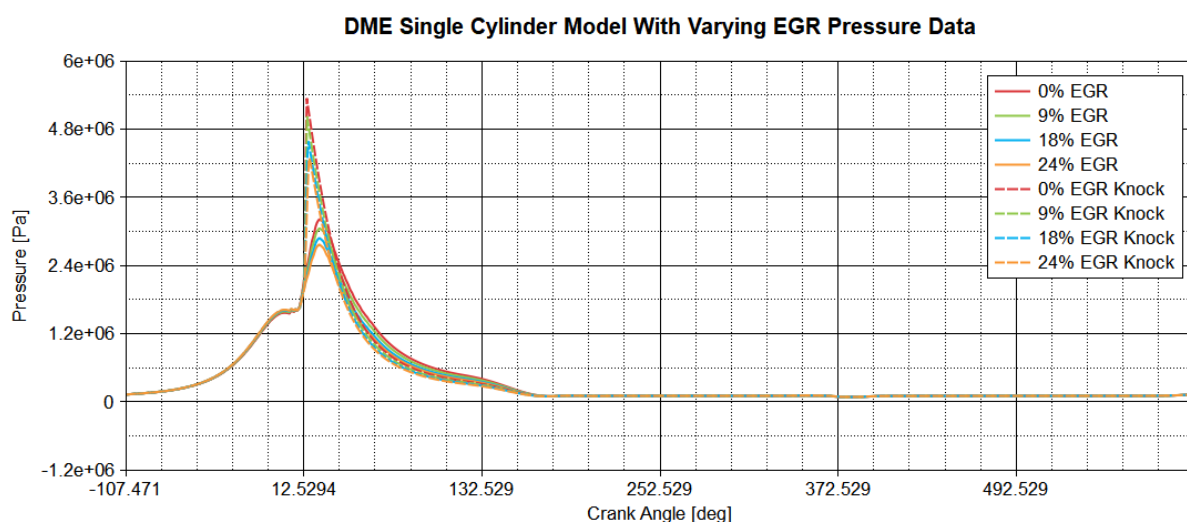


Graph 18: ANSYS Forte Mass Fraction Burned Rate Results for Each Fuel at 1000 RPM.

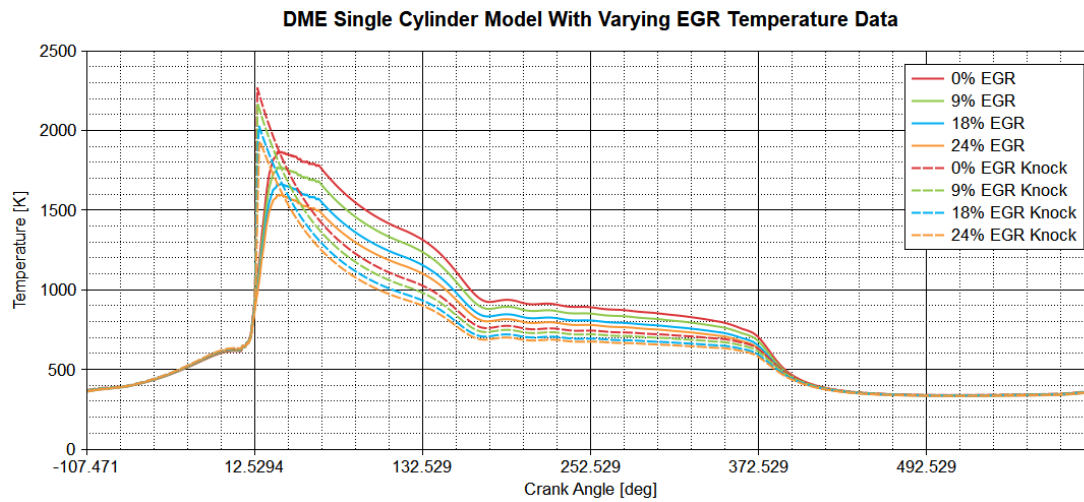
4.4 Alternative Fuels in Wave Results

The finalised previous model was then used to test DME; as predetermined, DME has a lower air–fuel ratio (8.94) than iso-octane and indolene. With the new air–fuel ratio and ANSYS Forte mass fraction burn profile from Graph 18 integrated into the Wave setup, testing was again conducted at 1000 RPM. Due to DME’s high cetane number, concerns arise regarding auto-ignition in an SI engine’s combustion chamber. To assess this, a baseline test was performed without knock modelling, followed by a run with knock modelling. As predicted, significant knock occurred, with a sudden spike in pressure and temperature at 12.5° crank angle detailed in Graphs 19 and 20, indicating unacceptable autoignition levels.

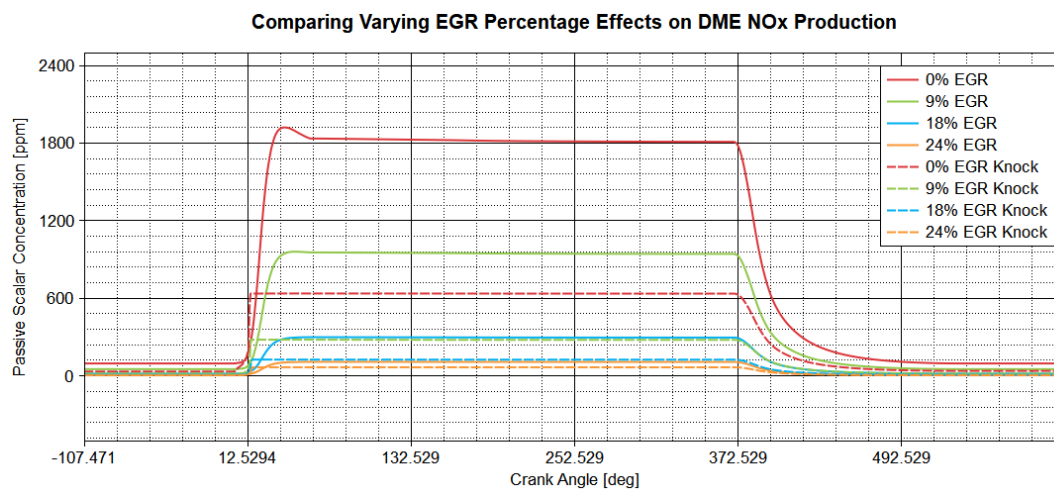
The quantity of NO_x produced by DME (graph 21) is seen to be significantly lower for DME when compared to indolene. The knock NO_x results are questionable in validity where 0% EGR with knock is significantly lower than without the knock model. The lower NO_x production for DME is due to reduced cylinder pressure and temperature compared to indolene. The cause of this is suspected too be due to differences in ignition timing where DME is set to 3 degrees start of combustion at 1000 rpm and indolene -12 degrees.



Graph 19: Pressure Data Results for Single Cylinder SI Modelling of DME with and without Knock Modelling and Variable EGR Percentages.

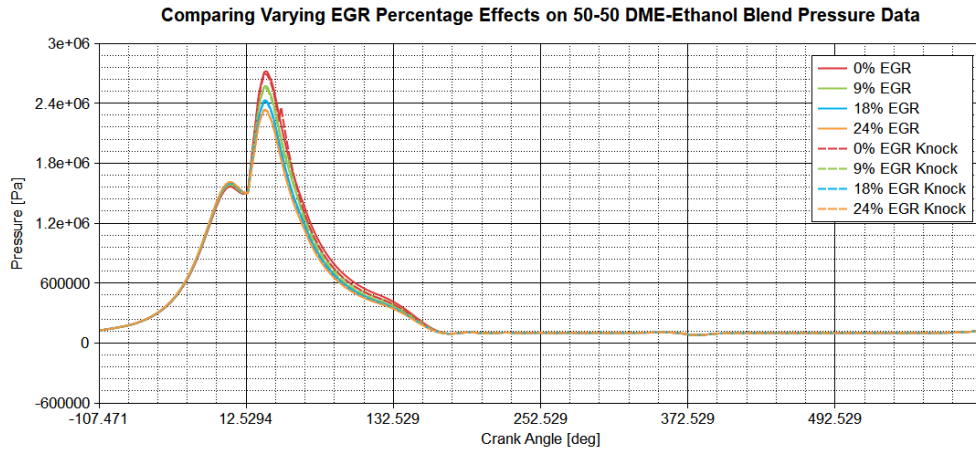


Graph 20: Temperature Data Results for Single Cylinder SI Modelling of DME with and without Knock Modelling and Variable EGR Percentages.

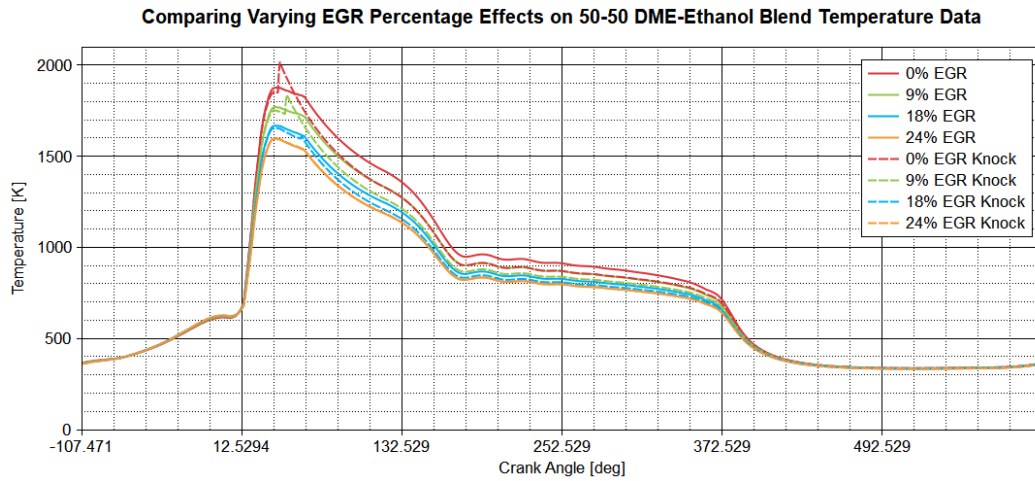


Graph 21: NOx Production Results for Single Cylinder SI Modelling of DME with and without Knock Modelling and Variable EGR Percentages.

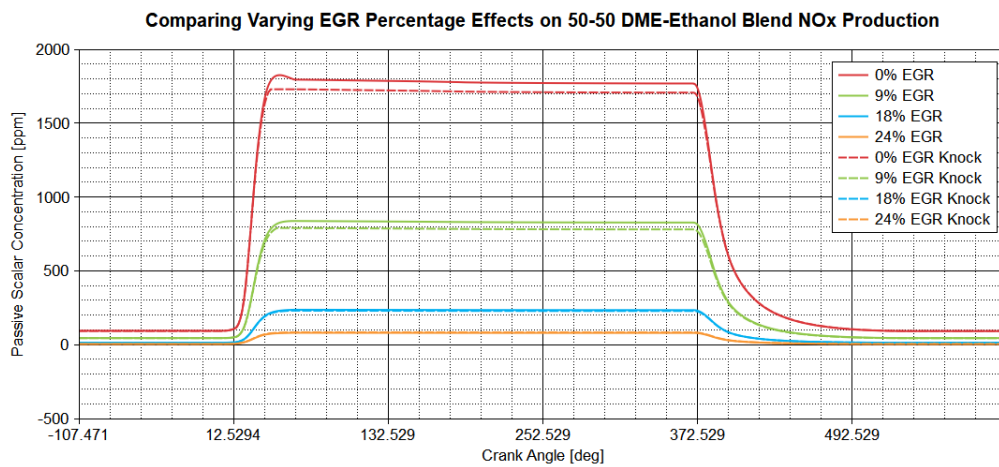
Due to significant knock during DME combustion in SI engines, a 50-50 DME-ethanol blend was tested to leverage ethanol's higher octane rating. Graph 22 shows the pressure data for this blend, indicating that increasing EGR reduces flame temperature sufficiently to prevent knock (Szybist, et al., 2017). Knock gradually diminishes until it is entirely eliminated between 18% and 24% EGR (Graph 23). Additionally, NOx levels are slightly lower (Graph 24 versus Graph 21), and while carbon monoxide emissions initially spike during knock, they drop by over 50% at higher EGR levels (Graph 25).



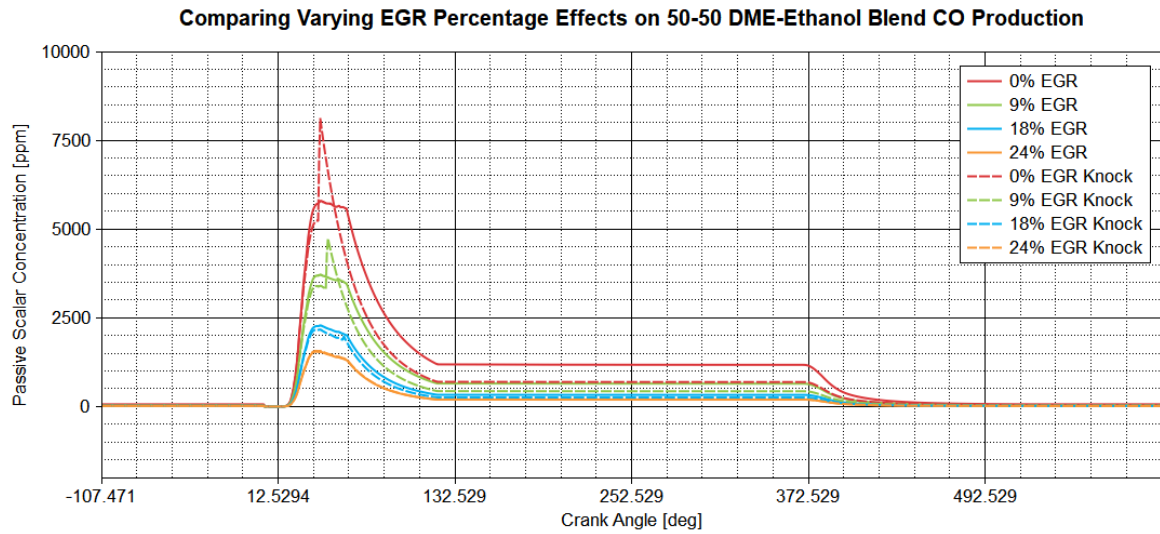
Graph 22: Pressure Data Results for Single Cylinder SI Modelling of 50-50 DME-Ethanol Blend with and without Knock Modelling and Variable EGR Percentages.



Graph 23: Pressure Data Results for Single Cylinder SI Modelling of 50-50 DME-Ethanol Blend with and without Knock Modelling and Variable EGR Percentages.

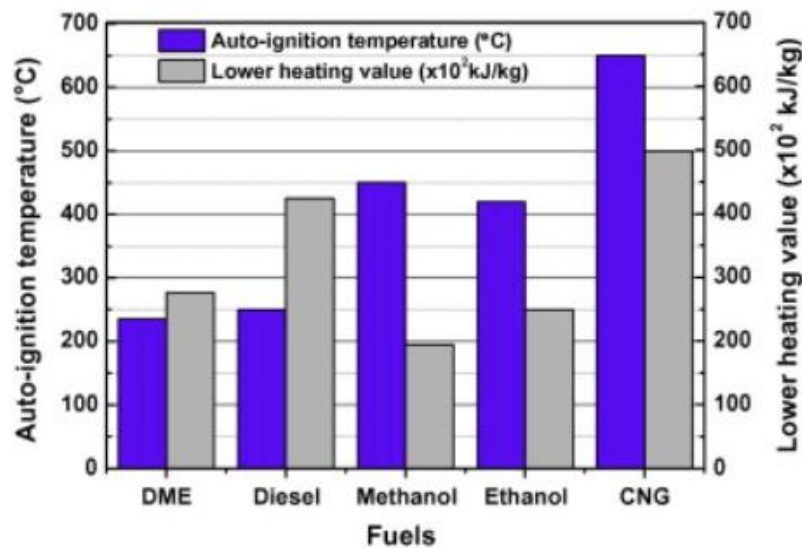


Graph 24: NOx Production Results for Single Cylinder SI Modelling of 50-50 DME-Ethanol Blend with and without Knock Modelling and Variable EGR Percentages.



Graph 25: CO Production Results for Single Cylinder SI Modelling of 50-50 DME-Ethanol Blend with and without Knock Modelling and Variable EGR Percentages.

Additional methods of reducing knocking include reductions to compression ratio and adjustments to inlet (an intercooler could be beneficial) and engine temperatures (Gupta, 2012). It is suggested that fuels with higher self-ignition temperatures are selected; whilst this is the case for ethanol (graph 26), DME has a lower self-ignition temperature of 200-250 degrees Celsius (Lee & Park, 2014). Investigations into blends with higher quantities of ethanol could prove better suited. Other knock reduction methods are detailed table 4.



Graph 26: Auto-Ignition Temperatures and Lower Heating Values of Different Fuels (Lee & Park, 2014).

Table 4: Reducing Knock in CI and SI Engines (Gupta, 2012).

<i>Factor</i>	<i>SI engines</i>	<i>CI engines</i>
Self-ignition temperature	High	Low
Time lag or delay period of fuel	Long	Short
Compression ratio	Low	High
Inlet temperature	Low	High
Inlet pressure	Low	High
Combustion chamber wall temperature	Low	High
Engine speed, rpm	High	Low
Cylinder size	Small	Large

4.5 Wave Analysis

The Ricardo Wave results above establish the baseline emissions of a simplistic SI engine running on indolene, setting a benchmark for alternative fuel testing. Simulations with DME and a 50-50 DME-ethanol blend indicate that emissions can be significantly reduced relative to conventional fossil fuels. However, the challenges posed by these fuels, particularly DME, necessitate modifying conventional SI engines. Both current results and supporting experiments identify DME's high cetane number as a cause of significant knocking and the primary obstacle in an SI engine (Singh, et al., 2021). Mitigation strategies, such as implementing 18–24% EGR, effectively reduce or eliminate knock while lowering NO_x and CO emissions (Gupta, 2012). Additionally, cylinder pressure and temperature were carefully regulated to avoid DME's auto-ignition threshold (Lee & Park, 2014). Blending ethanol further reduces NO_x and enhances engine stability, as its high auto-ignition temperature and octane rating support DME's use in SI engines. Although using DME in SI engines may seem unconventional, controlling ignition timing to prevent auto-ignition enables significant reductions in cylinder temperatures and NO_x. However, storage challenges associated with DME, and the related issues in fuel blending chemistry of DME and ethanol may render this blend strategy as impractical for large-scale automotive applications (De Ojeda & Wu, 2025). Therefore, further research should explore RCCI combustion with DME and ethanol injected separately, as DME's favourability for CI engines might yield even better results.

Conclusion

The studies of DME and ethanol as separate and blended fuels proved critical to understanding both individual and blended combustion characteristics and fuel properties within a SI engine. A DME-ethanol blend was found to enhance the properties of standalone DME where both ethanol and EGR significantly reduce and regulate knock, NO_x, and CO emissions in a spark ignition engine. Ethanol-induced temperature reductions help prevent DME auto-ignition, an issue observed with pure DME. Although the 50-50 blend performs well theoretically, practical challenges in storing DME and preparing the blend may limit its real-world suitability. Further research should consider testing ethanol and DME in an RCCI configuration to address these storage and manufacturing concerns. The assumptions made within simulation studies in this report can be significant; whilst trendlines and percentages of emission reduction are close to previous studies, the lack of testing surrounding such a unique blend prove challenging to truly validate data values. Beneficial to the theoretical aspect of this investigation would be a practical element conducting the same studies. However, due to the aims of this investigation being met theoretically, the final proposed strategy as a result is a: 50-50 DME-ethanol blend for a lower compression ratio (8:1), high EGR (20-24%), flat piston bowl geometry for the Sigma based SI engine.

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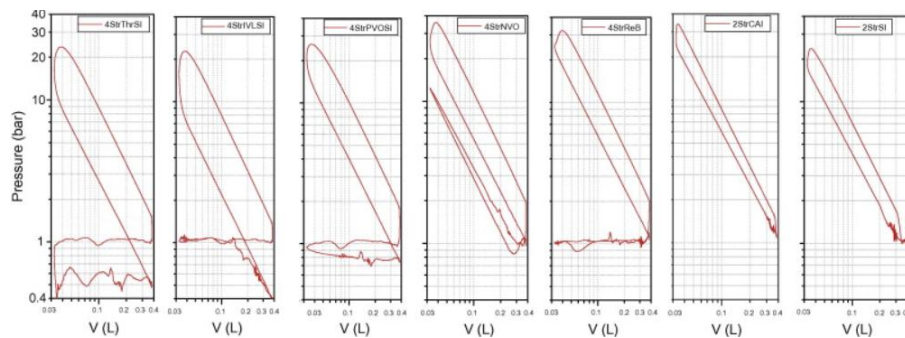
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Appendix A

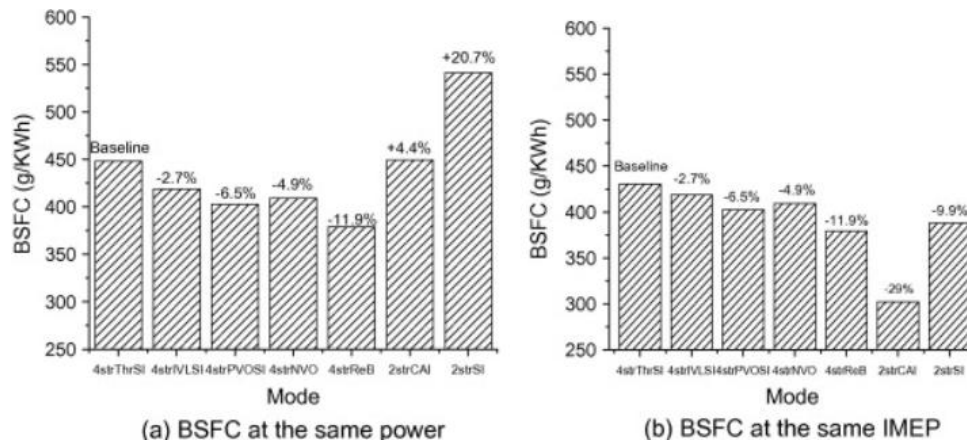
Internal combustion engines use various technologies to optimize performance and fuel efficiency. Depending on vehicle design and requirements, either 2-stroke or 4-stroke engines are employed. For example, cars typically use 4-stroke engines, while certain motorcycles, large diesel ship engines, and smaller outboard engines utilise 2-stroke engines for simplicity, low maintenance, and compactness (Dempsey, 2009). Due to higher fuel efficiency, lower emissions, and greater reliability, 4-stroke engines dominate the automotive and van industries. A study by Zhang & Zhao (2014) compared configurations of 4-stroke and 2-stroke engines, referring to these configurations as modes, detailed in Table 5 with pressure-volume plots in Graph 27. Graphs 28 and 29 indicate that the 2-stroke spark ignition engine has the lowest combustion efficiency, due to prolonged and incomplete combustion, while the 4-stroke exhaust rebreathing compression auto-ignition engine achieves the highest efficiency with shorter combustion durations. Regarding emissions, both 2-stroke configurations emit more carbon monoxide than 4-stroke engines, although NO_x production is higher in 4-stroke throttle-controlled spark ignition engines due to the absence of diluting exhaust gases. Even when downsized 2-stroke compression auto ignition engines offer benefits. However, the durability, lower fuel consumption, and reduced hydrocarbon emissions of 4-stroke engines make them the preferred choice for this application.

Table 5: Setup Configurations of a Study Comparing 2 and 4-Stroke Direct Injection Petrol Engines with either Spark Ignition, Controlled Auto Ignition and Homogeneous Charge Compression Ignition (Zhang & Zhao, 2014).

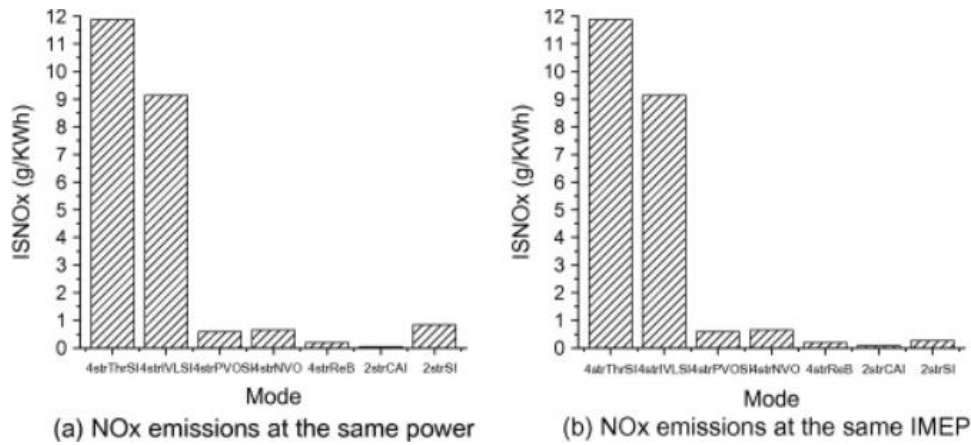
Mode Number	Number of Strokes	Description
Mode 1	4	Throttle controlled spark ignition engine.
Mode 2	4	Intake valve throttled spark ignition engine.
Mode 3	4	Positive valve overlap in spark ignition engine
Mode 4	4	Negative valve overlap in controlled auto ignition engine.
Mode 5	4	Exhaust rebreathing controlled auto ignition engine.
Mode 6	2	Controlled auto ignition engine.
Mode 7	2	Spark ignition engine.



Graph 27: Pressure-Volume Diagrams for Each Engine Mode (Zhang & Zhao, 2014).



Graph 28: Comparing Brake Specific Fuel Consumption of the 7 Engine Configurations (Zhang & Zhao, 2014).



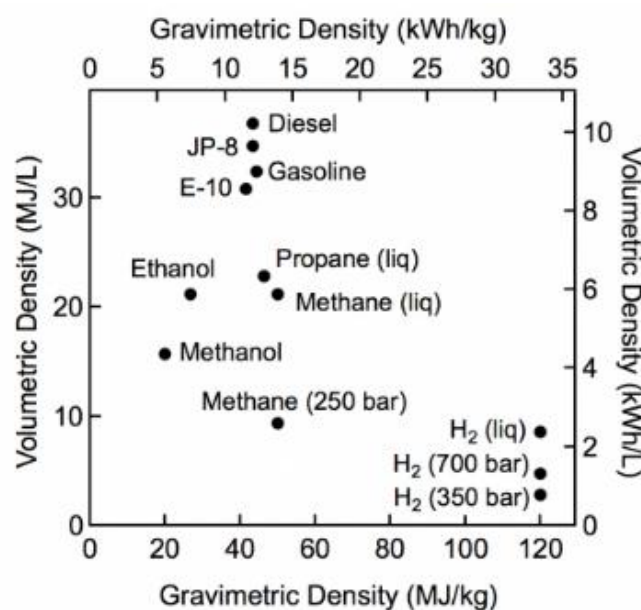
Graph 29: Comparing NOx Production of the 7 Engine Configurations (Zhang & Zhao, 2014).

Additional Engine Technology

Alternative combustion techniques include prechamber combustion systems (Alvarez, et al., 2018), premixed charge compression ignition (PCCI), and others. While each offers distinct advantages, current developments in HCCI and RCCI have proven capable of significantly reducing fuel consumption and emissions, though further testing and refinement are needed. Additional engine technologies designed to increase performance, improve fuel efficiency or reduce emissions include forced induction (turbochargers and superchargers), variable valve timing (VVT), variable valve lift (VVL), exhaust gas recirculation (EGR), variable compression ratio (VCR), among others (Gupta, et al., 2020). Utilizing one or a combination of these systems enhances performance and efficiency in internal combustion engines.

Appendix B

Hydrogen is a promising alternative to fossil fuels and alcohols. Its carbon-free nature eliminates hydrocarbon emissions such as carbon monoxide and carbon dioxide. However, storing hydrogen in everyday vehicles remains challenging. JCB is the first mainstream brand to receive licensing approval for hydrogen combustion engines in the agricultural and construction sectors (JCB, 2025), a development that may prompt further investigation by other manufacturers. Graph 30 shows that hydrogen's high gravimetric density enables leaner combustion compared to most alternative fuels (U.S. Department of Energy, n.d.). In contrast, its low volumetric density means that a tank of equal size stores significantly less hydrogen than petrol or diesel. Additionally, hydrogen's high adiabatic flame temperature can lead to increased NO_x formation, although leaner mixtures with EGR systems can mitigate this effect (Heffel, 2003).



Graph 30: Comparing Alternative Fuels Gravimetric and Volumetric Densities (U.S. Department of Energy, n.d.).