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Comparison of 1D and 3D Combustion Simulation Models
Masters Project ACFB7003

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September 2026

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Word count: 14,836

Abstract

This study compares 1D Ricardo WAVE and 3D ANSYS Forte simulations of a Suzuki GSX1100F spark-ignition engine using octane, methanol, ethanol and hydrogen over 1,000–10,000 rpm. The models were evaluated through combustion, cylinder pressure, temperature and engine-performance predictions. Results showed that model disagreement was dependent on both fuel and engine speed. For the liquid fuels, WAVE generally predicted increasingly delayed and prolonged combustion at higher speeds compared with Forte. Hydrogen showed a contrasting response, with WAVE predicting earlier combustion and higher peak pressure and temperature at high speed. This demonstrates that disagreement is not a fixed bias towards either model, but is influenced by fuel, combustion-model formulation and configuration. WAVE offers greater computational efficiency, while Forte provides greater spatial detail but requires more extensive setup. Without experimental validation or validated fuel-specific setup parameters, neither model can be considered definitively more accurate.

Declaration of authenticity

'This thesis is the result of the author's original research. It has been composed by the author and has not been previously submitted for examination which has led to the award of a degree.'

Signed:

A handwritten signature in black ink, appearing to read 'U. Broek', with a horizontal line underneath.

Dated: 06/09/2026

Acknowledgements

I would like to thank my supervisor, Dr Owen Williams, for their guidance and support throughout the course of this work; their advice and assistance were instrumental in helping me carry it out effectively.

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Nomenclature

Symbol	Definition	Unit
S_{lam}	Laminar flame speed	$m\ s^{-1}$
u_n	Hydrogen burning velocity	$m\ s^{-1}$
u_{n0}	Reference stretched laminar burning velocity	$m\ s^{-1}$
W	Laminar flame-speed correlation constant	$m\ s^{-1}$
E_M	Equivalence-ratio exponent	–
ζ	Equivalence-ratio multiplier	–
ϕ	Equivalence ratio	–
T_u	Unburned-gas temperature	K
T	Gas temperature	K
T_0	Reference temperature	K
P	Pressure	Pa
p	Pressure	Pa
p_0	Reference pressure	Pa
α	Temperature exponent	–
β	Pressure exponent	–
α_T	Hydrogen temperature exponent	–
β_p	Hydrogen pressure exponent	–
D_K	EGR-fraction multiplier	–
D_M	EGR-fraction exponent	–
r_0	Initial flame/kernel radius	mm
r_{min}	Minimum kernel radius	mm
C_{m1}	Kernel-to-G-equation switch constant	–
C_{m2}	Flame-development coefficient	–
C_{ms}	Flame-stretch coefficient	–
b_1	G-equation tuning coefficient	–
b_3	G-equation tuning coefficient	–
n	Polytropic index	–
V	Cylinder volume	m^3
t	Time	s
θ	Crank-angle duration	$^{\circ}CA$

1 Introduction

Computational engine simulation provides a means of evaluating combustion behaviour while reducing the time and cost associated with experimental development. However, model complexity varies considerably between simulation approaches. One-dimensional (1D) models offer relatively low computational cost and rapid evaluation across broad operating ranges, whereas three-dimensional (3D) CFD can resolve spatial variations in flow, turbulence and combustion at substantially greater computational expense. Determining how these approaches differ in their predictions is therefore important when selecting an appropriate modelling method for engine development.

This project compares Ricardo WAVE, representing the 1D modelling approach, with ANSYS Forte, representing 3D CFD, using a Suzuki GSX1100F spark-ignition engine as a common basis. Four fuels—octane, methanol, ethanol and hydrogen—are investigated from 1000 to 10,000 rpm under equivalent nominal operating conditions. The fuels provide contrasting combustion and thermophysical characteristics, allowing the comparison to examine whether disagreement between the models changes with both fuel and engine speed.

The aim is to evaluate how the two modelling approaches predict combustion, cylinder pressure, temperature and engine-performance behaviour, while considering their respective assumptions, sensitivity and computational requirements. Rather than treating agreement between the models as experimental validation, the study assesses their relative responses under consistently imposed conditions and considers the implications for their practical use in engine simulation.

1.1 Project Planning and Time-Scale Control

The timing plan for the project including the durations of the research, simulation and analysis activities, is presented in Figure

Masters Thesis Timeline

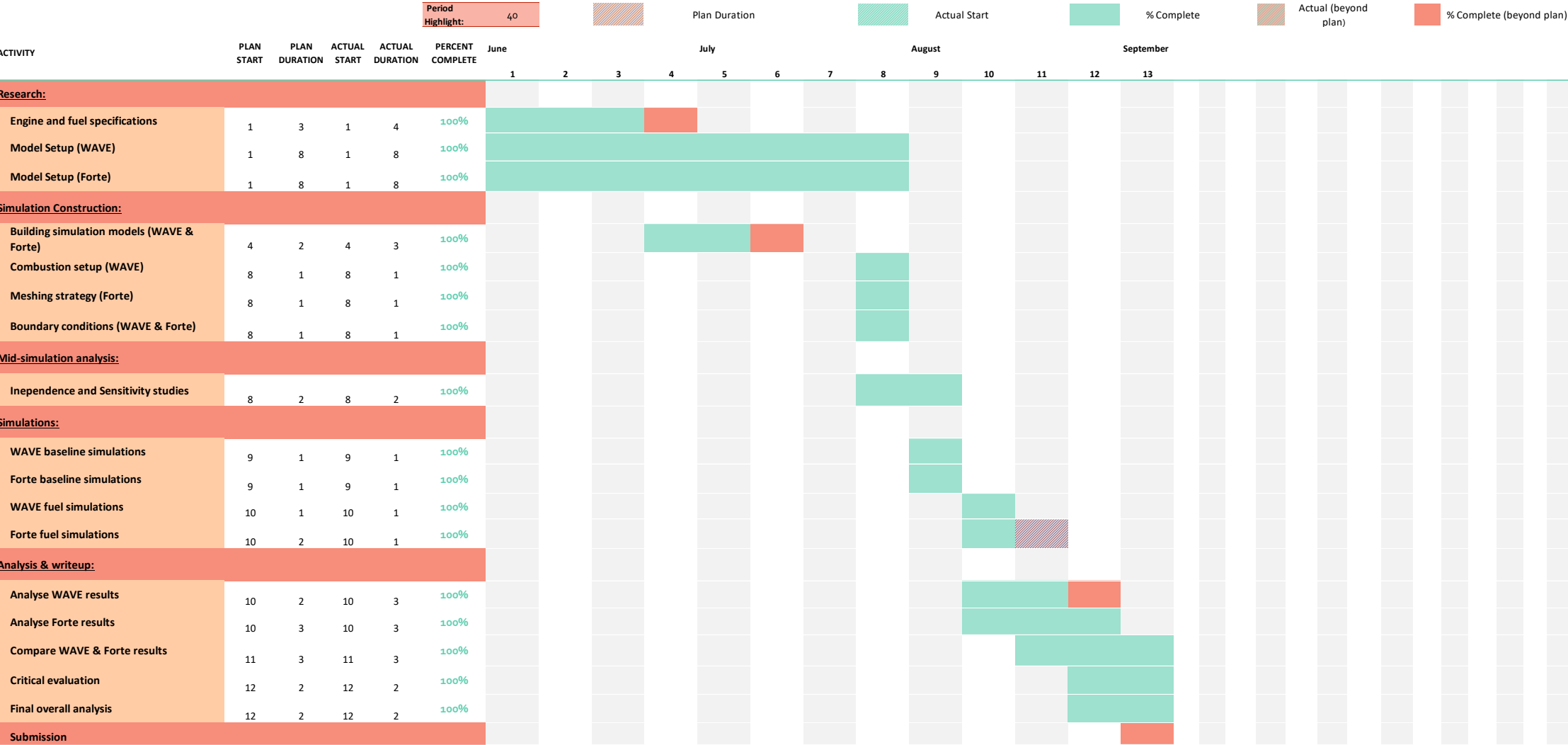


Figure 1: Gnatt chart - project timeline

2 Engine and Fuel Specification

The same physical engine specification was used as the basis for the Ricardo WAVE and ANSYS Forte models to provide a consistent platform for comparison. The engine selected was the Suzuki GSX1100F, a naturally aspirated, four-cylinder spark-ignition engine. Manufacturer data were used where available, supplemented by direct measurements from the engine and clearly defined assumptions where the information required for simulation was unavailable.

Four fuels were considered: octane, methanol, ethanol and hydrogen. Octane was used as the conventional hydrocarbon reference fuel. Where a required property could not be obtained specifically for octane, an iso-octane value was used as a substitute and identified accordingly.

2.1 Geometry

The Suzuki GSX1100F uses a four-cylinder, four-stroke spark-ignition engine with a double overhead camshaft (DOHC) valvetrain and Suzuki's Twin Swirl Combustion Chamber (TSCC). The manufacturer's service manual specifies a bore of 78.0 mm, stroke of 59.0 mm, total displacement of 1127 cm³ and compression ratio of 10.0:1 (Suzuki Motor Corporation, 1990).

These dimensions were used as the principal cylinder geometry in both simulation approaches. The resulting displacement per cylinder is approximately 281.8 cm³. Maintaining the same bore, stroke and compression ratio between the models was necessary because these parameters determine the variation in cylinder volume throughout the engine cycle and therefore directly influence the compression and expansion processes.

Table 1: GSX 1100F geometry data (Suzuki Motor Corporation, 1990)

PARAMETER	VALUE
ENGINE CONFIGURATION	Four-cylinder, four-stroke SI
ASPIRATION	Naturally aspirated
BORE	78.0 mm
STROKE	59.0 mm
TOTAL DISPLACEMENT	1127 cm ³
DISPLACEMENT PER CYLINDER	281.8 cm ³
COMPRESSION RATIO	10.0:1
VALVETRAIN	DOHC
COMBUSTION CHAMBER	TSCC
CONNECTING-ROD LENGTH	117 mm
INTAKE VALVE DIAMETER	27.5 mm
EXHAUST VALVE DIAMETER	25 mm

The service manual did not provide all of the geometric information required by the simulation models. Valve lift was therefore measured directly from the available engine components, as described in Section 2.2. Any remaining parameters that could not be obtained from the manufacturer were introduced as modelling assumptions and are identified accordingly. The geometry data used in this study can be seen in table 1.

2.2 Valve Lift Profiles

Valve lift was measured directly from the engine to provide engine-specific lift data for the WAVE model. The measured maximum intake and exhaust valve lifts were 8 mm and 7.51 mm, respectively. The resulting profiles used within the simulations are shown in Figure 2.

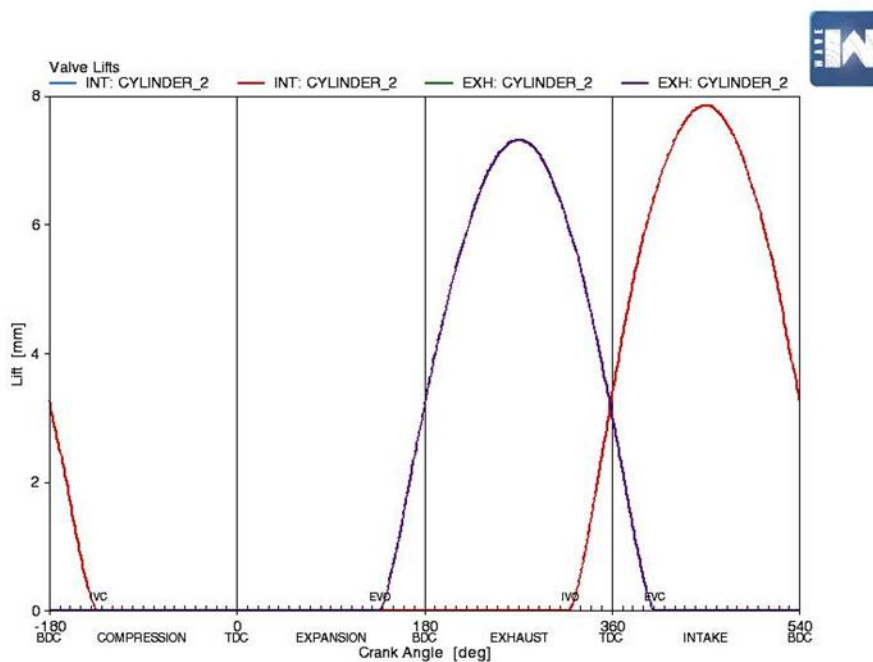


Figure 2: Intake and exhaust valve lift profiles derived from measured valve lift, with assumed valve phasing

Although valve lift could be measured, the original cam timing relative to crank angle was not available from the measurements. It was therefore not possible to position the profiles using measured valve opening and closing crank angles. To provide a consistent valve motion for the numerical models, each measured lift profile was centred on the corresponding gas-exchange stroke. Maximum exhaust-valve lift was positioned at the midpoint of the exhaust stroke (270° CA in the crank-angle convention shown in Figure 2), while maximum intake-valve lift was positioned at the midpoint of the intake stroke (450° CA).

The opening and closing locations shown in figure 2 are therefore consequences of this assumed phasing and the measured lift-profile duration. They should not be interpreted as measured GSX1100F cam timing. The same assumed profiles were used when defining the

valve motion in both simulation approaches to prevent differences in assumed cam phasing from influencing the comparison.

This assumption is a limitation of the engine representation because valve timing affects cylinder filling, residual-gas retention and gas exchange. Its influence is considered further when assessing the limitations of the models.

2.3 Operating Range

The models were evaluated between 1000 and 10000 rpm, in increments of 1000 rpm. This provides ten operating speeds spanning low-speed through to high-speed engine operation while remaining below the manufacturer's stated maximum permitted engine speed of 11,300 rpm (Suzuki Motor Corporation, 1990).

Using the same speed points for each fuel and both simulation methods provides a controlled basis for determining whether the relationship between the WAVE and Forte predictions changes with engine speed. Increasing speed also changes the physical time available per engine cycle and the interaction between turbulence and combustion. Saliba et al. (2026) identify engine speed as one of the parameters affecting the turbulent and chemical timescales governing in-cylinder flame propagation.

All speed points were simulated at wide-open-throttle conditions, with the same nominal operating conditions applied to both WAVE and Forte. The selected operating range was therefore intended to provide a consistent basis for comparison between the two modelling approaches rather than reproduce a complete experimentally measured GSX1100F performance map.

2.4 Ignition Timing

Ignition timing defines the crank-angle position at which ignition occurs and is therefore an important input for accurately representing the combustion process. The Suzuki service manual specifies an ignition timing of 13° BTDC below 1500 rpm, advancing to 32° BTDC above 2375 rpm (Suzuki Motor Corporation, 1990). The same spark-timing schedule was applied in both the WAVE and Forte simulations, with 13° BTDC used at 1000 and 2000 rpm and 32° BTDC from 3000 to 10,000 rpm, as the timing at 2000 rpm wasn't specified.

2.5 Fuels and Properties

The four fuels selected for comparison were octane, methanol, ethanol and hydrogen. Octane was used as the conventional hydrocarbon reference, methanol and ethanol represent oxygenated

alcohol fuels, while hydrogen provides a gaseous fuel with substantially different physical and combustion properties. This range of fuels allows the response of the two modelling approaches to be compared across fuels with differing energy content, stoichiometric requirements, vaporisation characteristics and flame behaviour.

The properties most relevant to the present investigation are summarised in table 2. Where possible, values correspond directly to the investigated fuel. Where a suitable octane property was unavailable, an iso-octane value was used.

Table 2: Fuel properties

PROPERTY	OCTANE	METHANOL	ETHANOL	HYDROGEN
LOWER HEATING VALUE (MJ/KG)	44.4 (Glassman and Yetter, 2008)	20.0 (Boretti, 2020)	26.8 (Saharin, 2013)	120 (Madia <i>et al.</i> , 2025)
STOICHIOMETRIC AFR	15.05	6.44	8.95	34.32
OXYGEN CONTENT (MASS %)	0	49.9	34.7	0
LATENT HEAT OF VAPORISATION (KJ/KG)	364 (Glassman and Yetter, 2008)	1177 (Boretti, 2020)	919 (Saharin, 2013)	N/A
LAMINAR FLAME SPEED (M/S)	0.410 (Glassman and Yetter, 2008)	0.480 (Glassman and Yetter, 2008)	0.443 (Gllder, 1984)	2.10 (Glassman and Yetter, 2008)
RON	100 (Madia <i>et al.</i> , 2025)	108.7 (Naegeli <i>et al.</i> , 1989)	108.7 (Foong <i>et al.</i> , 2014)	93.7 (Madia <i>et al.</i> , 2025)
MON	100 (Madia <i>et al.</i> , 2025)	88.6 (Naegeli <i>et al.</i> , 1989)	90.45 (Foong <i>et al.</i> , 2014)	60 (Madia <i>et al.</i> , 2025)
OCTANE SENSITIVITY (RON-MON)	0	20.1	18.25	33.7
DENSITY (KG/M³)	700 (Glassman and Yetter, 2008)	786 (Boretti, 2020)	790 (Saharin, 2013)	0.089 (Madia <i>et al.</i> , 2025)

Boretti (2020) reports a lower heating value of 20 MJ kg⁻¹, stoichiometric AFR of 6.4, RON of 108.7, latent heat of vaporisation of 1.177 MJ kg⁻¹ and density of 786 kg m⁻³ at 298 K and 1 bar for methanol. The same study reports substantially lower latent heat of vaporisation for gasoline, demonstrating the considerably greater vaporisation energy associated with methanol (Boretti, 2020). Glassman and Yetter (2008) similarly give latent heats of vaporisation of 263 cal g⁻¹ for methanol and 87 cal g⁻¹ for octane, supporting the same distinction.

Ethanol was included as a second oxygenated alcohol fuel. Bata *et al.* (2018) identify its high octane number and high latent heat of vaporisation as important characteristics in SI-

engine applications, with the latter contributing to charge cooling during fuel vaporisation. These properties distinguish ethanol from the hydrocarbon reference and affect the conditions under which combustion develops (Baêta et al., 2018).

Hydrogen provides the greatest contrast with the liquid fuels. Wylie and Panesar (2026) describe hydrogen as having high gravimetric but low volumetric energy density, together with a wide flammability range and low minimum ignition energy. Their analysis also found increasing hydrogen concentration to increase thermal diffusivity and accelerate combustion.

A particularly important property for the combustion models is laminar flame speed. The WAVE model uses the Gülder correlation with separate coefficients for each investigated fuel. The reference (W) constants used in the model are 0.4658 m s^{-1} for octane, 0.492 m s^{-1} for methanol, 0.465 m s^{-1} for ethanol and 2.203 m s^{-1} for hydrogen (Gülder, 1984). These values are subsequently modified by the correlation according to equivalence ratio, temperature and pressure. The complete constants and implementation are presented in Section 4.1.4.

The substantially greater reference value for hydrogen means that it provides a useful test of whether the relationship between the 1D and 3D approaches remains consistent when the underlying flame-propagation characteristics differ considerably from those of the liquid fuels. Saliba et al. (2026) also demonstrate that fuel-dependent combustion cannot be characterised by unstretched laminar flame speed alone, as turbulence, flame stretch and preferential diffusion can alter the effective in-cylinder propagation rate.

The lower heating value and stoichiometric AFR are additionally important when interpreting fuel consumption. A lower gravimetric heating value means that a greater mass of fuel is required for the same nominal chemical-energy input. Consequently, direct comparison of fuel mass consumption between octane, methanol, ethanol and hydrogen without considering their respective energy contents would not provide an equivalent comparison.

2.6 Assumptions

A number of assumptions were required to provide equivalent inputs to WAVE and Forte where measured or manufacturer data were unavailable.

The most significant geometric assumption concerns the valve timing. Valve lift was measured, but its crank-angle phasing was not. The measured exhaust and intake lift profiles were therefore centred at the midpoint of their respective exhaust and intake strokes, as previously described. The resulting opening, closing and overlap periods are model inputs arising from this assumption rather than measured cam events.

The investigated mixtures were defined at stoichiometric conditions ($\phi=1$) for the baseline comparisons. This provides a consistent mixture basis while allowing the required fuel-air ratio to change according to the stoichiometric requirement of each fuel. The same nominal mixture definition was imposed in both modelling approaches.

Octane was retained as the reference hydrocarbon fuel wherever suitable properties and model coefficients were available. Where a required value could not be obtained for octane, iso-octane was used as a substitute. Any such substitution should be identified alongside the relevant value because iso-octane is being used as a modelling surrogate rather than being assumed to be physically identical to octane.

The production GSX1100F uses carburettors, but the complete physical carburetion process was not reproduced. The simulations therefore simplify mixture preparation to establish the required fuel-air conditions for combustion. Effects such as detailed droplet atomisation, carburettor transient behaviour and wall wetting are consequently outside the intended scope of the comparison.

Finally, experimental pressure and combustion measurements from the physical engine were not available for validation across the investigated fuels and speeds. The purpose of the models is therefore to compare the predictions and practical characteristics of the 1D and 3D approaches under equivalent imposed conditions. Agreement between WAVE and Forte should not, by itself, be interpreted as validation against the physical GSX1100F. This assumption applies only to the WAVE model, as the Forte simulation begins at IVC and ends at EVO and therefore does not model valve motion.

3 Simulation Setup

3.1 WAVE Setup (1D modelling)

3.1.1 Ignition Radius

An ignition radius of 2 mm was specified to represent the established flame kernel from which subsequent flame propagation is calculated. Bozza et al. (2005b) assume that the computation begins after the flame-kernel initiation process, at which point an established, spherically shaped flame of approximately 2 mm radius is present. This provides a literature basis for adopting an initial flame radius of 2 mm in the present model.

The same initial radius was subsequently employed by Bozza et al. (2007b), who explicitly specified $r_0=2\text{mm}$ as part of the combustion-model constant set. That set was applied at 1750, 4000 and 5500 rpm. The importance of the parameter itself is further supported by Fontana et al. (2010), who identify initial flame-kernel radius as one of the factors controlling combustion variability and specifically state that it affects ignition-delay time. On this basis, 2 mm was adopted as a literature-derived initial flame radius.

3.1.2 Burning Area Decay Exponent

The Burning Area Decay Exponent controls the reduction in flame-front area towards the end of combustion. Peng (2023) demonstrates that increasing this parameter predominantly extends the latter combustion period, with a greater influence on MFB50–90 than MFB10–50. The requirement to represent this final combustion stage is also supported by fractal SI models that introduce a decaying combustion rate following flame interaction with the chamber walls (Bozza et al., 2005a; De Bellis et al., 2014).

For numerical selection, Peng (2023) used a baseline Burning Area Decay Exponent of 0.86, with optimized parameter sets producing values of 0.87 and 0.93. Therefore, 0.87 was adopted as a literature-derived value, as it represents an optimized value while remaining close to the baseline used in the study.

3.1.3 Turbulent Flame

3.1.3.1 Model Selection

The Speed dependent turbulent-flame formulation was selected because engine speed is explicitly linked to the development of flame wrinkling in published fractal SI combustion models. Bozza et al. (2005b) describe the flame-wrinkling rate as increasing with both instantaneous flame radius and turbulence intensity, with turbulence intensity in the formulation related to engine speed. Their

non-dimensional flame-wrinkling expression therefore includes both a reference flame radius and a reference engine speed.

This treatment is retained in the later model of Bozza et al. (2007b), which states that during initial flame development the wrinkling rate increases as a function of flame radius and engine speed. Independent support for the importance of engine-speed dependence is provided by Demesoukas et al. (2013), who found that the coefficient governing their fractal combustion model exhibited an increasing trend with engine speed and concluded that the fractal-model coefficient varies with both speed and load. These findings support use of the Speed dependent formulation rather than treating turbulent flame development as invariant with engine speed

3.1.3.2 Reference Flame Radius

The speed-dependent formulation requires a reference flame radius to scale flame-wrinkling development. Demesoukas et al. (2013) show that flame wrinkling increases with flame radius as more turbulent structures interact with the developing flame front. Bozza et al. (2005b) use a reference flame radius of 1.2 cm (12 mm) within their experimentally assessed gasoline-SI fractal model and retain this value across the investigated operating conditions. Therefore, 12 mm was adopted for the present model.

However, Bozza et al. (2005b) identify reference flame radius as a tuneable parameter and, following tuning against experimental pressure cycles, employ a value of 1.2 cm (12 mm). This value was subsequently kept fixed across all engine operating conditions investigated in their study. As engine-specific experimental calibration was outside the scope of the present study, 12 mm was adopted as a literature-derived reference flame radius for the present model.

3.1.4 Laminar Flame Speed Constants

Table 3: Laminar Flame Speed Constants

LAMINAR FLAME SPEED CONSTANTS	OCTANE	METHANOL	ETHANOL	HYDROGEN
w	0.4658 m/s	0.492 m/s	0.465 m/s	2.203 m/s
EM	-0.326	0.25	0.25	0.3588
ζ	4.48	5.11	6.34	2.703
α	1.56	1.75	1.75	1.232
β	-0.22	-0.20	-0.17	0.1027

The laminar flame speed parameters for ethanol, methanol and iso-octane presented in table 3 were primarily obtained from Gülder (1984). Gülder (1984) developed fuel-specific laminar burning velocity correlations for use in SI-engine cycle simulations, accounting for equivalence ratio, unburned-mixture temperature and pressure. The coefficients W , n and ζ reported by Gülder (1984) were used for the WAVE laminar speed constant W , equivalence-ratio exponent EM , and equivalence-ratio multiplier ζ , respectively. The interpretation of Gülder's coefficients for engine simulation is supported by Kaul (2008), who implemented

Gülde's ethanol correlation in a multi-cycle SI-engine model and explicitly identified the ethanol coefficients as W , the equivalence-ratio exponent and ζ .

The temperature exponent α and pressure exponent β were also obtained from Gülde (1984). As the present simulations were conducted at the stoichiometric equivalence ratio of $\phi=1.0$, the ethanol and methanol pressure exponents were obtained by evaluating Gülde's lean and rich expressions at this condition; both expressions give the same value at $\phi=1.0$. The iso-octane pressure exponent is given directly as a constant by Gülde (1984). No EGR was implemented in the present engine model; therefore, the EGR correction was disabled by setting the WAVE EGR fraction multiplier DK to zero, while the EGR fraction exponent DM was retained at 1.

For hydrogen Verhelst and Sierens (2007) give a mean temperature exponent of $\alpha=1.232$. As the simulations were stoichiometric ($\phi=1$), the pressure exponent was obtained from Equation 1.

Equation 1: Pressure exponent for $\phi \geq 0.6$ (Verhelst and Sierens, 2007, Eq. 15).

$$\beta_p = 0.0246\phi + 0.0781$$

At $\phi=1$, Equation 1 gives $\beta=0.1027$. The remaining WAVE constants W , E and ζ were not provided in the literature. They were therefore determined using calculated hydrogen laminar flame speeds and the WAVE SI Predictive laminar flame-speed equation (equation 2).

Equation 2: WAVE SI Predictive laminar flame-speed equation (Ricardo plc, 2019).

$$S_{lam} = W\phi^{EM} e^{-\zeta(\phi-1.075)^2} \left(\frac{T_u}{298}\right)^\alpha \left(\frac{P}{101325}\right)^\beta (1.0 - DK \times EGR^{DM})$$

For curve fitting, temperature and pressure were fixed to ambient conditions and there is no EGR when calculating the flame-speed dataset, allowing Equation 2 to be reduced.

Equation 3: Reduced WAVE SI Predictive laminar flame-speed equation.

$$S_{lam} = W\phi^{EM} e^{-\zeta(\phi-1.075)^2}$$

Equation 3 allows W , E and ζ to be estimated by curve fitting when plotting flame speed against equivalence ratio. The required hydrogen flame speeds were calculated using the reference stretched laminar burning velocity equation (equation 4).

Equation 4: Reference stretched laminar burning-velocity correlation (Verhelst and Sierens, 2007, Eq. 13).

$$u_{n0} = -4.77\phi^3 + 8.65\phi^2 - 0.394\phi - 0.296$$

At $\phi=1$, Equation 4 gives 3.190 m/s at the reference conditions of 365 K and 5 bar. This was converted to 298.15 K and 1 atm using Equation 5.

Equation 5: Hydrogen burning-velocity correlation (Verhelst and Sierens, 2007, Eq. 12).

$$u_n = u_{n0} \left(\frac{T}{T_0}\right)^{\alpha_T} \left(\frac{p}{p_0}\right)^{\beta_p} (1 - \gamma f)$$

As residual-gas dilution (EGR) was not modelled, $f=0$, reducing $(1-\gamma f)$ to one. Equation 5 gives 2.110 m/s at 298.15 K and 1 atm, compared with 2.10 m/s reported by Law (1993, cited in Glassman and Yetter, 2008), giving a difference of 0.49%. (Glassman and Yetter (2008) identify entries marked (L) in table F1 as data from Law.)

The flame-speed dataset was then calculated at 298.15 K and 1 atm over $\phi=0.6$ –1.6 in increments of 0.1, using the same equation methodology as the preceding verification. The lower limit follows from the $\phi \geq 0.6$ pressure exponent equation of Verhelst and Sierens (2007). As the simulations were conducted at $\phi=1$, the upper limit was set to 1.6 so that the stoichiometric condition was centred within the fitting range.

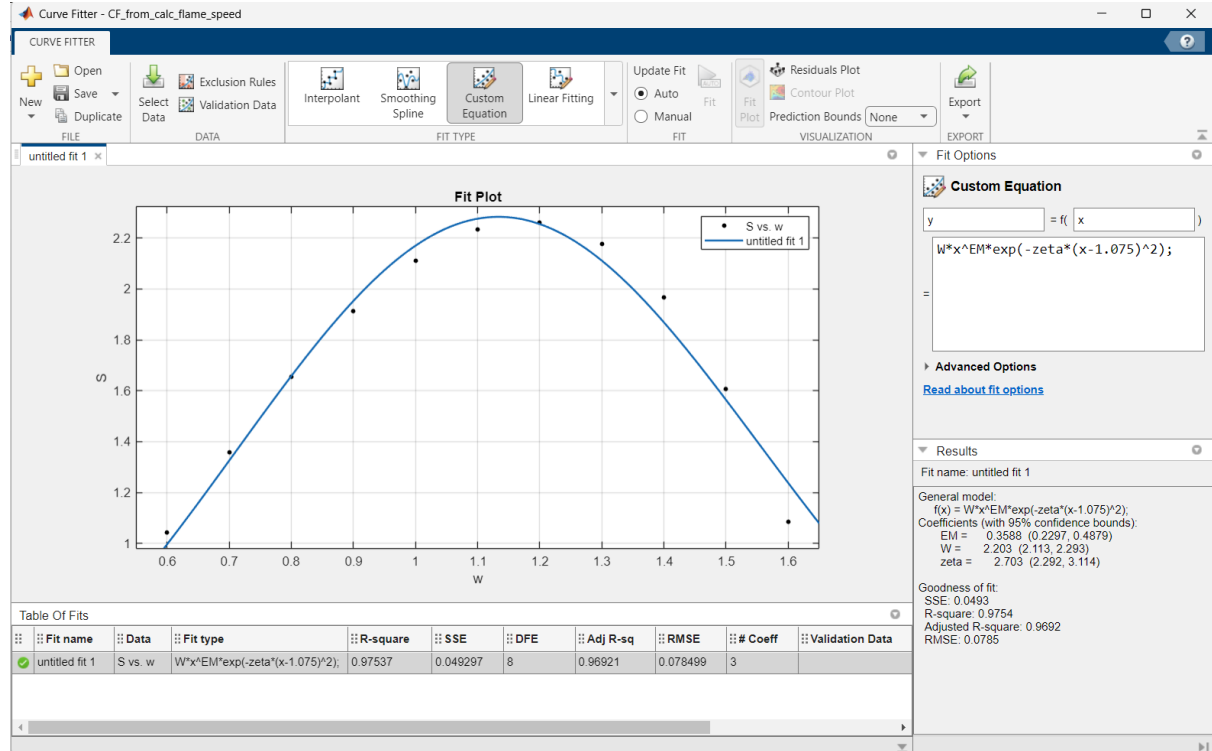


Figure 3: Hydrogen flame speed against equivalence ratio and fitted equation 3.

Fitting Equation 3 to the calculated data, shown in figure 3, gave $W=2.203$ m/s, $EM=0.3588$ and $\zeta=2.703$, with (coefficient of determination) $R^2=0.9754$ and a (root mean square error) RMSE of 0.0785 m/s and a 95% confidence bounds. The R^2 value of 0.9754 indicates that the fitted equation accounts for 97.54% of the variation in the calculated flame-speed data, while the RMSE of 0.0785 m/s represents the typical difference between the calculated data points and the fitted curve. The 95% confidence bounds give the upper and lower limits for each fitted coefficient, with a 5% risk that the method used to calculate these bounds does not capture the true value.

Finally, the fitted coefficients were applied to Equation 2 together with $\alpha=1.232$ and $\beta=0.1027$. At $\phi=1$, 298.15 K and 101325 Pa, the resulting WAVE flame speed was 2.171 m/s, which is 3.39% higher than the 2.10 m/s reported by Law (1993, cited in Glassman and Yetter, 2008).

3.1.5 Turbulence and Flow Model

The Predictive K-k model was selected for the WAVE turbulence and flow calculation. This choice is consistent with the turbulence treatment commonly used alongside quasi-dimensional predictive SI combustion models. Bozza et al. (2007a) describe a fractal SI combustion model in which the in-cylinder turbulence is represented using a K-k formulation, with separate treatment of mean-flow

kinetic energy and turbulent kinetic energy. Lee and Filipi (2011) similarly use a WAVE-coupled predictive SI framework in which turbulence production, dissipation and turbulent kinetic energy are explicitly represented, supporting the use of a predictive turbulence model rather than a prescribed turbulence profile or piston-speed correlation.

Further support is provided by Irimescu et al. (2017), who found that a turbulence-model-based approach gave flame-area predictions closer to optical measurements than a simpler intake-flow-based treatment during flame propagation. Consequently, Predictive K-k was selected as the most appropriate of the available WAVE turbulence-model options for use with the SI Predictive combustion model.

3.2 FORTE Setup (3D modelling)

3.2.1 Mesh sizing

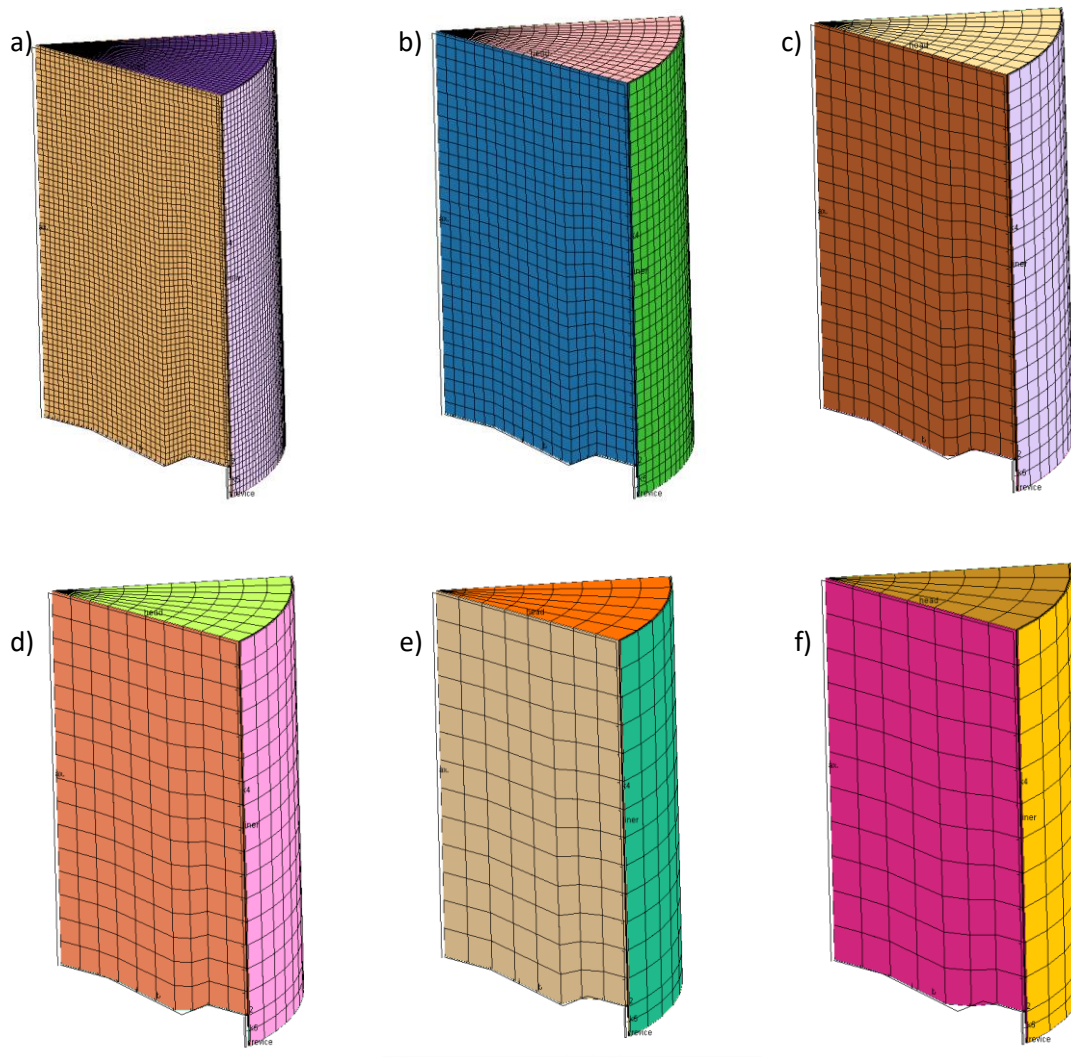


Figure 4: sector mesh representative of global mesh sizes: a) 1 mm, b) 2 mm, c) 3 mm, d) 4 mm, e) 5 mm, f) 6 mm

The mesh sizes selected for the present mesh- sensitivity study were derived from published three-dimensional engine CFD studies using ANSYS Forte and comparable Cartesian-mesh solvers. Hafiz et al. (2023) investigated global mesh sizes of 2 mm and 3 mm in ANSYS Forte, both with and without local refinement. The refined 3 mm case produced a peak-cylinder-pressure difference of approximately 1.2% relative to the refined 2 mm case, while requiring only a third of the computational time. This demonstrates that global resolutions in the 2–3 mm range can provide closely converged combustion predictions when appropriate local refinement is retained.

Saliba et al. (2026) independently employed a 3 mm global mesh in an ANSYS Forte combustion model. Their model also retained local mesh refinement within the combustion domain, showing that a 3 mm global resolution can be combined with finer local regions rather than requiring the entire chamber to be resolved at the smallest cell size. Aktas (2022) provides further Forte-specific support at a slightly finer intermediate resolution, using a 2.25 mm global mesh in a spark-ignition natural-gas engine simulation alongside local refinement at the spark and wall boundaries. Aktas and Karyeyen (2022) similarly adopted a 2.25 mm global mesh with local refinement in their ANSYS Forte simulation of a converted spark-ignition natural-gas engine. These two studies therefore place a commonly used Forte global resolution between the 2 and 3 mm cases considered in the present study.

A broader global-mesh range is supported by studies using CONVERGE. Ferretti (2018) used a 4 mm base mesh for a natural-gas/hydrogen spark-ignition engine while applying fixed embedding and adaptive refinement within the combustion chamber. De Candia (2019) similarly adopted a 4 mm base grid in a spark-ignition natural-gas/hydrogen model, again supplementing the base resolution with local embedding at important combustion-chamber surfaces. These studies demonstrate that a coarser global resolution of approximately 4 mm can be employed successfully when local refinement is maintained in regions requiring higher spatial resolution.

The upper end of the proposed range is supported by Wu et al. (2025), who considered 2, 4 and 6 mm base meshes in a natural-gas engine CFD model. Their comparison showed a 6.1% error between the 6 and 4 mm cases, whereas the error between the 4 and 2 mm cases was 0.8%. This indicates that 6 mm represents a useful coarse-grid condition while the solution begins to approach mesh independence as the global resolution is reduced toward approximately 4 mm and below.

Collectively, the seven studies establish a literature-supported global/base mesh range extending from approximately 2 to 6 mm, with several independent studies concentrated between 2 and 4 mm. Local mesh refinement was retained in these studies, meaning that the quoted global sizes represent the background mesh rather than the finest resolution present within the combustion chamber. On this basis, nominal mesh sizes of 6, 5, 4, 3 and 2 mm were selected for the present mesh- sensitivity study. The 2, 3, 4 and 6 mm values are directly represented within the reviewed literature, while the 5 mm case is included as an intermediate point between the literature-supported 4 and 6 mm cases, providing a uniform 1 mm progression across the complete test range.

Based on the published results, it is hypothesised that the 6 mm mesh will remain too coarse to provide mesh-independent combustion predictions, while progressively smaller changes will occur between the 4, 3 and 2 mm cases. In particular, the small difference reported by Hafiz et al. (2023) between refined 3 and 2 mm meshes, together with the reduced sensitivity reported by Wu et al. (2025) between 4 and 2 mm, suggests that mesh independence is likely to occur within the 3–4 mm global-size region. The purpose of the present study is therefore to evaluate the literature-derived range of nominal mesh sizes to determine the coarsest resolution that achieves reasonable mesh independence, thereby minimising computational time cost without compromising the reliability of the predicted combustion behaviour. A mesh-refinement approach was also investigated; however, it could not be implemented due to limitations in producing the required CAD geometry. The proposed methodology is provided in Appendix A for completeness. Consequently, a 1 mm case was introduced, and the sector mesh generator was used with calculated mesh spacing to produce sector meshes with approximately equivalent nominal resolutions. Meshes are shown in figure 4, as can be seen the coarser meshes deform more around the valve cutout in the piston crown reducing accuracy.

3.2.2 Spark

3.2.2.1 Model Selection

A review of published three-dimensional spark-ignition engine simulations was undertaken to identify an appropriate approach for representing spark initiation and subsequent flame propagation. Patsey and Galyshev (2018) modelled a premixed natural-gas spark-ignition engine using ANSYS Forte, employing the Discrete Particle Ignition Kernel (DPIK) model for spark ignition and the G-equation model for natural-gas combustion. Liu (2018) similarly employed the DPIK and G-equation framework within a three-dimensional CFD investigation of a heavy-duty diesel engine converted to natural-gas spark-ignition operation, providing evidence from an engine configuration particularly comparable to the present application.

Liu, Szybist and Dumitrescu (2018) provide further evidence for this modelling approach using ANSYS Forte 17.2. The authors employed the G-equation combustion model together with the Discrete Particle Ignition Kernel (DPIK) flame model for their spark-ignition engine simulation. The study also identifies the flame development coefficient (C_{m2}) as a tuning parameter governing the transition from laminar to fully developed turbulent flame behaviour, with larger values accelerating this transition. This demonstrates the combined application of DPIK and G-equation modelling while also highlighting the importance of defining the associated flame-development parameters appropriately.

Further application of the G-equation is demonstrated by Aktas (2022), who used ANSYS Forte to model a diesel engine converted to operate entirely as a natural-gas spark-ignition engine. The study employed the G-equation combustion model together with spark-kernel modelling to investigate the resulting combustion behaviour. Kruczek et al. (2019) also applied a G-equation-based combustion framework to the numerical simulation of methane and biogas combustion in a spark-ignition

engine, providing additional evidence for its application to gaseous-fuel combustion. The approach is not limited to natural-gas engines, with Yan et al. (2022) also employing the G-equation and DPIK model in a three-dimensional gasoline direct-injection spark-ignition engine simulation.

The reviewed literature therefore demonstrates repeated application of the DPIK and G-equation modelling framework to three-dimensional spark-ignition engine simulations, including multiple applications in ANSYS Forte and, importantly, natural-gas engines converted from diesel operation. On the basis of this literature evidence, the DPIK model was selected to represent spark-kernel development, coupled with the G-equation model for subsequent flame propagation in the present simulation.

3.2.2.2 DPIK Settings

3.2.2.2.1 Spark Duration

Spark duration defines the period over which energy is supplied to the developing ignition kernel. The reviewed DPIK studies specify spark duration in either time or crank-angle degrees (CAD). To provide a consistent basis for comparison, values reported in CAD were converted to seconds using Equation 6.

Equation 6: spark duration conversion

$$t = \frac{\theta}{6 \text{ RPM}}$$

Liu (2018) specifies a spark duration of 1 ms using ANSYS Forte 17.2. Liu and Dumitrescu (2019) also use 1 ms; however, both publications originate from the same heavy-duty natural-gas SI engine research programme and are therefore treated as corroborating evidence rather than two independent selections.

Kruczek et al. (2019) use a spark duration of 15 CAD at 1500 rpm in ANSYS Forte. Using Equation 2, this corresponds to 1.67 ms. Mauro et al. (2018) use 7 CAD at 600 rpm in ANSYS Forte, corresponding to 1.94 ms. Liang and Reitz (2006) directly specify a time-based spark duration of 2 ms within their DPIK implementation. The independent literature therefore supports spark durations of 1.00, 1.67, 1.94 and 2.00 ms.

Mauro et al. (2018) provides the most relevant basis for the base model because the study uses a spark-ignition engine fuelled with iso-octane, matching the baseline fuel used in this study. Its equivalent spark duration of 1.94 ms is closely supported by the 2 ms value independently specified by Liang and Reitz (2006). Kruczek et al. (2019) provides additional ANSYS Forte support at 1.67 ms, whereas the lower 1 ms value reported by Liu (2018) is derived from a heavy-duty engine converted to natural-gas spark-ignition operation.

A spark duration of 2 ms was therefore selected for the present model. This value closely represents the 1.94 ms duration used in the most relevant iso-octane study while also being directly supported by an independent DPIK application. The agreement between these studies provides sufficient justification for the selected value without requiring an additional spark-duration sensitivity study. To be used in this study the duration is required in degrees. The 2 ms was converted to crank angle duration for each rpm, shown in table 4.

Table 4: calculated spark duration in degrees for 1,000 - 10,000 RPM

ENGINE SPEED (RPM)	1000	2000	3000	4000	5000	6000	7000	8000	9000	10000
CA DURATION (°)	12	24	36	48	60	72	84	96	108	120

3.2.2.2.2 Energy Release Rate

The DPIK energy-release rate defines the rate at which electrical energy is supplied to the developing ignition kernel and therefore influences its early development. Tan and Reitz (2006) demonstrated that variations in deposited electrical energy produce corresponding changes in ignition-kernel growth, establishing the sensitivity of the early ignition process to the prescribed energy input. Within the DPIK formulation, the kernel develops under the influence of this energy deposition until the conditions required for transition to the G-equation flame-propagation model are satisfied. Consequently, the specified energy-release rate influences both the development of the ignition kernel and the point at which the simulation transitions from the DPIK to the G-equation model (Tan and Reitz, 2006).

A lower energy-release rate of 20 J/s is consistently reported across several DPIK applications. Ratnak et al. (2015) specified 20 J/s within a FORTE DPIK simulation of an SI engine operating with PRF90, providing direct relevance to the iso-octane baseline considered in the present study. Ortiz-Imedio et al. (2022) independently employed 20 J/s within an ANSYS Forte SI model considering hydrogen, methane and coke-oven gas, extending the evidence to fuels relevant to the alternative-fuel investigation. Pérez de Albéniz Azqueta (2020) also specified 20 J/s within an ANSYS Forte natural-gas SI application. The repeated implementation of this value across gasoline-surrogate and gaseous-fuel simulations establishes 20 J/s as a credible lower literature value for the present model.

A second group of studies supports energy-release rates within an upper range of approximately 28–30 J/s. Lee et al. (2024) employed a spark energy-release rate of approximately 30 J/s for a single-spark-plug configuration operating with RON95 gasoline. This provides particularly relevant upper-range evidence because both the fuel and ignition configuration are representative of the baseline application. Kantaroğlu (2025) independently specifies 30 J/s per spark plug within an ANSYS Forte model validated against experimental gasoline-engine data before its application to several alternative fuels. Further support for this range is provided by Fan and Reitz (2000), whose DPIK kernel-validation case employed 2.8 mJ over 100 μ s, corresponding to an average energy-release rate of 28 J/s. Although this case considers a CH₄-air mixture rather than gasoline, it provides independent evidence from the development of the DPIK methodology that an energy-release rate close to 30 J/s is physically applicable within the model. The literature therefore provides three independent examples supporting the upper 28–30 J/s range, with the gasoline applications of Lee et al. (2024) and Kantaroğlu (2025) providing the greater application relevance.

The reviewed literature therefore establishes a literature-supported parameter uncertainty in the DPIK energy-release rate, with credible values spanning approximately 20–30 J/s. Both the 20 J/s and upper 28–30 J/s regions are supported by relevant SI applications, including studies employing gasoline or primary-reference fuels representative of the present baseline.

However, the literature does not provide sufficient evidence to determine which region is more appropriate for the specific ignition system considered in the present study. An average value of 25 J/s was therefore adopted as the nominal input for the simulations, representing the centre of the literature-supported parameter uncertainty. Separate baseline sensitivity cases at 20 and 30 J/s were conducted to quantify the influence of this uncertainty on the predicted combustion behaviour. This approach enables a consistent nominal value to be applied throughout the main simulations while explicitly assessing the potential error associated with uncertainty in the DPIK energy-release-rate specification.

3.2.2.2.3 Initial Kernel Radius

The initial kernel radius defines the starting size from which flame-kernel development is calculated and is therefore an important initial condition within the ignition model. Chang and Han (2014) demonstrated that a minimum flame-kernel size is required for self-sustaining flame propagation, with the required radius dependent on mixture conditions, including air–fuel ratio. Their analysis showed that kernel development depends on the balance between heat generated within the reacting mixture and heat lost from the kernel surface, demonstrating the physical importance of appropriately representing the initial flame-kernel size (Chang and Han, 2014).

A value of 0.5 mm is consistently supported across the reviewed SI ignition-modelling literature. Tan and Reitz (2003) assumed an initial radius of 0.5 mm approximately 1 μ s after spark breakdown, based directly on experimental observations, and retained the same experimentally supported value in the subsequent improved DPIK formulation (Tan and Reitz, 2006). Lee et al. (2008) independently adopted an initial kernel radius of 0.5 mm in an SI-engine combustion model, again referencing experimental evidence for its selection. Ranasinghe (2013) similarly employed a 0.5 mm nascent kernel radius within an improved DPIK model, subsequently applied to gasoline-fuelled SI-engine simulations. More recently, Ravindran et al. (2022) initialized a 0.5 mm spherical kernel in a G-equation-based ignition model for a gasoline DISI engine and explicitly related this radius to experimental measurements of early spark-kernel development. The agreement across these model developments and SI-engine applications, together with the repeated experimental basis for the value, provides a clear justification for adopting an initial kernel radius of 0.5 mm in the present simulations.

3.2.2.2.4 Energy Transfer Efficiency

The DPIK energy-transfer efficiency, η_{eff} , represents the proportion of the specified spark-discharge energy transferred effectively to the developing ignition kernel after losses associated with the spark plug. Within the DPIK formulation, this efficiency scales the electrical-energy contribution to kernel growth, with losses to the spark-plug electrodes reducing the energy available to the surrounding mixture (Tan and Reitz, 2006). The parameter therefore directly influences the contribution of the electrical discharge to early kernel development.

An energy-transfer efficiency of 0.30 (30%) is established within the DPIK literature from consideration of the physical spark-discharge process. Tan and Reitz (2003) distinguish the extremely short, high-efficiency breakdown period from the subsequent discharge phases, stating that most of the ignition process occurs under conditions for which the effective energy-transfer efficiency is

relatively low, at approximately 30%. Ranasinghe (2013) similarly attributes the limited efficiency of energy transfer to the flame kernel to substantial heat losses at the spark-plug electrodes, supporting an efficiency of approximately 30%. The applicability of this value to the present baseline is further supported by its use in gasoline and gasoline-reference-fuel engine simulations. Liang and Reitz (2006) use $\eta_{\text{eff}}=0.3$ within DPIK and apply the model to a gasoline direct-injection engine, while Ratnak et al. (2015) explicitly select 30% in a FORTE/DPIK simulation using PRF90. The latter provides particularly relevant evidence because PRF90 represents gasoline behaviour using an iso-octane/n-heptane reference-fuel mixture.

A higher value of 0.50 is also present in ANSYS Forte applications. Kantaroğlu (2025) specifies an Energy Transfer Efficiency of 0.50 within a Forte model developed from a gasoline engine baseline, demonstrating that this value is applicable within the same CFD software and to a petrol-relevant engine. However, the study does not provide an application-specific derivation or assessment supporting the selection of 0.50 and employs a twin-spark configuration rather than the single-spark arrangement considered here. Pérez de Albéniz Azqueta (2020) identifies 0.50 as the default Energy Transfer Efficiency in Forte, although the worked application uses methane. Consequently, these studies establish 0.50 as a valid Forte setting but provide less application-specific justification for its selection than the physical and gasoline-reference-fuel evidence supporting 0.30.

An Energy Transfer Efficiency of 0.30 was therefore adopted for the present model. This value was selected based on its established physical basis within the DPIK literature and its subsequent application to gasoline-engine and FORTE/PRF90 simulations, providing stronger justification and closer correspondence with the present petrol baseline than the alternative 0.50 setting, which is also identified as the default value within Forte (Pérez de Albéniz Azqueta, 2020).

3.2.2.2.5 Number of Flame Particles

The Number of Flame Particles defines the number of particles used to track the flame-front location of the spark-ignition kernel within DPIK. Particle number affects the numerical representation of the flame surface; Ranasinghe (2013) states that increasing the number of marker particles improves the accuracy of the calculated local flame-surface density but also increases computational cost and therefore selected 4000 particles as a compromise.

Limited evidence was identified from which an application-specific optimum could be established. ANSYS, Inc. (2026) recommends 3000 flame particles per spark plug for tracking the ignition-kernel flame front, while Pérez de Albéniz Azqueta (2020) also reports this recommended value within a Forte simulation guide. Ranasinghe (2013) employed 4000 marker particles, but this was implemented outside Forte and does not demonstrate an accuracy advantage over 3000 for the present application. Consequently, the ANSYS-recommended value of 3000 flame particles was adopted, as insufficient application-specific evidence was identified to justify deviation from the recommended Forte setting.

3.2.2.3 G-Equation Parameters

The Kernel Flame to G-equation Switch Constant (C_{m1}) defines one of the criteria controlling the transfer from the ignition-kernel model to the G-equation. Forte switches when the kernel radius

becomes greater than C_{m1} multiplied by the local turbulence integral length scale, while the minimum-radius criterion must also be satisfied (ANSYS, Inc., 2026). The 2026 Forte User's Guide gives a suggested C_{m1} range of 1.5–2.5, and the 2025 Theory Manual identifies $C_{m1} = 2.0$ as a typical value (ANSYS, Inc., 2026; ANSYS, Inc., 2025). Liu, Szybist and Dumitrescu (2018) likewise use $C_{m1} = 2.0$, describing it as always equal to 2.0 in their Forte 17.2 formulation. Adamczyk et al. (2020) instead calibrated $C_{m1} = 2.0$ for CNG and 1.50 for biogas; these values formed part of separately adjusted parameter sets chosen to minimise disagreement with experimental cylinder-pressure traces rather than being calculated directly from fuel properties. With no equivalent experimental calibration available for the present model, $C_{m1} = 2.0$ was therefore retained for all fuels.

The Minimum Kernel Radius for Kernel-to-G-equation Switch (r_{min}) provides the second kernel-size requirement: the spark-ignition kernel must grow beyond this absolute radius before it is permitted to transfer to the G-equation (ANSYS, Inc., 2026). The Forte User's Guide gives a suggested value of $r_{min} = 0.1$ cm (1.0 mm) (ANSYS, Inc., 2026). Adamczyk et al. (2020) obtained $r_{min} = 0.1$ cm for CNG and 0.20 cm for biogas as part of the same experimentally calibrated parameter sets described above. The literature therefore provides evidence that this value may be altered during engine-specific calibration but does not provide a general fuel-based method for selecting it. In the absence of such calibration data, the Forte suggested value of $r_{min} = 0.1$ cm (1.0 mm) was adopted for all fuels. This is greater than the previously selected initial flame-kernel radius of 0.5 mm, ensuring that the kernel must grow beyond its initial size before it can satisfy the minimum-radius condition for transfer to the G-equation.

The Flame Development Coefficient (C_{m2}) controls the increasing influence of turbulence as the flame develops from the laminar-kernel stage towards fully developed turbulent combustion; increasing C_{m2} accelerates this development (ANSYS, Inc., 2026). ANSYS specifies a suggested range of 0.5–2.5 (ANSYS, Inc., 2026). Liu, Szybist and Dumitrescu (2018) investigated C_{m2} alongside the flame-stretch coefficient C_{ms} , demonstrating that multiple combinations could reproduce experimental combustion behaviour and therefore that C_{m2} cannot be considered an independently transferable calibration value. Adamczyk et al. (2020) further demonstrate this calibration dependence, obtaining $C_{m2} = 1.1$ for CNG and $C_{m2} = 9.0$ for biogas at $\lambda = 1.0$. Under leaner biogas operation, the original calibration produced pressure instabilities and excessively rapid combustion, requiring further adjustment of C_{m2} and the associated G-equation calibration (Adamczyk et al., 2020). Consequently, the available literature does not establish a uniquely transferable C_{m2} for the present engine. A central value of $C_{m2} = 1.5$ within the Forte-suggested range was therefore selected and retained between fuels. To assess the sensitivity of the model to this uncertain coefficient, $C_{m2} = 0.5$ and 2.5 will additionally be evaluated as lower- and upper-bound cases while the remaining G-equation parameters are held constant.

3.2.2.4 Time Step Parameters

3.2.2.4.1 Initial Simulation Time Step

The Initial Simulation Time Step defines the starting temporal resolution from which Forte's adaptive time-step algorithm begins. The actual time step is subsequently increased or reduced according to the development of solution gradients, allowing the solver to balance computational efficiency with the need to resolve rapidly changing flow and combustion behaviour (ANSYS, Inc., 2026). Bahar

(2023) used an initial time step of 5×10^{-7} s in ANSYS Forte SI-engine simulations and states that the selected range was informed by validated SI-engine simulations and preliminary simulation results.

The same initial value is reported independently across substantially different engine applications. Pérez de Albéniz Azqueta (2020) specifies 5×10^{-7} s in an ANSYS Forte SI-engine setup. Hamdi et al. (2025) also use 5×10^{-7} s in ANSYS Forte 2023 R1 for a validated natural-gas/n-heptane dual-fuel engine model, while Li et al. (2025) independently specify 5×10^{-7} s as the initial time step in a CONVERGE ammonia/diesel engine simulation, demonstrating that the value is not restricted to Forte. Aktas (2022) additionally reports the same initial setting in a natural-gas SI Forte study.

Given the agreement between SI Forte studies, independent Forte engine work and a separate CFD solver, there is no evidence in the reviewed sources that a different initial value would be more appropriate for the present study. An Initial Simulation Time Step of 5×10^{-7} s was therefore selected and will be retained between fuels. The subsequent adaptive time-step control remains responsible for adjusting the actual temporal resolution as the combustion solution develops.

3.2.2.4.2 Maximum Simulation Time Step

The Maximum Simulation Time Step defines the upper limit permitted for Forte's adaptively calculated transport time step. The actual time step may remain below this value as required by changes in the solution, meaning the parameter acts as an upper temporal limit rather than a fixed time step throughout combustion (ANSYS, Inc., 2026). This provides a balance between maintaining sufficient temporal resolution and avoiding unnecessary computational cost.

The reviewed literature identifies 5×10^{-6} s and 1×10^{-5} s as the two principal candidate values. Aktas (2022) used 5×10^{-6} s for natural-gas SI combustion, with related Aktas studies retaining the same setting; these were therefore considered collectively rather than as independent evidence. Independent support for 5×10^{-6} s is provided by Hamdi et al. (2025) for natural-gas/n-heptane RCCI combustion and Gonca (2026) for diesel, ammonia and hydrogen engine simulations. Conversely, Bahar (2023) used a maximum of 1×10^{-5} s for SI-engine simulations, Pérez de Albéniz Azqueta (2020) specified the same maximum in a Forte SI setup, and Kantaroğlu (2025) selected 1×10^{-5} s following a dedicated SI-engine time-step sensitivity assessment. The available evidence therefore supports both values, with the 1×10^{-5} s evidence being particularly relevant due to its greater representation in SI-engine applications.

As neither value is sufficiently dominant to justify selection from literature alone, a maximum-time-step sensitivity study will be conducted using 5×10^{-6} and 1×10^{-5} s. Kantaroğlu (2025) provides a useful indication of the expected behaviour, having assessed 2.5×10^{-6} , 5×10^{-6} , 1×10^{-5} , and 2×10^{-5} s. The results remained closely consistent across the tested range from 2.5×10^{-6} to 1×10^{-5} s, whereas the 2×10^{-5} s case produced poorer flame resolution and changes in heat-release behaviour. Therefore, the two proposed values are expected to produce closely comparable cylinder-pressure and heat-release behaviour if adequate temporal resolution has been achieved. If the sensitivity study confirms negligible differences, 1×10^{-5} s will be selected to reduce computational cost; if meaningful differences are observed, 5×10^{-6} s will be retained to provide greater temporal resolution.

3.2.2.4.3 Maximum Change in Crank Angle per Time Step

The Crank Angle Based Time-Step Restriction limits the maximum crankshaft rotation permitted during a simulation time step. ANSYS, Inc. (2026) specifies 1.1°CA as the Forte default, while Pérez de Albéniz Azqueta (2020) also explicitly applies a 1.1°CA restriction in an SI Forte setup.

Further support is provided by Kantaroğlu (2025), who retained 1.1°CA in a validated SI/Forte model across gasoline, e-hydrogen, e-methanol, e-kerosene, e-ammonia and three syngas compositions. Although different crank-angle restrictions were not sensitivity tested, its successful use across these fuels supports retaining a common value. Gonca (2026) additionally reports the closely comparable value of 1.0°CA.

Therefore, 1.1°CA was selected and will be retained between fuels, as it corresponds to the Forte default and is supported by directly applicable SI/Forte literature.

3.2.2.5 Turbulence Model

ANSYS, Inc. (2026) recommends RANS RNG k-epsilon for engine applications and indicates improved suitability over the standard k-epsilon model for engine combustion. If an LES approach is adopted, ANSYS recommends the Dynamic Structure model; however, LES is not required for engine simulations.

Independent engine-CFD evidence supports the RNG k-epsilon selection. Zhang et al. (2022) selected RNG k-epsilon over standard k-epsilon for an SI-engine model, while Dal Forno Chuahy et al. (2021) considered the model adequate for representing average-cycle SI combustion behaviour. Ayyanar et al. (2025) additionally applied RNG k-epsilon to in-cylinder SI flow with experimental pressure and PIV validation. Chiriches (2019) showed that LES provides greater resolution of instantaneous turbulent structures but requires finer spatial resolution and greater computational resources.

As the present study requires repeated simulations across multiple fuels and engine speeds rather than detailed resolution of cycle-to-cycle turbulence, RANS RNG k-epsilon was selected as an appropriate balance between engine applicability and computational cost. The associated turbulence-model coefficients were retained at their Forte default values, consistent with ANSYS, Inc. (2026), which recommends that these parameters generally remain unchanged.

4 Numerical Verification and Test Method

4.1 Numerical Verification and Sensitivity Studies

Before conducting the main fuel simulations, a series of numerical verification and sensitivity studies were carried out to establish suitable setup parameters for the simulation models. These studies were used to assess the influence of key numerical and modelling parameters on the predicted engine behaviour, allowing appropriate values to be selected for the final simulation setup. This ensured that the main simulations were conducted using a consistent numerical approach while maintaining an appropriate balance between solution accuracy, numerical stability, and computational cost. Undetermined values were set to the forte baseline until the sensitivity and verification simulations were carried out to maintain comparability and show development.

4.1.1 Mesh Sensitivity Study

A mesh sensitivity study was conducted using the iso-octane case at 10,000 rpm, with nominal mesh sizes of 1, 2, 3, 4, 5 and 6 mm used to generate the sector meshes. With common geometry, initial conditions and piston motion, the fitted compression polytropic index remained between 1.3425 and 1.3443 across all meshes, a variation of only 0.13%, consistent with the overlapping pre-combustion paths in Figure 5.1. The strong mesh sensitivity therefore develops primarily once the spatially resolved combustion process begins.

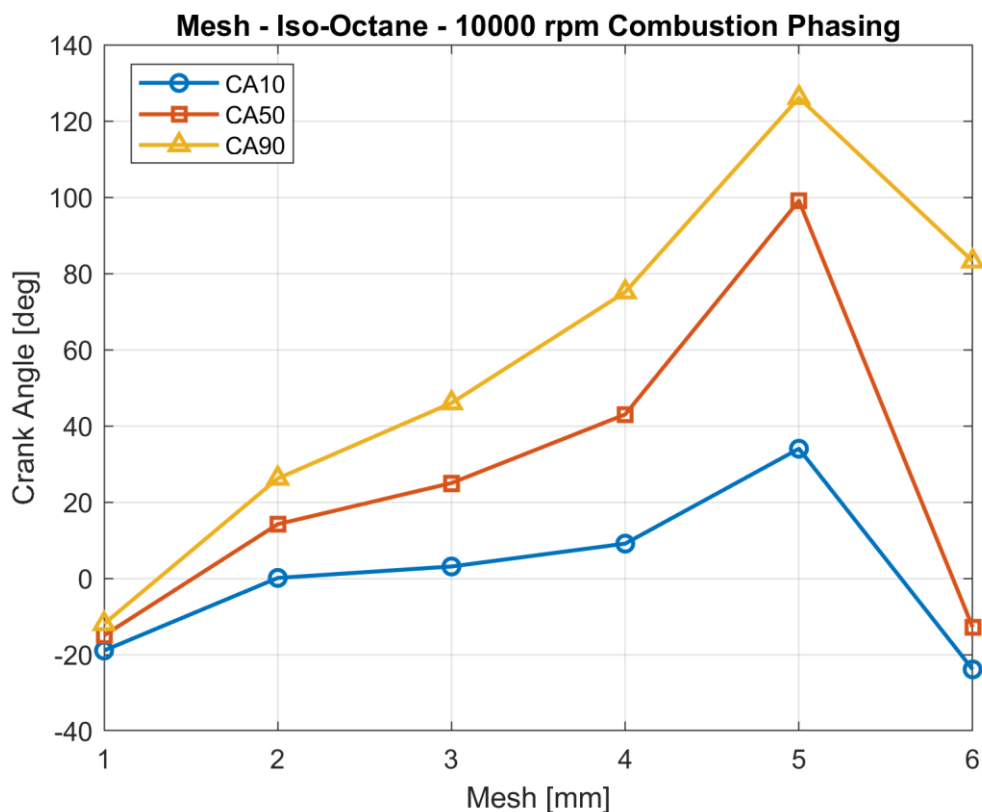


Figure 5: Effect of nominal mesh size on the combustion phasing for the iso-octane case at 10,000 rpm.

The effect of mesh size is particularly clear from the combustion phasing and burn duration as can be seen in figure 5. For the 2 mm mesh, CA50 occurred at 14.2° ATDC with a CA10-90 duration of 26.1°CA. Reduced spatial resolution with progressive mesh coarsening delayed and prolonged the predicted combustion, with CA50 moving to 25.0°, 43.0° and 99.1° ATDC for the 3, 4 and 5 mm meshes respectively, while CA10-90 increased to 42.9°, 66.0° and 92.1°CA. As coarser meshes produced slower, more prolonged combustion, peak heat release weakened and shifted progressively later, shifting from 16.0° ATDC at 2 mm to 34.0°, 49.0° and 115.3° ATDC for the 3, 4 and 5 mm meshes, respectively. At 6 mm, the combustion response broke down into an unusual and extremely weak event, with a combustion efficiency of only 0.91%. Although the normalised mass-fraction-burned curve still produces measurable combustion-phasing values, these should be interpreted cautiously because they correspond to a very small overall amount of combustion. The effect of mesh size on mass fraction burned and combustion phasing is shown in Figure 6.

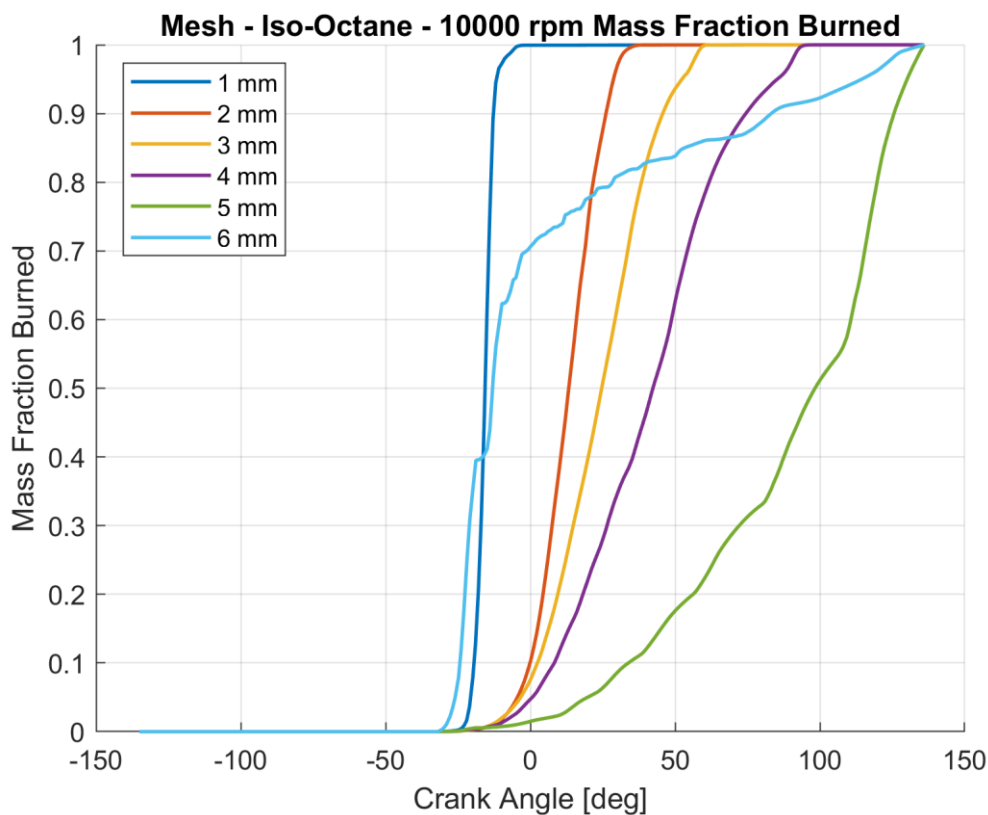


Figure 6: Effect of nominal mesh size on mass fraction burned and combustion phasing for the iso-octane case at 10,000 rpm.

This progressive change in combustion behaviour is also reflected in the pressure-volume response shown previously in Figure 5.1. The 1 to 3 mm meshes exhibit clearly developed combustion and expansion paths, while the 4 and 5 mm meshes show increasingly delayed pressure development. At 6 mm, the trace approaches a motored-like compression and expansion response, consistent with the near absence of combustion. The 1 mm result is also distinctly different from the 2 mm solution, but in the opposite direction. It produces substantially earlier and more rapid combustion, with CA50 occurring at approximately 15.0° BTDC and CA10-90 lasting only 7.0°CA. Its peak apparent heat-release rate also increases to 165.7 J/°CA, compared with 41.4 J/°CA for the 2 mm case. The 1 mm mesh therefore

represents a substantially different combustion regime rather than a small refinement of the 2 mm result.

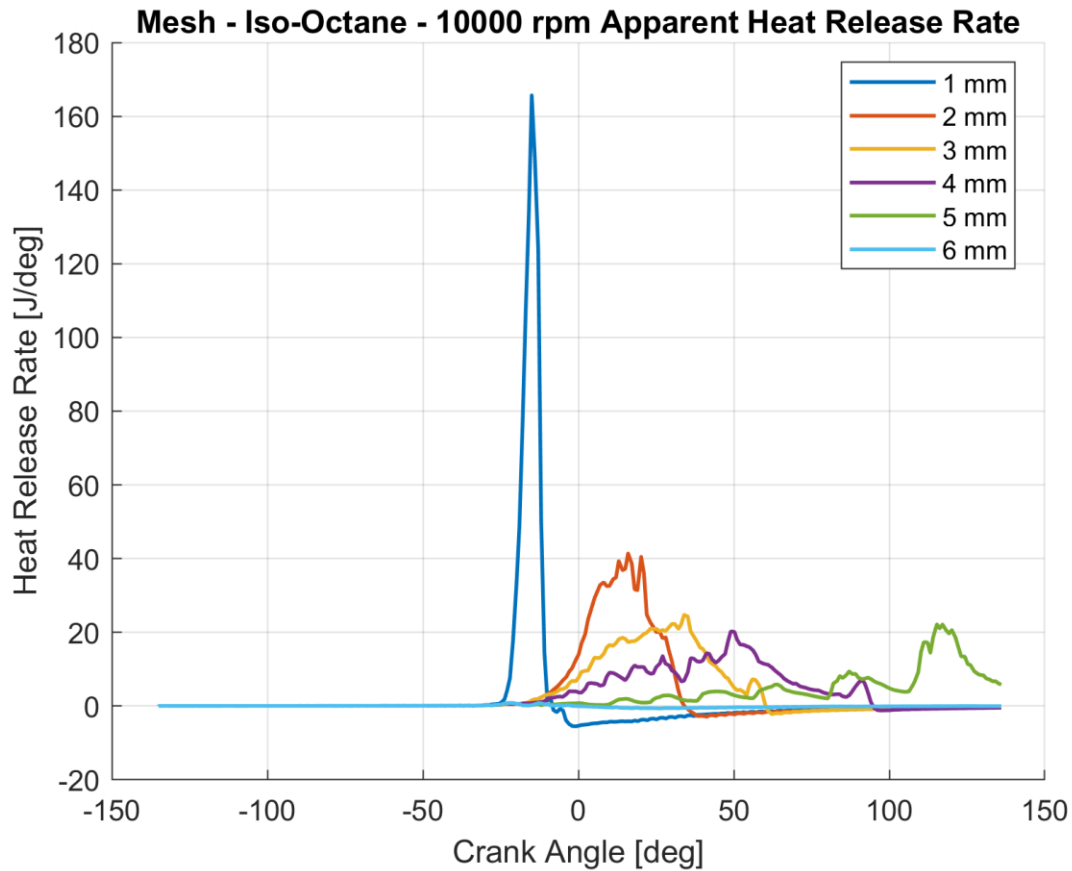


Figure 7: Effect of nominal mesh size on apparent heat-release rate for the iso-octane case at 10,000 rpm.

Figure 7 shows that the mesh sensitivity in combustion phasing is accompanied by substantial changes in both the magnitude and timing of apparent heat release. The 1 mm mesh produces the earliest and most intense heat-release event, reaching a peak apparent heat-release rate of 165.7 J/°CA at approximately 15.0° BTDC. In comparison, the 2 mm mesh reaches 41.4 J/°CA at 16.0° ATDC. Progressive coarsening from 2 to 5 mm shifts the peak progressively later, occurring at 34.0°, 49.0° and 115.3° ATDC for the 3, 4 and 5 mm meshes, respectively. The 6 mm case produces only a very weak apparent heat-release response, consistent with the unusual combustion behaviour observed in the mass-fraction-burned results.

The 2 mm solution was also considered more representative of the combustion behaviour expected for this high-speed operating condition. In contrast, the 1 mm solution predicts an unusually rapid and tightly phased combustion event, with CA10–90 completed within only 7.0°CA and CA50 occurring approximately 15° BTDC. At 10,000 rpm, this burn duration corresponds to approximately 0.12 ms. Together with the substantially higher apparent heat-release and pressure-rise rates, this indicates that the 1 mm solution is producing an unusually rapid and advanced combustion response. The 2 mm solution, with a longer burn duration and later combustion phasing, was therefore considered more physically plausible for the subsequent simulations. However, in the absence of experimental cylinder-pressure or combustion-phasing data, this interpretation represents a physical plausibility assessment rather than validation of the 2 mm mesh.

In addition to these differences in predicted combustion behaviour, the nominal mesh settings produced substantial differences in the resulting fluid-cell population. At approximately -30° ATDC, the fluid-cell population increased from 250 cells for the 6 mm mesh to 4,770 cells for the selected 2 mm mesh and 33,356 cells for the 1 mm mesh. These values are representative of the mesh population at this crank-angle position rather than fixed cell counts throughout the simulation, as the moving engine mesh is dynamically rezoned with piston motion. The cell counts therefore provide additional quantitative context for the substantial change in computational resolution between the investigated nominal mesh settings. The principal mesh-resolution and combustion characteristics for all six cases are summarised in Table 5.

Table 5: Summary of mesh resolution and combustion characteristics for the iso-octane mesh sensitivity study at 10,000 rpm.

Nominal mesh size	Fluid cells near -30° ATDC	CA50 ($^\circ$ ATDC)	CA10-90 ($^\circ$ CA)	Peak apparent HRR (J/ $^\circ$ CA)	Peak HRR location ($^\circ$ ATDC)	Compression polytropic index
1 mm	33,356	-14.98	7.01	165.68	-14.98	1.3443
2 mm	4,770	14.20	26.10	41.41	16.00	1.3425
3 mm	1,390	25.00	42.90	24.75	34.00	1.3425
4 mm	704	43.00	66.00	20.28	49.00	1.3430
5 mm	390	99.10	92.10	22.19	115.30	1.3436
6 mm	250	-12.80	107.1	0.88	-21.80	1.3431

Based on these results, the 2 mm mesh was selected for the subsequent Forte simulations. The selection was primarily based on combustion phasing, burn duration, apparent heat-release behaviour and the $\log(P)$ - $\log(V)$ response. Meshes progressively coarser than 2 mm produced increasingly delayed and prolonged combustion, with the combustion response becoming extremely weak and atypical at 6 mm. Conversely, the 1 mm mesh produced a substantially different early and rapid combustion regime. The substantial difference between the 1 and 2 mm solutions indicates that full mesh independence was not achieved. However, the very short burn duration, advanced combustion phasing and substantially higher heat-release and pressure-rise rates predicted by the 1 mm mesh were considered less representative of the expected high-speed combustion behaviour than the 2 mm solution. In the absence of experimental cylinder-pressure or combustion-phasing data, this should be regarded as a physical plausibility assessment rather than confirmation that the 2 mm solution is quantitatively more accurate. The 2 mm mesh was therefore considered the most defensible resolution for the subsequent simulations among the meshes investigated, while recognising the remaining numerical uncertainty between the 1 and 2 mm solutions.

4.1.2 Energy Release Rate

As discussed during development of the Forte combustion model, an energy release rate of 25 J/s was selected for the main simulations as the central value within the investigated 20–30 J/s range. A sensitivity study was therefore conducted using energy release rates of 20 and 30 J/s for the iso-octane case at 10,000 rpm. The purpose of this comparison was not to identify an optimum value,

but to quantify the sensitivity of the predicted combustion behaviour to variation in this model parameter.

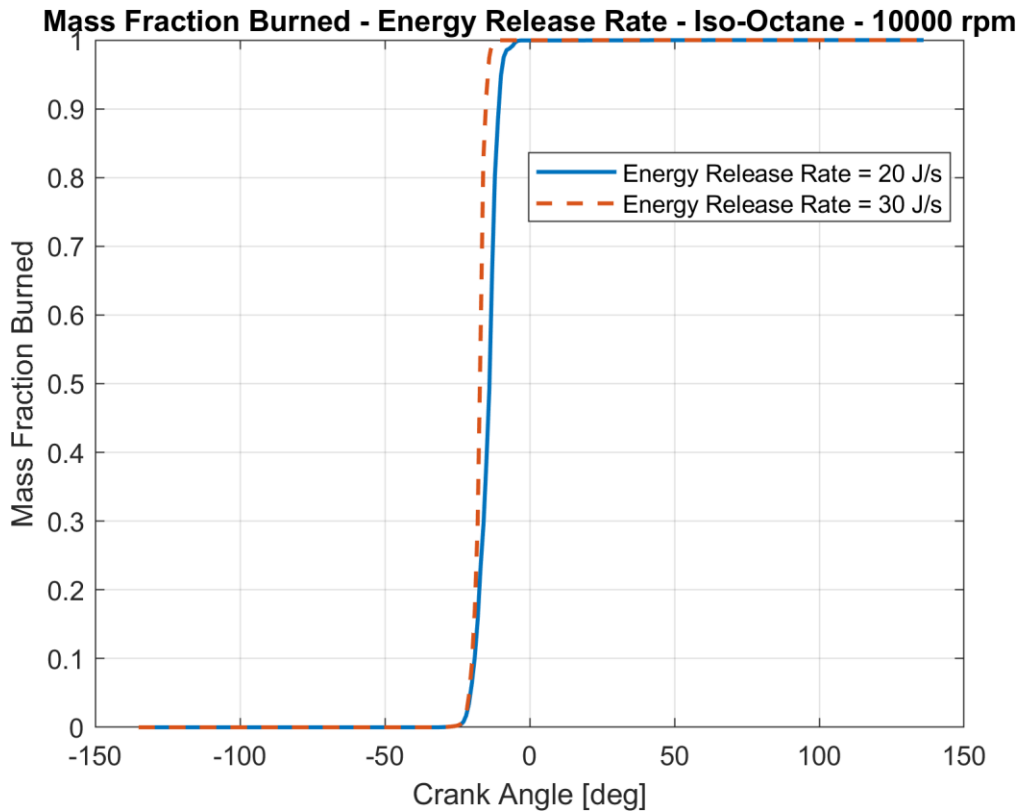


Figure 8: Effect of spark-model energy release rate on mass fraction burned for the iso-octane case at 10,000 rpm.

As shown in Figure 8, the mass-fraction-burned curves initially develop at almost the same crank angle but subsequently diverge, with the 30 J/s case exhibiting a substantially steeper and earlier combustion progression. CA10 remained effectively unchanged at approximately 18.94° BTDC for both cases. However, CA50 advanced from 12.98° BTDC at 20 J/s to 16.96° BTDC at 30 J/s, while CA90 advanced from 9.92° to 14.98° BTDC. The corresponding CA10–90 burn duration decreased from 9.02 to 3.97°CA, a reduction of approximately 56%. The similar initial development followed by divergence is consistent with the energy-release-rate setting having little influence on combustion onset, while the greater energy input at 30 J/s accelerates subsequent kernel and flame development.

Apparent Heat Release Rate - Energy Release Rate - Iso-Octane - 10000 rpm

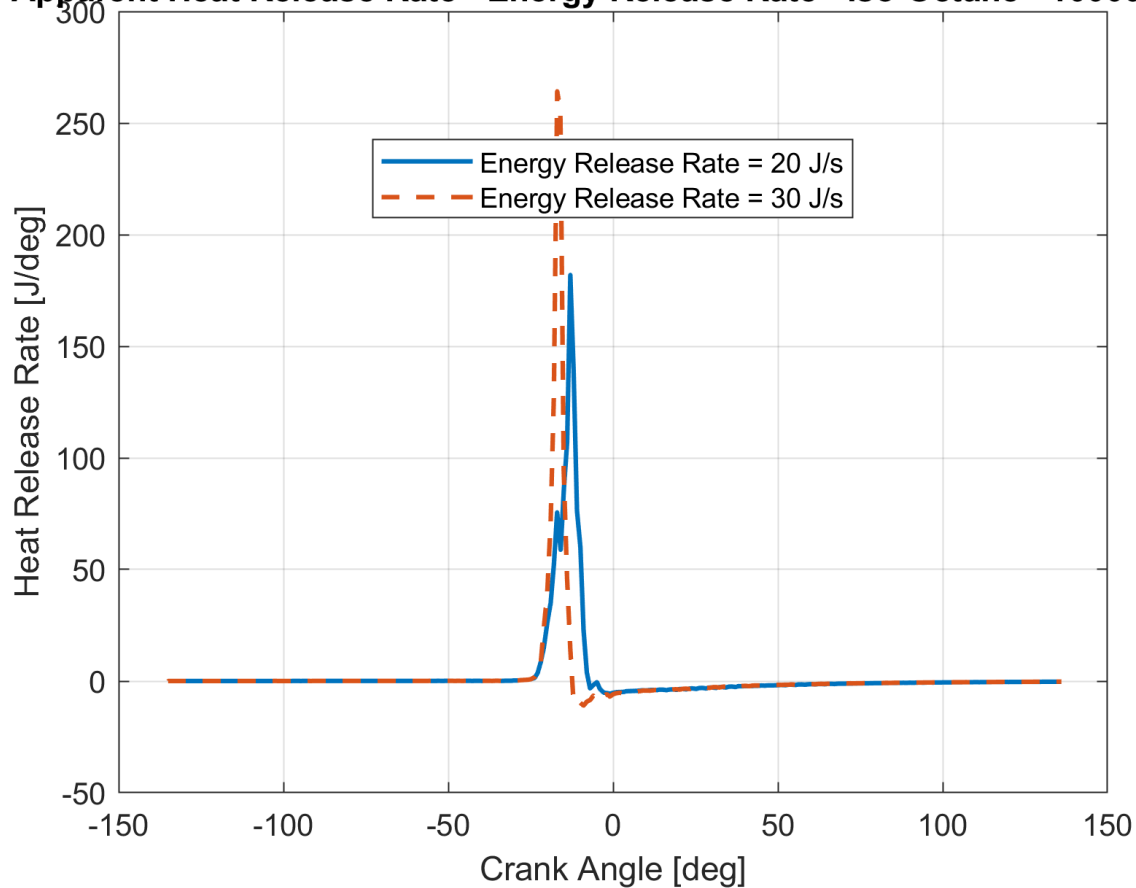


Figure 9: Effect of spark-model energy release rate on apparent heat-release rate for the iso-octane case at 10,000 rpm.

The accelerated combustion development observed in Figure 8 is reflected directly in the apparent heat-release histories shown in Figure 9. Increasing the energy release rate from 20 to 30 J/s increased the peak apparent heat-release rate from 181.97 to 264.22 J/°CA, corresponding to an increase of 45.2%. The peak also occurred approximately 3.99°CA earlier. The higher energy release rate therefore produced a substantially sharper and earlier heat-release event rather than simply altering the initial onset of combustion.

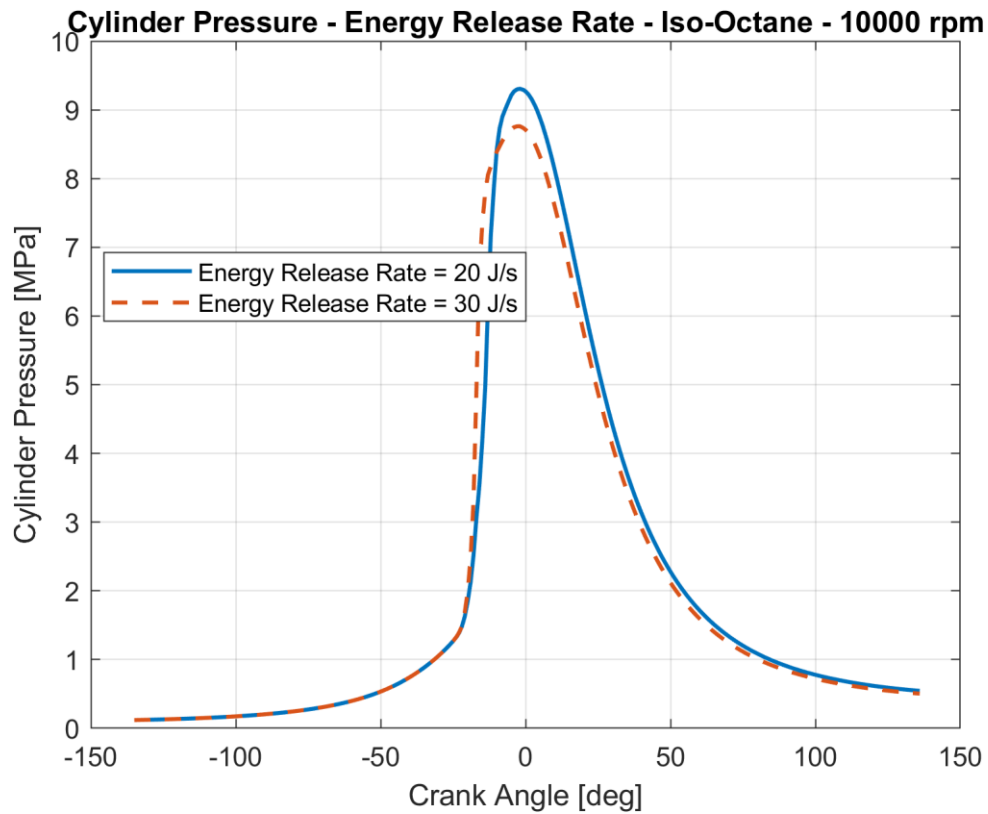


Figure 10: Effect of spark-model energy release rate on cylinder pressure for the iso-octane case at 10,000 rpm.

Figure 10 shows that the substantial change in combustion development produced a comparatively smaller change in peak cylinder pressure. Increasing the energy release rate from 20 to 30 J/s reduced peak cylinder pressure from 9.306 to 8.761 MPa, corresponding to a decrease of 5.86%. The crank angle of peak pressure remained almost unchanged at approximately 1.92° BTDC for both cases. Although combustion develops substantially faster, peak pressure remains near TDC where minimum cylinder volume strongly influences the pressure maximum; Pmax therefore changes much less than the combustion-rate quantities.

Rate of Change of Pressure - Energy Release Rate - Iso-Octane - 10000 rpm

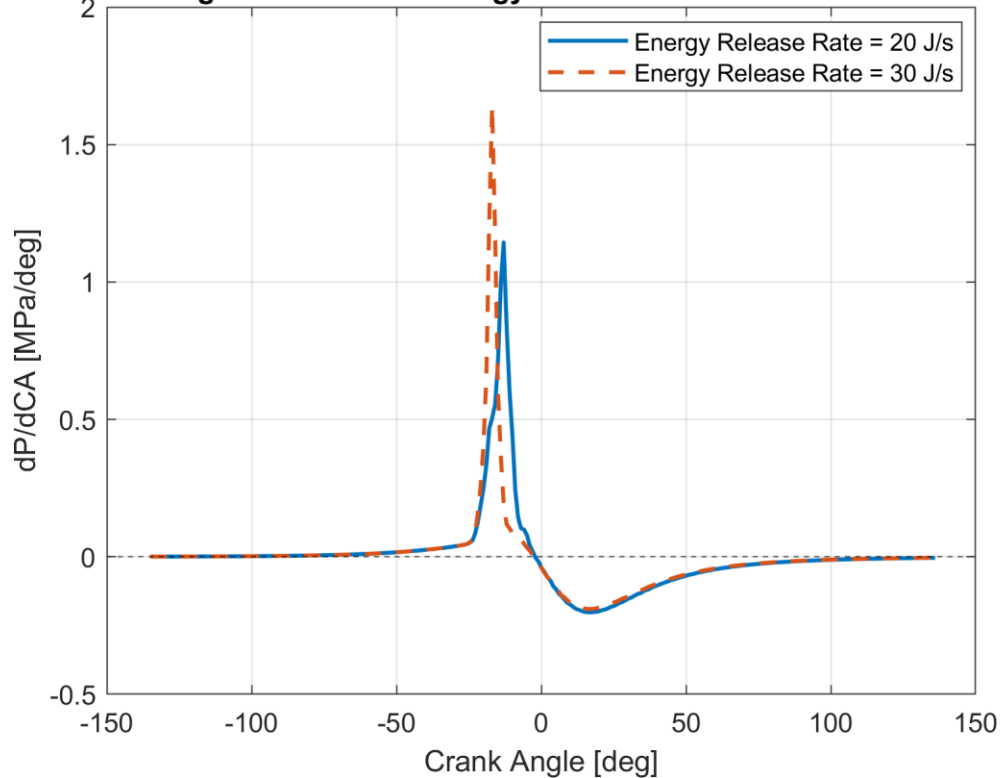


Figure 11: Effect of spark-model energy release rate on the rate of change of cylinder pressure for the iso-octane case at 10,000 rpm.

The greater sensitivity during the rapid combustion period is more clearly demonstrated by the rate of change of pressure in Figure 11. The maximum rate of change of pressure increased from 1.144 to 1.638 MPa/°CA (+43.1%) as the energy release rate increased from 20 to 30 J/s, with the maximum occurring approximately 3.99°CA earlier. The substantially greater change in the rate of change of pressure compared with peak cylinder pressure demonstrates that peak pressure alone does not fully represent the influence of the energy release rate on the combustion event.

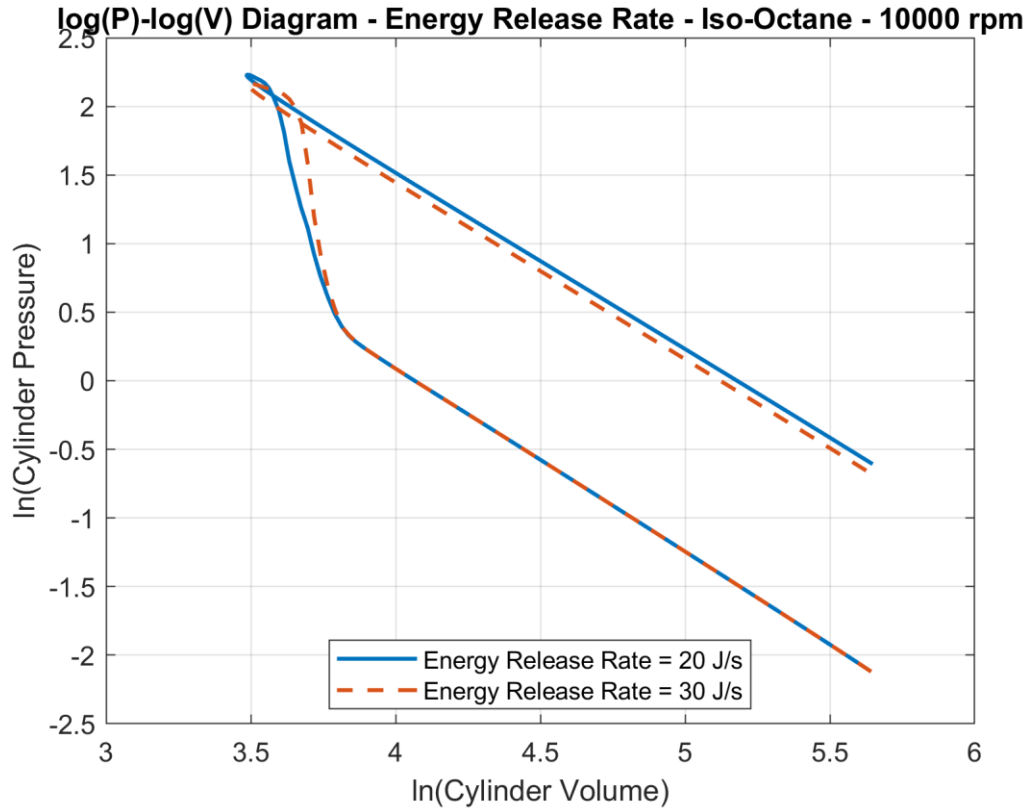


Figure 12: Effect of spark-model energy release rate on the $\log(P)$ - $\log(V)$ response for the iso-octane case at 10,000 rpm.

The influence of the altered combustion event on the thermodynamic cycle is further illustrated by the $\log(P)$ - $\log(V)$ response in Figure 12. As the energy release rate is a spark-model parameter, its influence begins with spark-kernel development; differences occurring during the preceding compression region are therefore not attributed to this parameter. Following combustion, the expansion paths diverge, with the fitted expansion polytropic index increasing from 1.2879 at 20 J/s to 1.2911 at 30 J/s, a difference of approximately 0.25%. The common geometry and expansion kinematics help keep the fitted expansion slopes similar, with n changing only 0.25% despite much larger combustion differences.

Table 6: Sensitivity of selected combustion and cycle parameters to spark-model energy release rate for the iso-octane case at 10,000 rpm.

PARAMETER	20 J/S	30 J/S	CHANGE FROM 20 TO 30 J/S
CA10 (°ATDC)	-18.936	-18.948	0.012°CA earlier
CA50 (°ATDC)	-12.975	-16.962	3.99°CA earlier
CA90 (°ATDC)	-9.917	-14.979	5.06°CA earlier
CA10–90 DURATION (°CA)	9.020	3.969	-56.0%
PEAK APPARENT HRR (J/°CA)	181.97	264.22	+45.2%
MAXIMUM RATE OF CHANGE OF PRESSURE (MPa/°CA)	1.144	1.638	+43.1%
PEAK CYLINDER PRESSURE (MPa)	9.306	8.761	-5.86%
IMEP (MPa)	1.1499	0.9924	-13.70%
WALL HEAT LOSS (J)	157.97	214.25	+35.62%
EXPANSION POLYTROPIC INDEX	1.2879	1.2911	+0.25%

The principal quantitative differences are summarised in Table 6. The earlier and more rapid combustion also alters the predicted cycle response: IMEP decreased from 1.150 to 0.992 MPa (-13.7%), while integrated wall heat loss increased from 157.97 to 214.25 J (+35.6%). The earlier heat release reduces the proportion of energy contributing to useful expansion work while increasing thermal loading of the chamber walls.

Overall, the sensitivity study demonstrates that the spark-model energy release rate has little influence on the compression process but a substantial influence on subsequent combustion development. Across the investigated 20–30 J/s range, CA10 remained effectively unchanged, while the greater spark-kernel energy input accelerated subsequent combustion development, producing earlier phasing, a 56% shorter CA10–90 burn duration, a 45.2% increase in peak apparent heat-release rate and a 43.1% increase in the maximum rate of change of pressure. The resulting changes in IMEP and wall heat loss further demonstrate that the altered combustion behaviour influences the predicted thermodynamic cycle.

The previously selected 25 J/s value was retained for the main simulations as the central value within the investigated 20–30 J/s range. The sensitivity study was intended to characterise the response across this range rather than identify an optimum setting. As the 25 J/s case was not included directly in this sensitivity comparison, the results do not demonstrate that its response lies midway between the two investigated cases or that it represents an experimentally validated optimum. The identified sensitivity should therefore be considered when interpreting subsequent combustion-phasing and burn-rate predictions.

4.1.3 Flame Development Coefficient

A flame development coefficient of 1.5 was selected for the main Forte simulations based on the range identified during model development. To assess the potential numerical uncertainty associated with this parameter, sensitivity simulations were conducted at the bounding values of 0.5 and 2.5 using the iso-octane case at 10,000 rpm. The purpose of this study was not to optimise the coefficient, but to establish how strongly variation across the identified range could affect the predicted combustion and cycle behaviour.

Mass Fraction Burned - Flame Development Coefficient - Iso-Octane - 10000 rpm

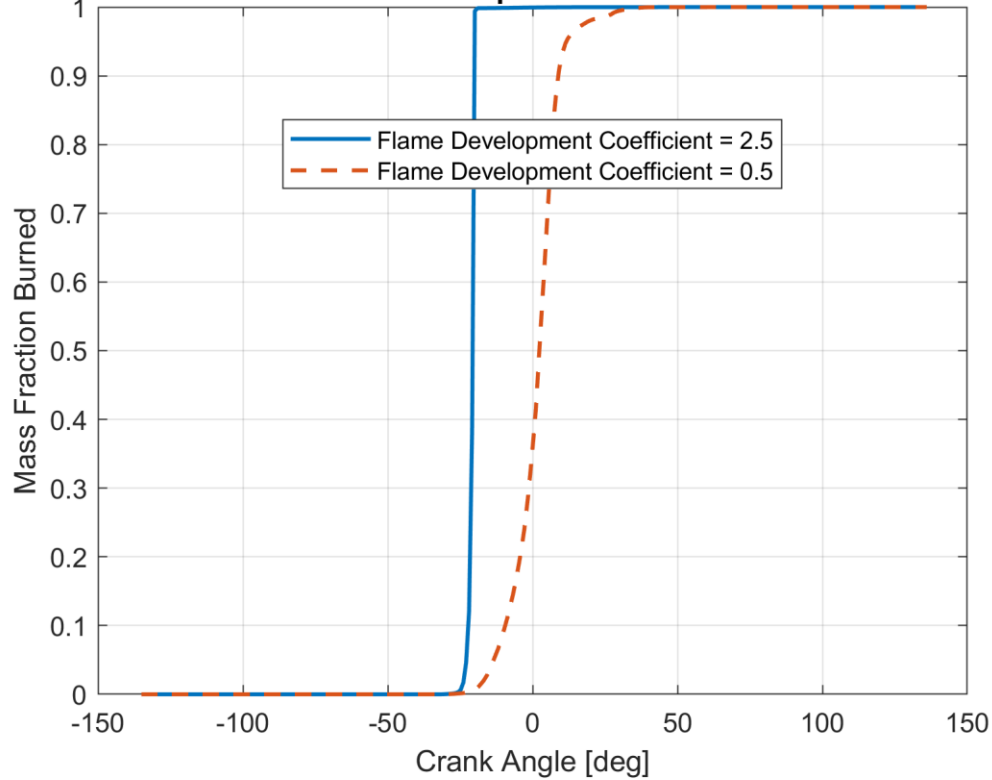


Figure 13: Effect of flame development coefficient on mass fraction burned for iso-octane at 10,000 rpm.

Figure 13 shows a substantial change in both combustion phasing and duration. Increasing the flame development coefficient from 0.5 to 2.5 advanced CA10 from 8.71° BTDC to 21.94° BTDC and CA50 from 3.06° ATDC to 19.96° BTDC. CA90 similarly advanced from 9.10° ATDC to 19.96° BTDC. Consequently, the CA10-90 burn duration decreased from 17.81 to 1.98°CA, a reduction of approximately 88.9%. The higher coefficient accelerates the modelled transition toward turbulent flame propagation, producing the large phasing advances and the 88.9% reduction in burn duration.

Apparent Heat Release Rate - Flame Development Coefficient - Iso-Octane - 10000

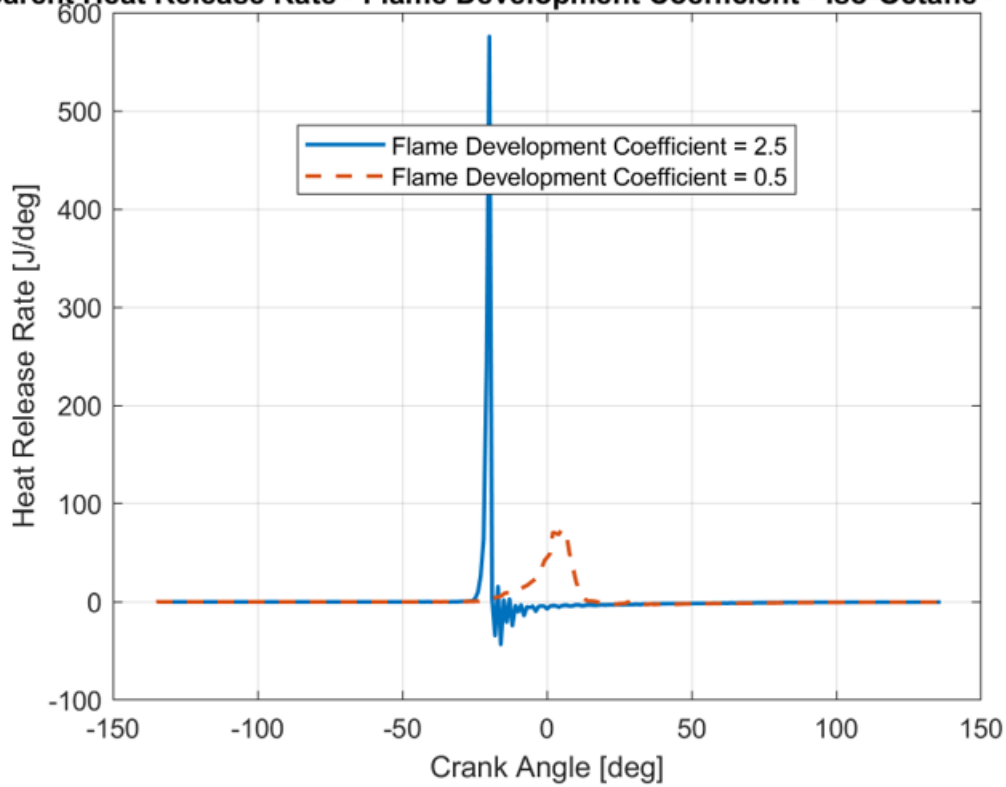


Figure 14: Effect of flame development coefficient on apparent heat-release rate for iso-octane at 10,000 rpm.

The change in combustion intensity is further demonstrated by Figure 14. Because the 2.5 case burns the charge over a much shorter crank-angle interval, the heat release is substantially more concentrated: peak apparent HRR increased from 74.1 to 575.8 J/°CA (+677%) and advanced by approximately 25.1°CA. The resulting rapid energy release also increased the maximum rate of change of pressure from 0.479 to 2.178 MPa/°CA (+355%), occurring approximately 22.9°CA earlier. Peak cylinder pressure changes by only 3.38% because its magnitude reflects both heat-release phasing and cylinder volume rather than combustion rate alone; Pmax therefore substantially understates the sensitivity.

Table 7: Sensitivity of selected combustion and cycle parameters to flame development coefficient for iso-octane at 10,000 rpm.

PARAMETER	COEFFICIENT 0.5	COEFFICIENT 2.5	CHANGE FROM 0.5 TO 2.5
CA10 (°ATDC)	-8.713	-21.940	13.23°CA earlier
CA50 (°ATDC)	3.058	-19.960	23.02°CA earlier
CA90 (°ATDC)	9.098	-19.960	29.06°CA earlier
CA10-90 DURATION (°CA)	17.811	1.979	-88.9%
PEAK APPARENT HEAT-RELEASE RATE (J/°CA)	74.06	575.77	+677.4%
MAXIMUM RATE OF CHANGE OF PRESSURE (MPA/°CA)	0.479	2.178	+355.1%
PEAK CYLINDER PRESSURE (MPA)	8.033	7.762	-3.38%
IMEP (MPA)	1.339	0.776	-42.0%
WALL HEAT LOSS (J)	70.85	246.31	+247.6%
EXPANSION POLYTROPIC INDEX	1.2843	1.2966	+0.96%

As summarised in Table 7, the altered combustion behaviour also propagates into cycle performance. Despite only a 3.38% difference in peak cylinder pressure, IMEP decreased by 42.0% and integrated wall heat loss increased by 247.6%. Combustion efficiency changed by only 0.51 percentage points, from 75.43% to 74.92%, showing that combustion completeness remained similar. Instead, the 2.5 case releases much more energy before TDC, increasing negative-work potential and thermal loading, which explains the lower useful indicated work and greater wall heat loss.

Cylinder Pressure - Flame Development Coefficient - Iso-Octane - 10000 rpm

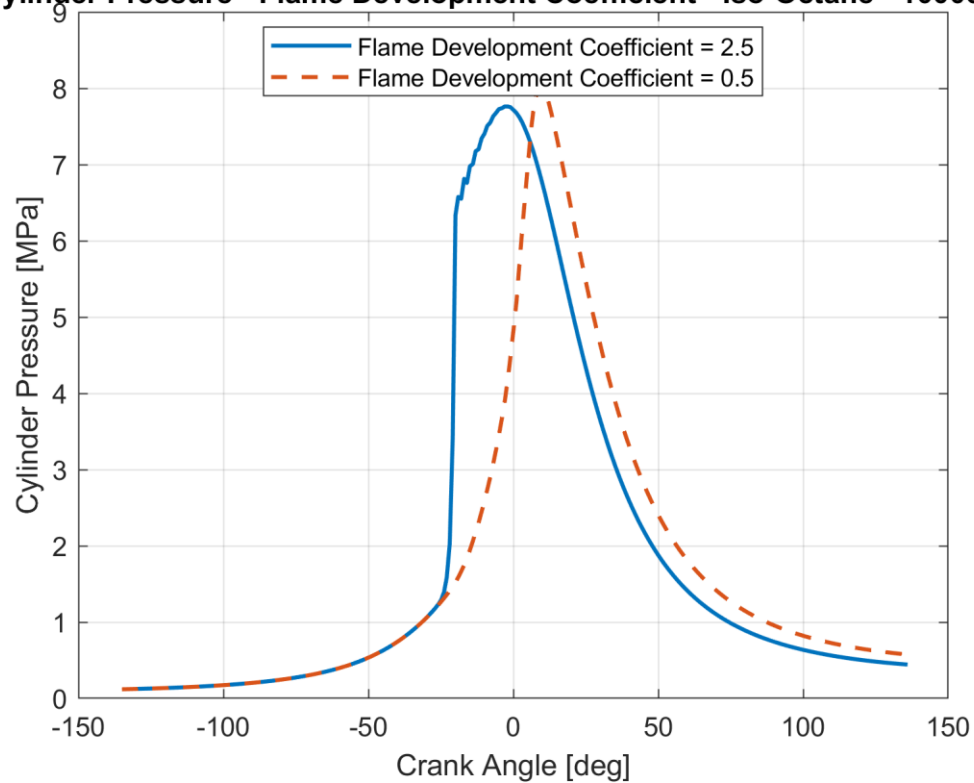


Figure 15: Effect of flame development coefficient on cylinder pressure for iso-octane at 10,000 rpm.

Figure 15 further demonstrates the effect of the altered combustion phasing on the cylinder-pressure history. Although the peak pressures remain relatively similar at 8.033 MPa and 7.762 MPa for coefficients of 0.5 and 2.5, respectively, the increase in cylinder pressure occurs substantially earlier for the 2.5 case, consistent with its considerably advanced combustion phasing.

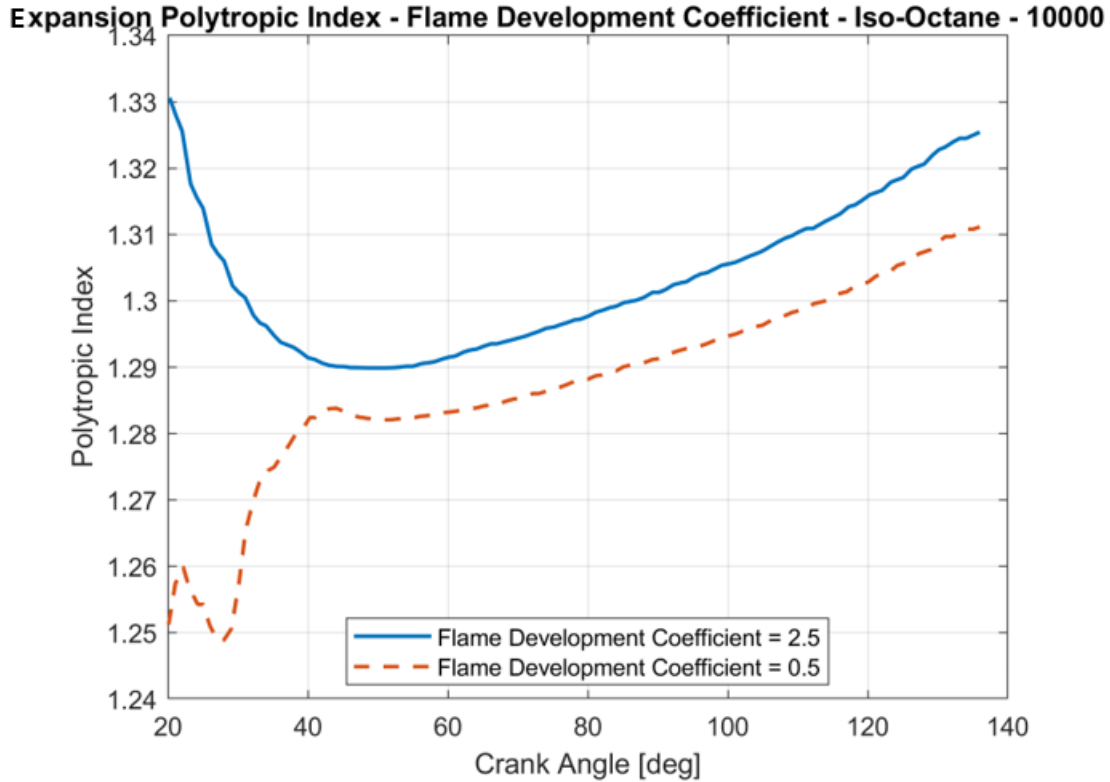


Figure 16: Effect of flame development coefficient on expansion polytropic index for iso-octane at 10,000 rpm.

Figure 16 shows that the altered combustion event also influences the subsequent expansion behaviour. The fitted expansion polytropic index increased from 1.2843 at a coefficient of 0.5 to 1.2966 at 2.5, a difference of approximately 0.96%. Despite the different combustion histories, common expansion geometry and piston motion help keep the fitted slopes similar, with the expansion polytropic index changing by only 0.96%.

The results therefore identify the flame development coefficient as a significant source of potential modelling uncertainty. Across the investigated range, substantial variation occurs in combustion phasing, burn duration, apparent heat-release rate and the resulting cycle work. The selected value of 1.5 was retained for the main simulations as the central value within the investigated range of 0.5 to 2.5. However, the sensitivity study does not demonstrate that 1.5 is an optimum or experimentally validated value, nor can its numerical response be assumed to lie midway between the two bounding cases. The sensitivity identified here should therefore be considered when interpreting the subsequent Forte combustion and performance predictions.

4.1.4 Maximum Time Step

Maximum time steps of 5×10^{-6} and 1×10^{-5} s were investigated using the iso-octane case at 10,000 rpm. Both values were identified as literature-supported settings during model development. The purpose of the comparison was to assess the sensitivity of the predicted solution to temporal resolution and to select a suitable maximum time step for the subsequent simulations. With only two values investigated, the study is treated as a sensitivity and selection exercise rather than evidence of formal time-step convergence.

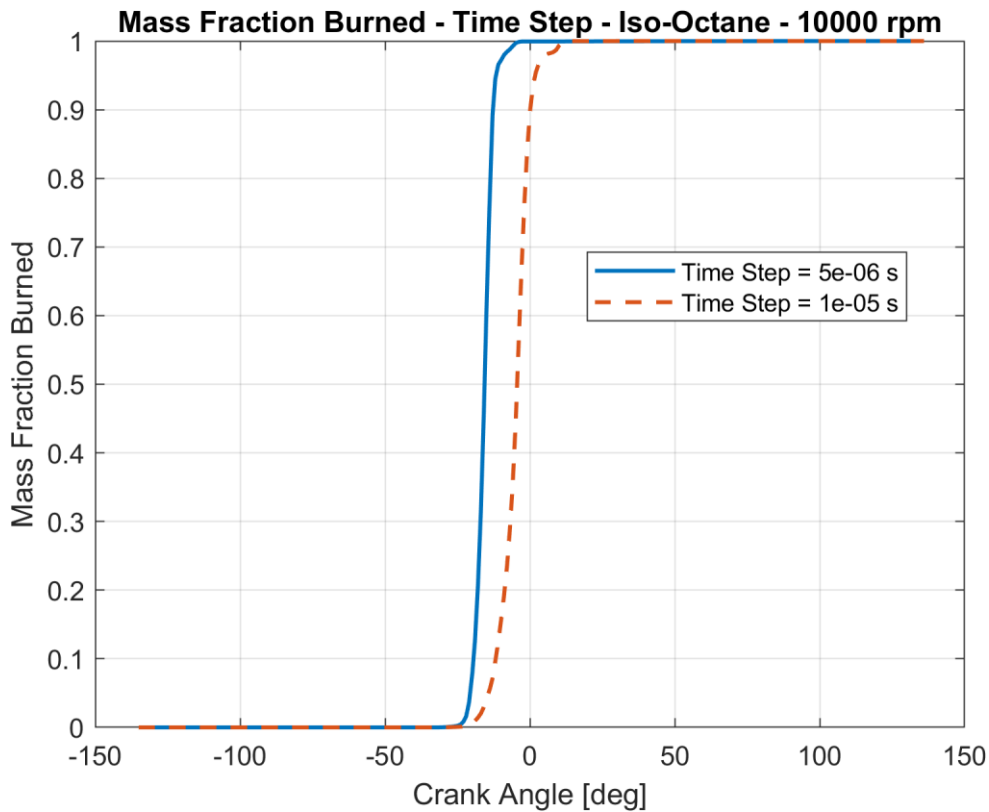


Figure 17: Effect of maximum time step on mass fraction burned for iso-octane at 10,000 rpm.

Figure 17 demonstrates that the principal sensitivity occurs during combustion. CA50 changes from approximately 15.0° BTDC at 5×10^{-6} s to 4.0° BTDC at 1×10^{-5} s, a retard of approximately 11°CA. The CA10–90 duration simultaneously increases from 7.01 to 12.05°CA, representing an increase of approximately 71.8%. These relatively large differences occur because combustion is a rapid, strongly coupled transient process in which flame development, heat release, temperature and pressure evolve over a short crank-angle interval. At 10,000 rpm, 5×10^{-6} and 1×10^{-5} s correspond to maximum increments of approximately 0.30 and 0.60°CA, respectively. Changing the maximum time step therefore changes the temporal resolution available during this rapid event, producing substantial differences in predicted combustion phasing and duration.

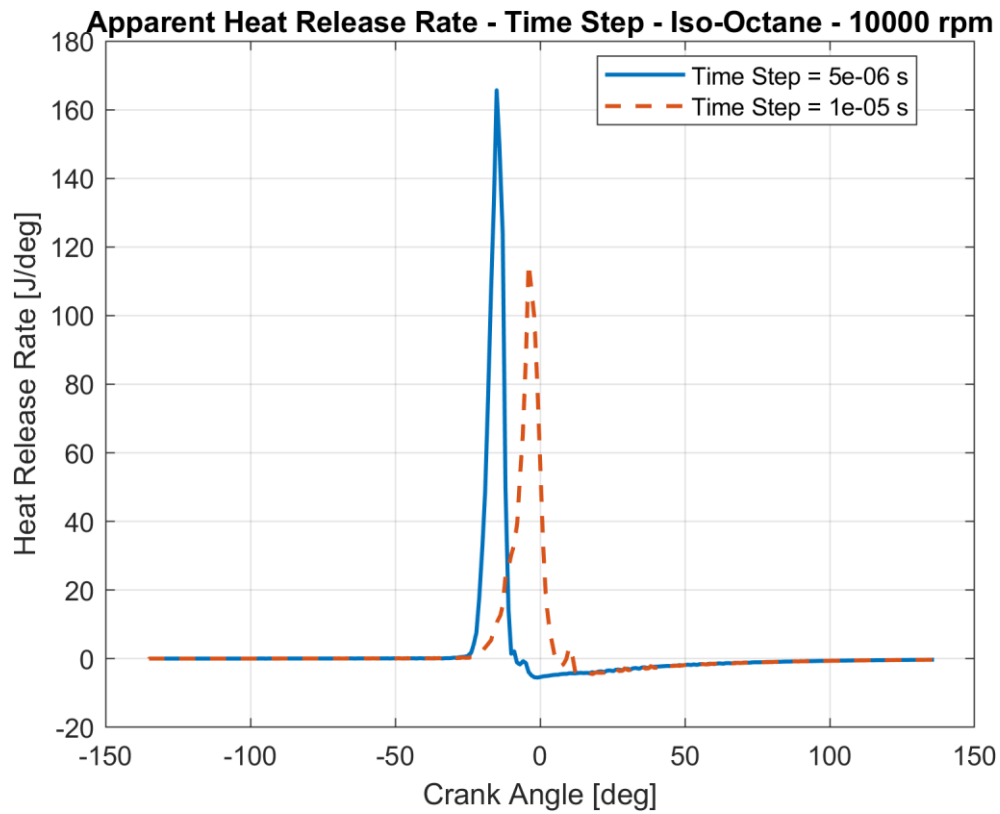


Figure 18: Effect of maximum time step on apparent heat-release rate for iso-octane at 10,000 rpm.

Figure 18 shows that the earlier and shorter combustion event predicted using 5×10^{-6} s is accompanied by a substantially more concentrated heat-release event. Peak apparent heat-release rate decreases from 165.68 to approximately 115.2 J/°CA when the maximum time step is increased to 1×10^{-5} s, a reduction of approximately 30.5%. This comparatively large difference is consistent with the 71.8% increase in CA_{10–90} duration, as distributing combustion over a wider crank-angle interval reduces the instantaneous rate at which energy is released.

Table 8: Sensitivity of selected combustion parameters to maximum time step for iso-octane at 10,000 rpm.

PARAMETER	5×10^{-6} S	1×10^{-5} S	DIFFERENCE
CA50 (°ATDC)	-14.98	≈ -3.96	≈ 11.0°CA later
CA10-90 DURATION (°CA)	7.01	≈ 12.05	≈ +71.9%
PEAK APPARENT HEAT-RELEASE RATE (J/°CA)	165.68	≈ 115.2	≈ -30.5%
PEAK BURN RATE (°CA)	≈ 0.156	≈ 0.112	≈ -28%
PEAK CYLINDER PRESSURE (MPA)	9.260	9.045	-2.33%
COMPRESSION POLYTROPIC INDEX	1.3443	1.3429	≈ 0.10% difference
EXPANSION POLYTROPIC INDEX	1.2897	1.2904	≈ 0.05% difference
MAXIMUM RATE OF CHANGE OF PRESSURE (MPA/°CA)	1.091	0.833	-23.6%
PEAK TEMPERATURE (K)	2507.8	2454.2	-2.14%
IMEP (MPA)	1.112	1.296	+16.6%
WALL HEAT LOSS (J)	173.64	99.92	-42.5%
COMBUSTION EFFICIENCY (%)	76.04	75.45	-0.59 pp

As summarised in Table 8, increasing the maximum time step from 5×10^{-6} to 1×10^{-5} s reduces the peak burn rate by approximately 28% and the peak apparent heat-release rate by 30.5%. These comparatively large reductions are consistent with the 71.8% increase in CA10–90 duration, as distributing combustion over a wider crank-angle interval reduces the instantaneous rates of burning and energy release. The maximum rate of change of pressure consequently decreases by 23.6% when the maximum time step is increased to 1×10^{-5} s, because the less concentrated heat release produces a less rapid pressure response. In contrast, peak cylinder pressure and peak temperature differ by only 2.33% and 2.14%, respectively. These comparatively small differences occur because pressure and temperature are thermodynamic state quantities influenced by the accumulated effects of compression and combustion, whereas burn rate and apparent heat-release rate respond directly to how rapidly combustion occurs. Similar peak values can therefore occur despite substantially different combustion phasing and rates.

The fitted compression and expansion polytropic indices differ by only approximately 0.10% and 0.06%, respectively. These small differences are expected because the compression and later expansion periods evolve more gradually than the rapid combustion event and are governed by the same engine geometry and piston motion in both simulations. The maximum time-step sensitivity is therefore concentrated predominantly within the short combustion period rather than producing a comparable change throughout the complete thermodynamic cycle.

The altered combustion phasing also changes the distribution of energy through the cycle. Increasing the maximum time step to 1×10^{-5} s increases IMEP by approximately 16.6% while reducing wall heat loss by 42.5%. The later and more prolonged combustion event reduces the period of high-temperature exposure before and around TDC and shifts a greater proportion of the heat release towards TDC and the expansion stroke, reducing wall heat transfer and pre-TDC negative work. In comparison, combustion efficiency decreases by only 0.59 percentage points, indicating that the large differences in IMEP and wall heat

loss arise primarily from changes in the timing and rate of energy release rather than a substantial change in overall combustion completeness.

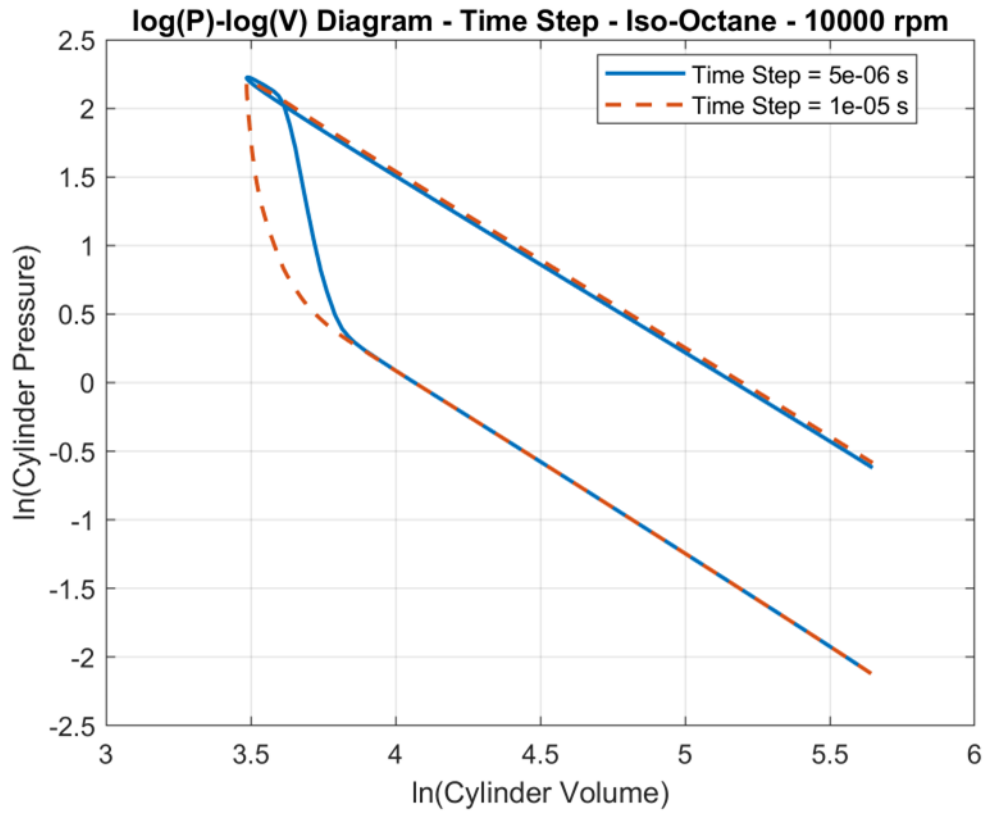


Figure 19: Effect of maximum time step on the $\log(P)$ - $\log(V)$ response for iso-octane at 10,000 rpm.

Figure 19 shows that the two cases follow closely similar $\log(P)$ - $\log(V)$ behaviour during compression but diverge around the combustion period. This is consistent with the quantitative results in Table 8, where the compression polytropic index changes by only approximately 0.10% while the combustion phasing and rate parameters show substantially larger differences. The $\log(P)$ - $\log(V)$ response therefore provides a visual representation of where the time-step sensitivity develops within the engine cycle, rather than being used independently to determine which maximum time step is preferable.

On the available evidence, 1×10^{-5} s was selected as the maximum time step for the subsequent Forte simulations. Both investigated values are literature-supported, and the smaller maximum time step cannot be assumed to provide greater physical accuracy solely because it provides finer temporal resolution. The comparison shows that compression and expansion behaviour remain essentially unchanged between the two settings, whereas the major sensitivity occurs during combustion, with an approximately 11°CA change in CA50 and substantial differences in burn duration, apparent heat-release rate and maximum rate of change of pressure. The 1×10^{-5} s value was therefore retained as a practical literature-supported setting for the wider study, while the observed combustion sensitivity is acknowledged as a numerical limitation. Experimental combustion-phasing data would be required to establish which predicted response more closely represents the physical engine.

4.2 Simulation Test Matrix and Comparison Method

Following completion of the numerical verification studies, the final model configurations were used to conduct the main simulation programme. Four fuels were investigated: octane, methanol, ethanol and hydrogen. Each fuel was simulated independently in both Ricardo WAVE and ANSYS Forte, with corresponding engine geometry and operating conditions maintained between the models where applicable.

For each fuel, engine speed was varied from 1,000 to 10,000 rpm in increments of 1,000 rpm. This provided ten operating points for each fuel in each model, resulting in 40 operating points in Ricardo WAVE and 40 in ANSYS Forte. In total, 80 operating points were therefore simulated. Using identical engine-speed intervals for each fuel and modelling approach ensured that results could be compared at corresponding operating points throughout the investigated speed range.

Octane was used as the baseline fuel for the fuel comparisons. The Ricardo WAVE results for methanol, ethanol and hydrogen were each compared against the corresponding octane results across the engine-speed range. The same comparison method was then applied independently to the ANSYS Forte results. This allowed the effect of changing fuel to be examined within each modelling approach before differences between the two simulation methods were considered.

The Ricardo WAVE and ANSYS Forte predictions were subsequently compared directly on a fuel-by-fuel basis. Corresponding results for octane, methanol, ethanol and hydrogen were compared at each engine speed, allowing differences between the 1D and 3D predictions to be examined across the operating range. The comparison considered the predicted pressure, heat release, temperature, flame development, fuel consumption, volumetric efficiency, knock tendency and emissions produced by the two models.

Neither model was treated as a reference solution against which the other was judged. Instead, agreement and disagreement between the predicted outputs were identified and considered in relation to the different modelling approaches and assumptions used by Ricardo WAVE and ANSYS Forte. This provided a consistent method for assessing both the effect of fuel selection within each model and the relationship between the predictions produced by the 1D and 3D simulation methods.

5 Results and Discussion

5.1 Fuel Effects Predicted by WAVE and Forte

5.1.1 Octane Baseline Results

The WAVE 1000 rpm case exhibited anomalous combustion behaviour, with combustion extending beyond the expected cycle interval. It is retained within the reported dataset for completeness but treated separately from the principal Iso-Octane speed-dependent behaviour.

5.1.1.1 Combustion Behaviour

The combustion characteristics of WAVE are presented in figures 20 & 21, with the corresponding combustion phasing and heat-release parameters given in table 9.

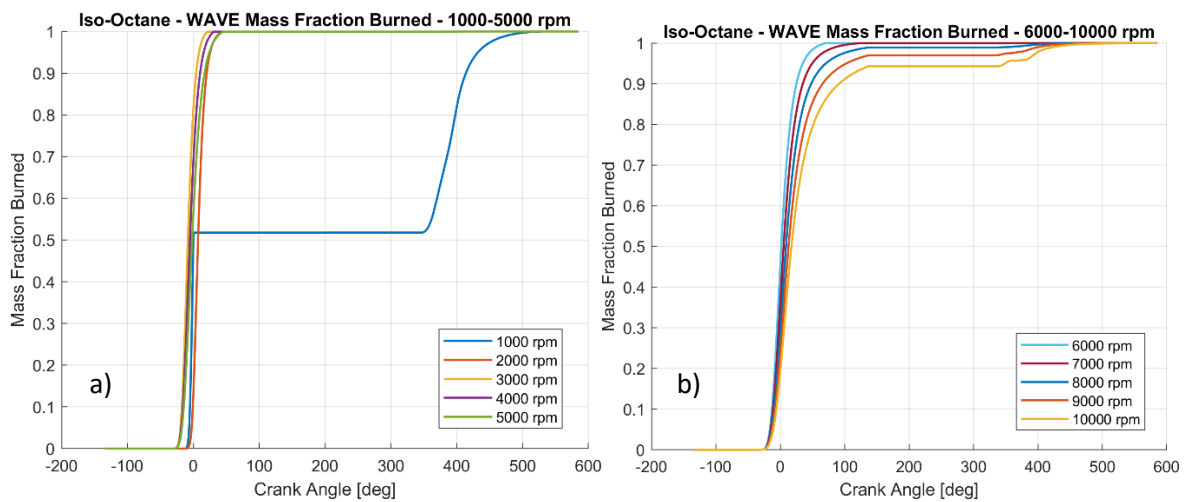


Figure 20: Iso-Octane WAVE mass fraction burned: a) 1000–5000 rpm, b) 6000–10000 rpm

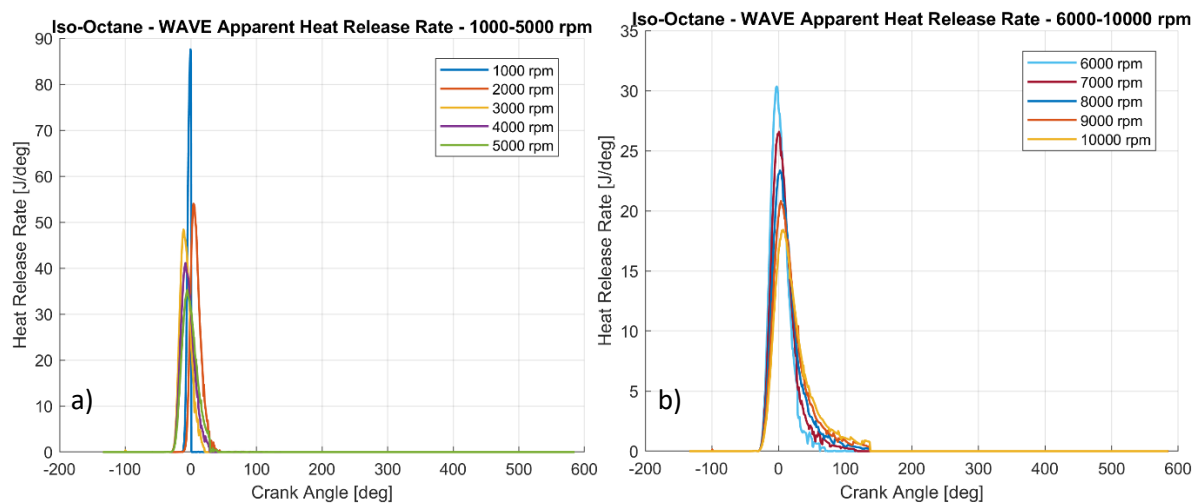


Figure 21: Iso-Octane WAVE apparent heat-release rate: a) 1000–5000 rpm, b) 6000-10000 rpm

Table 9: Iso-Octane WAVE combustion characteristics.

RPM	CA10 (°ATDC)	CA50 (°ATDC)	CA90 (°ATDC)	CA10–90 (°CA)	PEAK APPARENT HRR (J/°CA)	PEAK HRR CA (°ATDC)
1000	−4.93	0.19	414.60	419.53	87.61	−0.54
2000	−0.70	7.30	20.88	21.57	54.05	4.10
3000	−16.98	−7.81	4.69	21.67	48.40	−11.14
4000	−15.99	−4.73	10.85	26.84	41.14	−8.20
5000	−13.83	−1.45	17.12	30.95	35.25	−5.32
6000	−12.98	1.88	25.10	38.09	30.36	−2.76
7000	−10.57	5.92	35.40	45.97	26.59	0.71
8000	−8.78	9.08	48.73	57.51	23.37	2.13
9000	−7.33	13.60	66.30	73.64	20.82	3.55
10000	−5.33	17.93	91.37	96.70	18.43	6.76

The combustion characteristics of Forte are presented in figures 22 & 23, with the corresponding combustion phasing and heat-release parameters given in table 10.

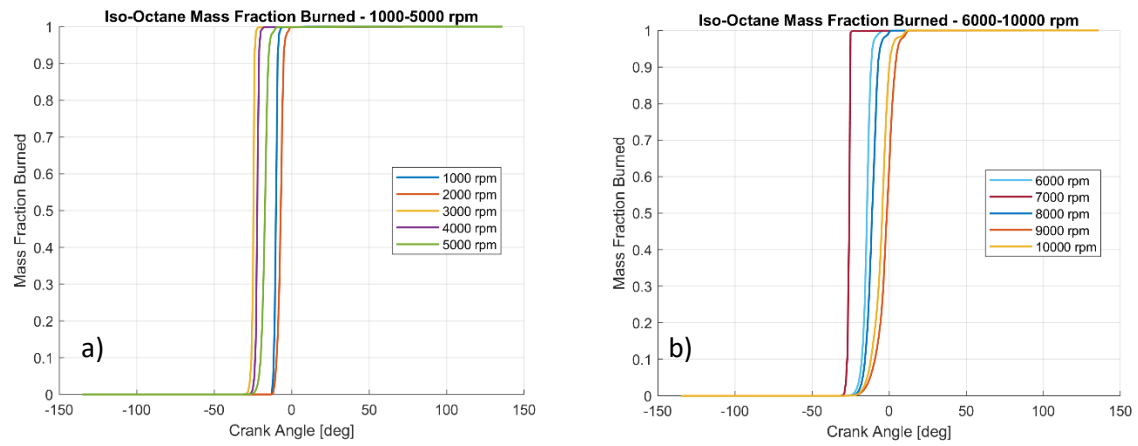


Figure 22: Iso-Octane Forte mass fraction burned: a) 1000–5000 rpm, b) 6000-10000 rpm

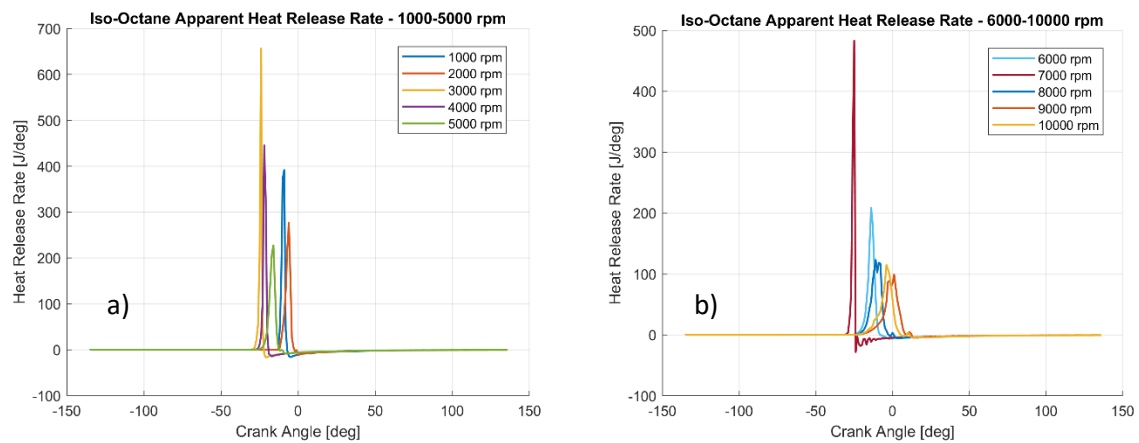


Figure 23: Iso-Octane Forte apparent heat-release rate: a) 1000–5000 rpm, b) 6000-10000 rpm

Table 10: Iso-Octane Forte combustion characteristics.

RPM	CA10 (°ATDC)	CA50 (°ATDC)	CA90 (°ATDC)	CA10–90 (°CA)	PEAK APPARENT HRR (J/°CA)	PEAK HRR CA (°ATDC)
1000	-10.99	-10.00	-8.99	2.00	391.80	-8.99
2000	-9.00	-6.96	-5.00	4.00	277.12	-5.98
3000	-25.00	-23.99	-23.99	1.01	656.71	-23.99
4000	-22.99	-21.98	-20.98	2.01	445.68	-21.98
5000	-19.00	-16.97	-14.99	4.01	227.50	-15.98
6000	-16.96	-13.93	-10.82	6.13	208.46	-13.93
7000	-26.98	-24.99	-24.99	2.00	483.42	-24.99
8000	-14.99	-10.98	-6.94	8.05	123.42	-10.98
9000	-8.98	-0.92	4.21	13.19	99.14	1.04
10000	-10.98	-3.96	1.07	12.05	115.20	-3.96

WAVE predicts increasingly delayed and prolonged iso-octane combustion above 3000 rpm. CA50 moves from -7.81° ATDC at 3000 rpm to 17.93° ATDC at 10,000 rpm, while CA10–90 increases from 21.67 to 96.70° CA and peak apparent HRR decreases from 48.40 to 18.43 J/°CA. Forte predicts substantially shorter combustion across the speed range, with CA10–90 increasing from 1.01° CA at 3000 rpm to 12.05° CA at 10,000 rpm, while peak HRR decreases from 656.71 to 115.20 J/°CA. Forte CA50 varies more strongly between individual speeds, including -23.99° at 3000 rpm, -24.99° at 7000 rpm and -3.96° at 10,000 rpm, whereas WAVE develops a clearer progressive high-speed retardation.

5.1.1.2 Cylinder Pressure and Temperature

The WAVE cylinder-pressure and temperature characteristics are presented in figures 24 & 25, with the corresponding peak values and crank-angle locations given in table 11.

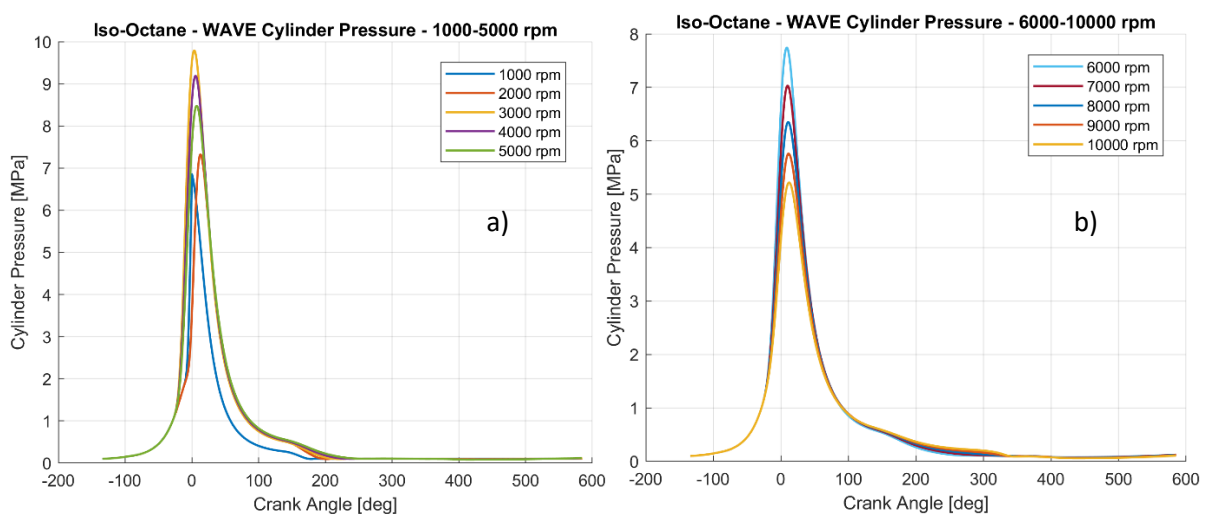


Figure 24: Iso-Octane WAVE cylinder pressure: a) 1000–5000 rpm, b) 6000–10000 rpm

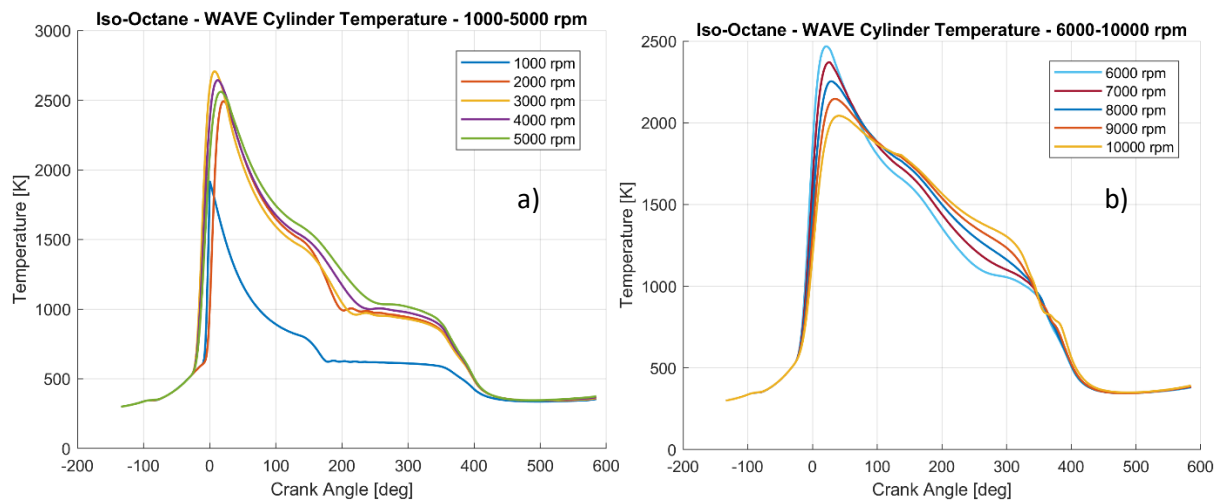


Figure 25: Iso-Octane WAVE cylinder temperature: a) 1000–5000 rpm, b) 6000–10000

Table 11: Iso-Octane WAVE cylinder pressure and temperature characteristics

RPM	PMAX (MPA)	PMAX CA (°ATDC)	MAX. RATE OF CHANGE OF PRESSURE (MPA/°CA)	TMAX (K)	TMAX CA (°ATDC)
1000	6.857	0.19	0.775	1916	0.19
2000	7.324	12.89	0.453	2492	20.88
3000	9.791	3.86	0.522	2707	7.19
4000	9.191	5.65	0.453	2643	11.71
5000	8.476	7.06	0.387	2560	16.34
6000	7.743	8.39	0.329	2468	21.39
7000	7.031	10.25	0.279	2370	25.87
8000	6.351	11.06	0.236	2254	28.91
9000	5.760	11.08	0.202	2146	34.52
10000	5.218	12.34	0.172	2043	41.18

The Forte cylinder-pressure and temperature characteristics are presented in Figures 26 & 27, with the corresponding peak values and crank-angle locations given in Table 12.

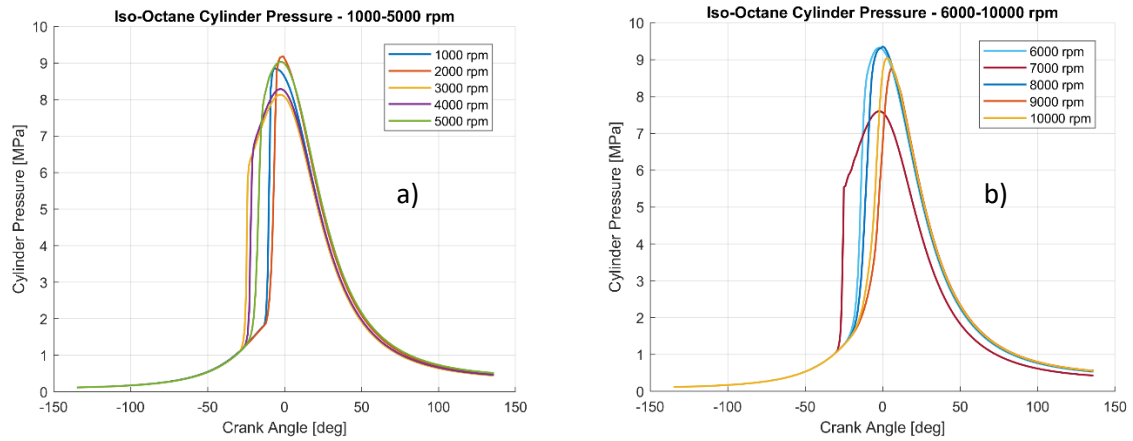


Figure 26: Iso-Octane Forte cylinder pressure: a) 1000–5000 rpm, b) 6000-10000 rpm

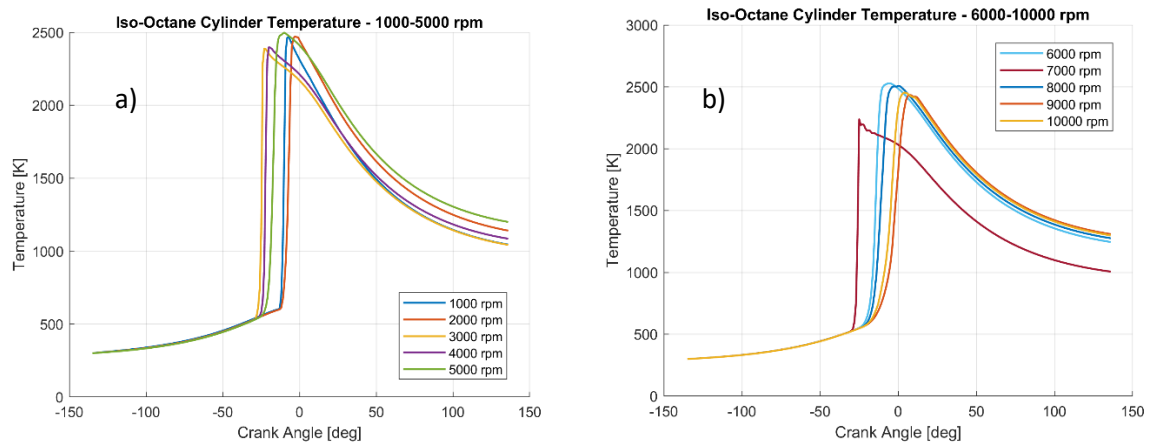


Figure 27: Iso-Octane Forte cylinder temperature: a) 1000–5000 rpm, b) 6000-10000 rpm

Table 12: Iso-Octane Forte cylinder pressure and temperature characteristics.

RPM	PMAX (MPa)	PMAX CA (°ATDC)	MAX RATE OF CHANGE OF PRESSURE (MPa/°CA)	TMAX (K)	TMAX CA (°ATDC)
1000	8.854	-5.98	2.560	2465	-7.00
2000	9.189	-0.97	1.752	2472	-2.98
3000	8.132	-2.96	2.081	2389	-22.99
4000	8.287	-2.76	2.060	2398	-19.98
5000	9.031	-1.92	1.398	2498	-9.94
6000	9.319	-1.99	1.333	2526	-5.99
7000	7.601	-2.00	1.930	2237	-24.99

8000	9.351	0.00	0.901	2508	0.00
9000	8.763	6.18	0.664	2434	7.04
10000	9.045	3.16	0.833	2454	4.08

WAVE peak pressure reaches 9.79 MPa at 3000 rpm before decreasing progressively to 5.22 MPa at 10,000 rpm. Over the same range, maximum rate of change of pressure falls from 0.522 to 0.172 MPa/°CA and peak temperature from 2707 to 2043 K, corresponding with the increasingly delayed combustion. Forte exhibits a less monotonic speed response, with peak pressure ranging from 7.60 MPa at 7000 rpm to 9.35 MPa at 8000 rpm before reaching 9.04 MPa at 10,000 rpm. Peak temperature similarly varies between 2237 and 2526 K from 3000–10,000 rpm, while the maximum rate of change of pressure decreases to 0.833 MPa/°CA at 10,000 rpm.

5.1.1.3 Engine performance

The WAVE indicated performance characteristics are presented in Figure 28, with the corresponding single-cylinder values given in Table 13.

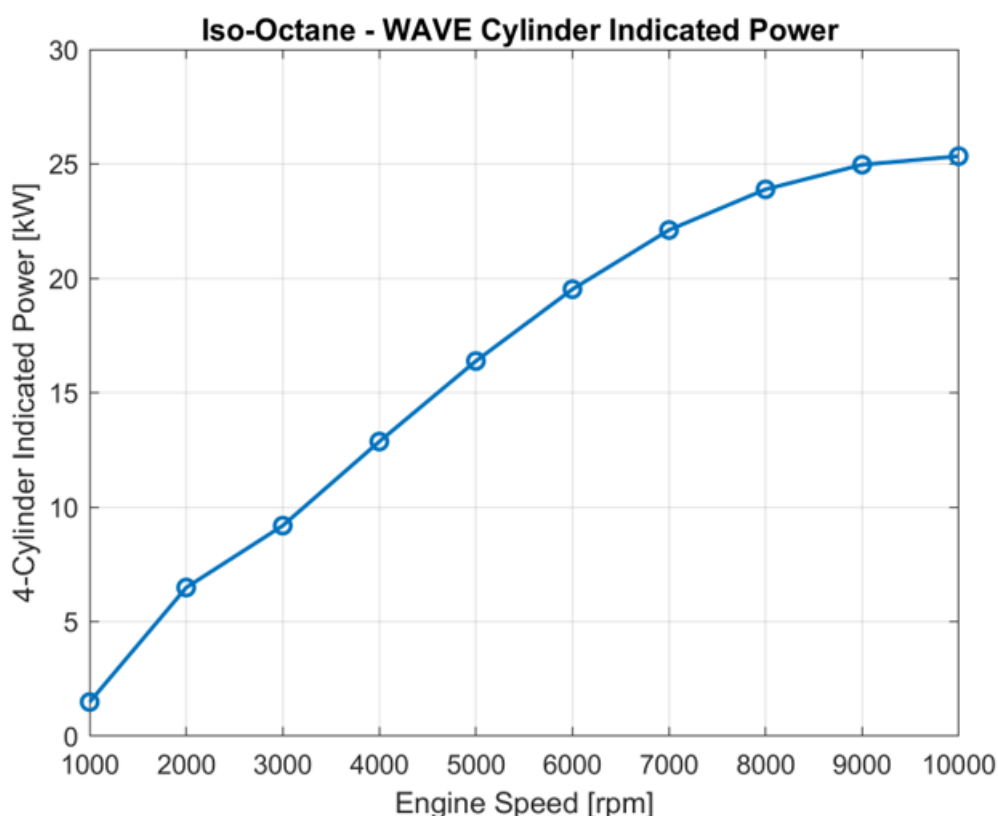


Figure 28: Iso-Octane WAVE indicated engine performance across engine speed

Table 13: Iso-Octane WAVE single-cylinder indicated performance.

RPM	IMEP (MPa)	INDICATED WORK/CYCLE (KJ)	INDICATED POWER (KW)
1000	0.633	0.0446	1.487
2000	1.381	0.0973	6.488
3000	1.304	0.0919	9.193
4000	1.369	0.0965	12.867
5000	1.395	0.0983	16.385
6000	1.385	0.0976	19.522
7000	1.344	0.0948	22.110
8000	1.271	0.0896	23.888
9000	1.181	0.0832	24.964
10000	1.079	0.0760	25.338

The Forte indicated performance characteristics are presented in Figure 29, with the corresponding single-cylinder values given in Table 14.

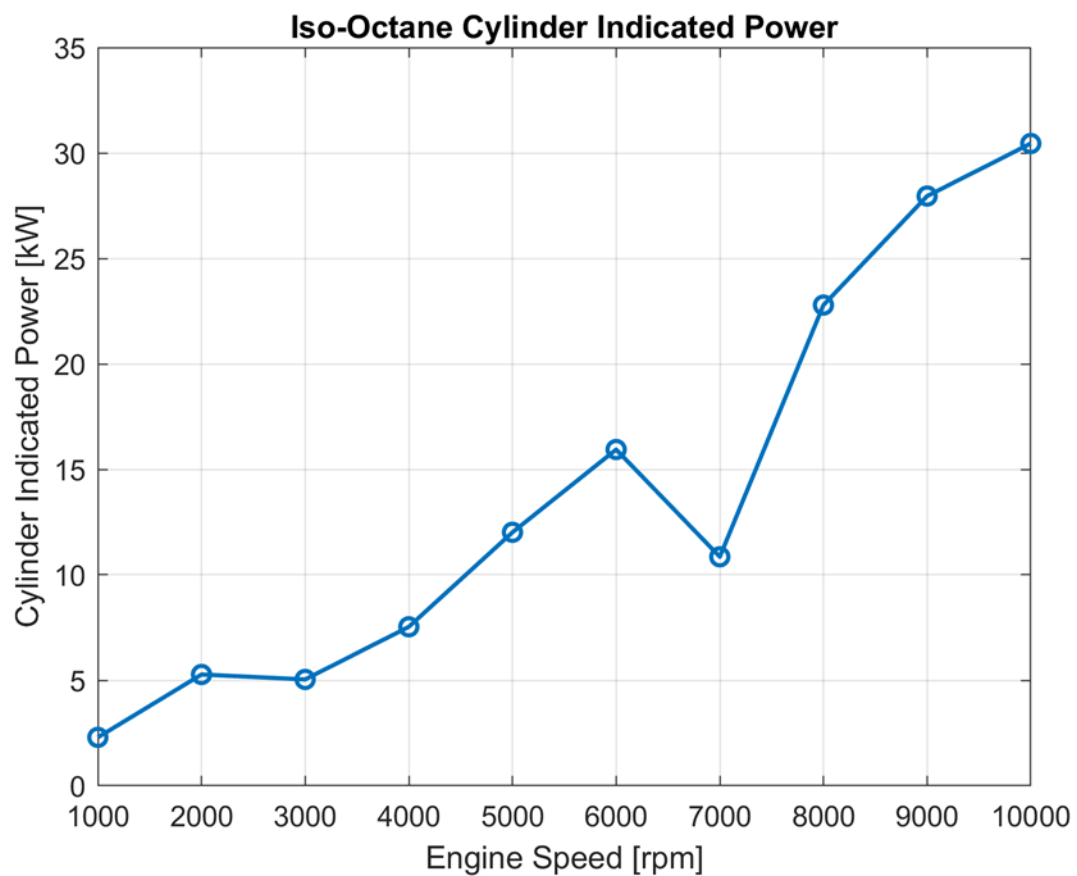


Figure 29: Iso-Octane Forte indicated engine performance across engine speed

Table 14: Iso-Octane Forte single-cylinder indicated performance

RPM	IMEP (MPa)	INDICATED WORK/CYCLE (KJ)	INDICATED POWER (KW)
1000	0.976	0.2752	9.17
2000	1.123	0.3165	21.10
3000	0.715	0.2016	20.16
4000	0.802	0.2261	30.14
5000	1.023	0.2884	48.07
6000	1.131	0.3188	63.77
7000	0.660	0.1861	43.42
8000	1.213	0.3419	91.16
9000	1.322	0.3728	111.83
10000	1.296	0.3654	121.80

WAVE IMEP remains around 1.30–1.40 MPa through much of the intermediate-speed range before decreasing to 1.079 MPa at 10,000 rpm, with indicated work falling from 0.0983 kJ/cycle at 5000 rpm to 0.0760 kJ/cycle. Increasing speed nevertheless raises indicated power from 16.38 kW at 5000 rpm to 25.34 kW at 10,000 rpm. Forte shows greater speed-to-speed variation, with IMEP decreasing to 0.660 MPa at 7000 rpm before rising to 1.322 MPa at 9000 rpm and 1.296 MPa at 10,000 rpm. Indicated work follows the same behaviour, reaching 0.3654 kJ/cycle at 10,000 rpm, while indicated power increases overall with engine speed.

5.1.1.4 Fuel Consumption and Efficiency

WAVE fuel consumption and thermal efficiency are presented in Figures 30, 31, & 32, with the corresponding numerical values given in Table 15.

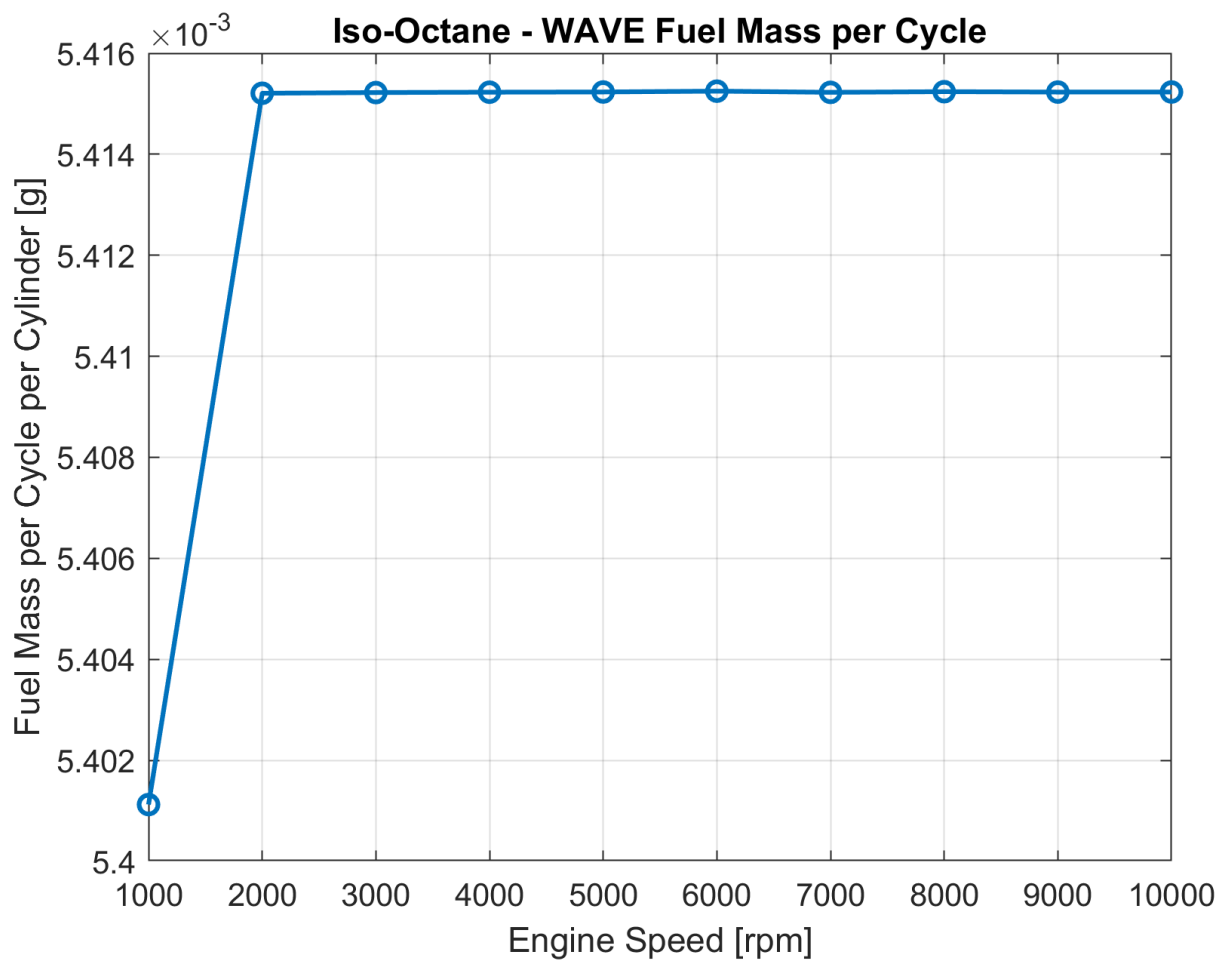


Figure 30: Iso-Octane WAVE fuel mass per cycle across engine speed

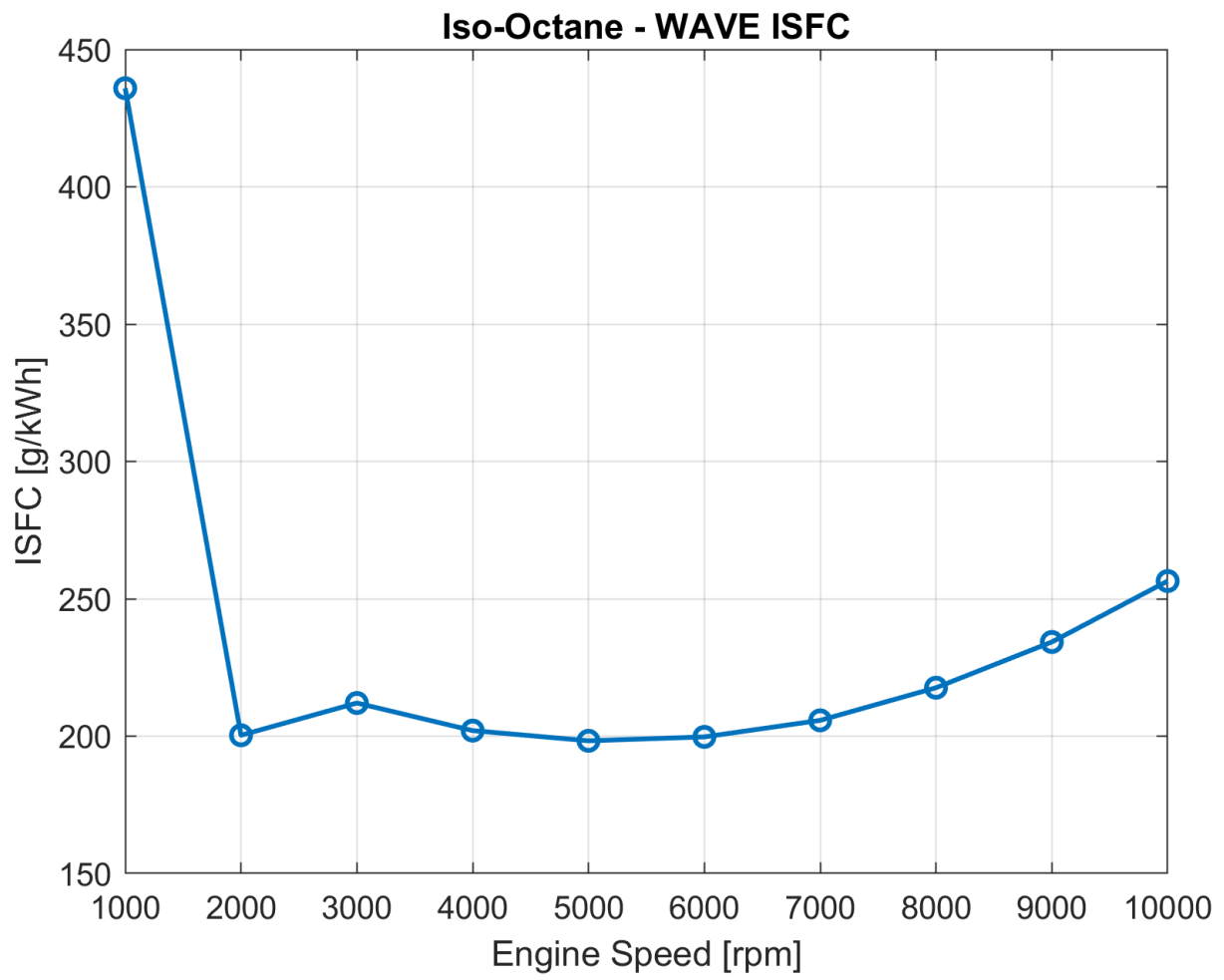


Figure 31: Iso-Octane WAVE indicated specific fuel consumption across engine speed

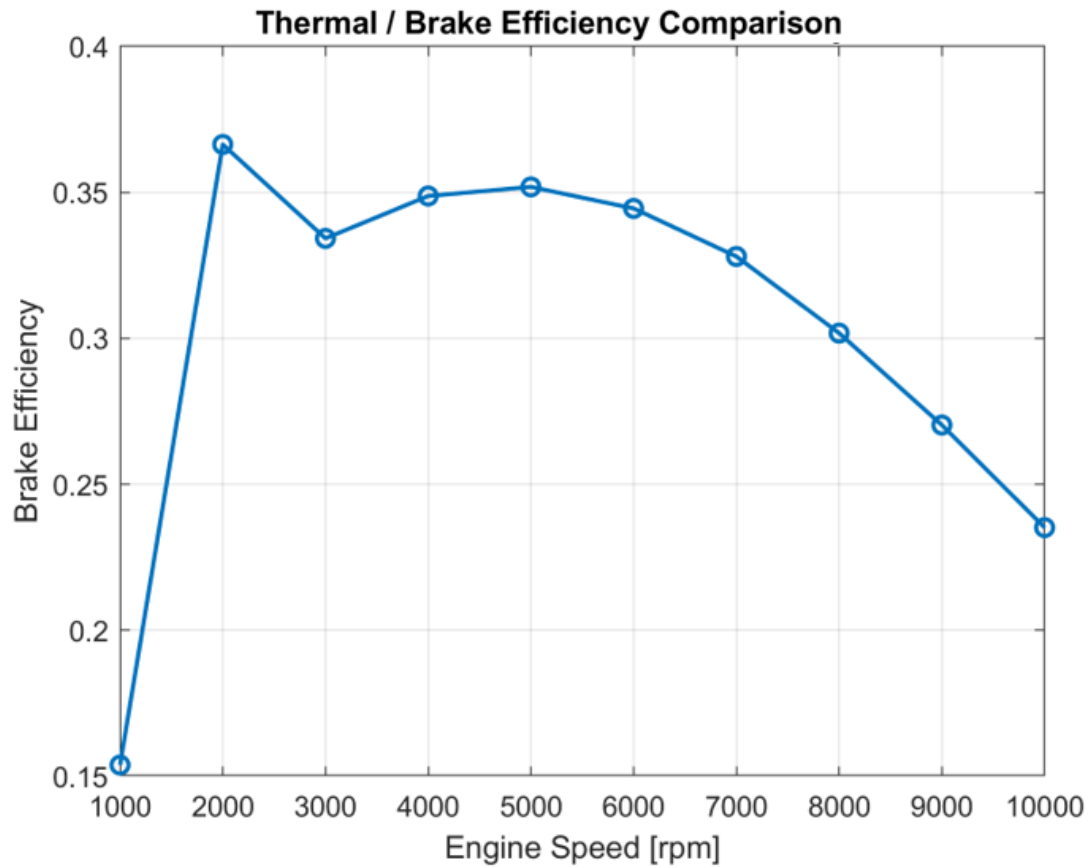


Figure 32: Iso-Octane WAVE thermal efficiency across engine speed

Table 15: Iso-Octane WAVE fuel consumption and efficiency characteristics

ENGINE SPEED (RPM)	FUEL MASS/CYCLE (MG)	FUEL FLOW (KG/H)	ISFC (G/KWH)	THERMAL EFFICIENCY (%)
1000	5.401	0.648	435.9	15.39
2000	5.415	1.300	200.3	36.64
3000	5.415	1.949	212.1	33.42
4000	5.415	2.599	202.0	34.87
5000	5.415	3.249	198.3	35.19
6000	0.005415	3.899	199.7	34.45
7000	0.005415	4.549	205.7	32.80
8000	0.005415	5.199	217.6	30.18
9000	0.005415	5.848	234.3	27.03
10000	0.005415	6.498	256.5	23.52

Forte fuel consumption and thermal efficiency are presented in Figures 33, & 34 with the corresponding numerical values given in Table 16.

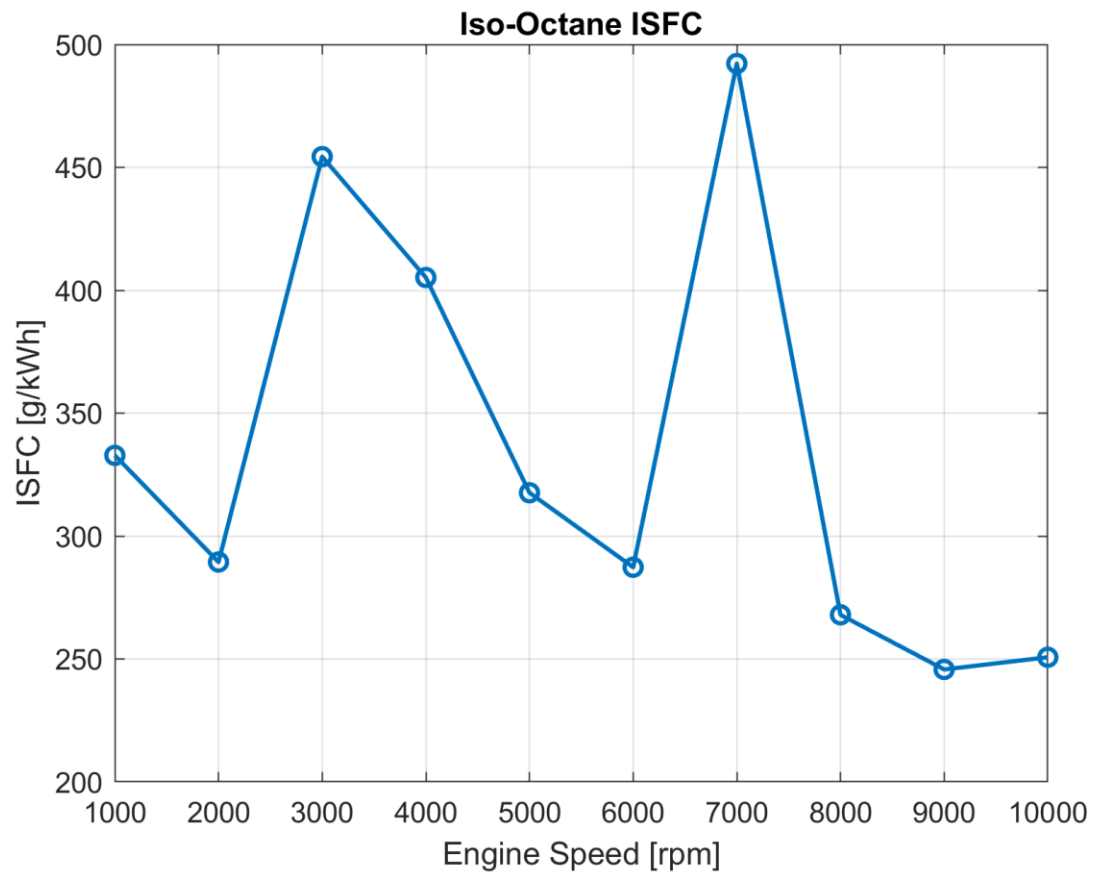


Figure 33: Iso-Octane Forte indicated specific fuel consumption across engine speed

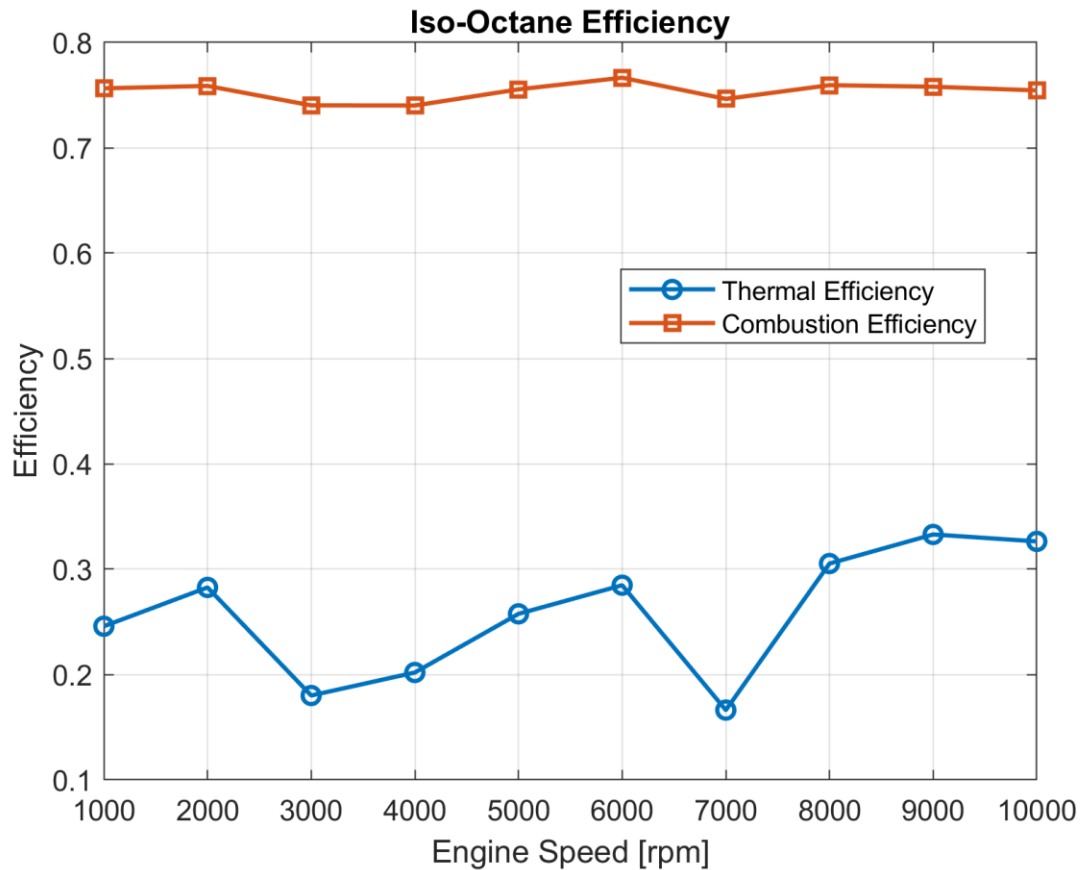


Figure 34: Iso-Octane Forte thermal and combustion efficiency across engine speed

Table 16: Iso-Octane Forte fuel consumption and efficiency characteristics

RPM	FUEL MASS/CYCLE (MG)	FUEL FLOW (KG/H)	ISFC (G/KWH)	THERMAL EFFICIENCY (%)
1000	25.445	3.053	332.8	24.58
2000	25.445	6.107	289.4	28.27
3000	25.445	9.160	454.4	18.01
4000	25.445	12.214	405.2	20.19
5000	25.445	15.267	317.6	25.76
6000	25.445	18.321	287.3	28.48
7000	25.445	21.374	492.2	16.62
8000	25.445	24.428	268.0	30.53
9000	25.445	27.481	245.7	33.30
10000	25.445	30.535	250.7	32.64

Fuel mass per cycle remains essentially constant within each model, causing fuel flow to increase with engine speed. WAVE thermal efficiency reaches 35.19% at 5000 rpm before progressively decreasing to 23.52% at 10,000 rpm, while ISFC increases from 198.3 to 256.5 g/kWh over the same range. Forte shows a less uniform efficiency response, falling to

16.62% at 7000 rpm before increasing to 33.30% at 9000 rpm and remaining at 32.64% at 10,000 rpm. Forte ISFC follows the corresponding variation in indicated output, reaching 492.2 g/kWh at 7000 rpm before decreasing to 250.7 g/kWh at 10,000 rpm.

5.1.2 Octane vs Methanol

5.1.2.1 Combustion Behaviour

The WAVE combustion characteristics of Methanol and Iso-Octane are compared in Figures 35 & 36, with the corresponding combustion phasing and heat-release parameters given in Table 17.

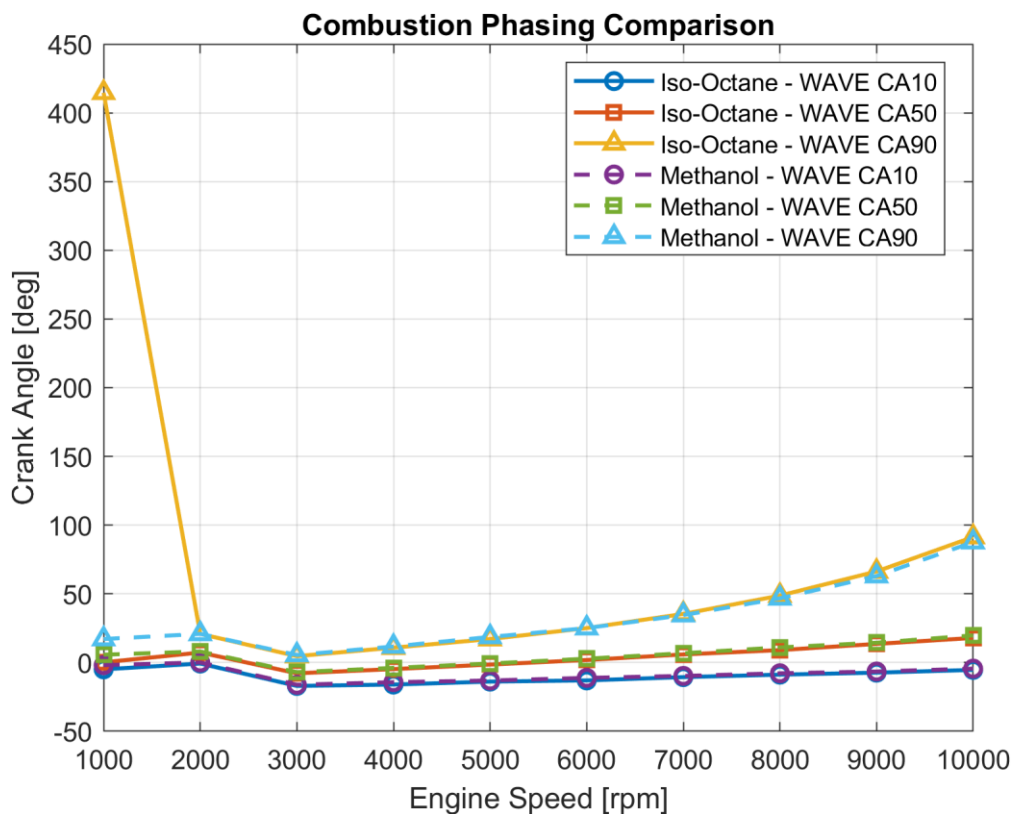


Figure 35: Iso-Octane and Methanol WAVE combustion phasing across engine speed

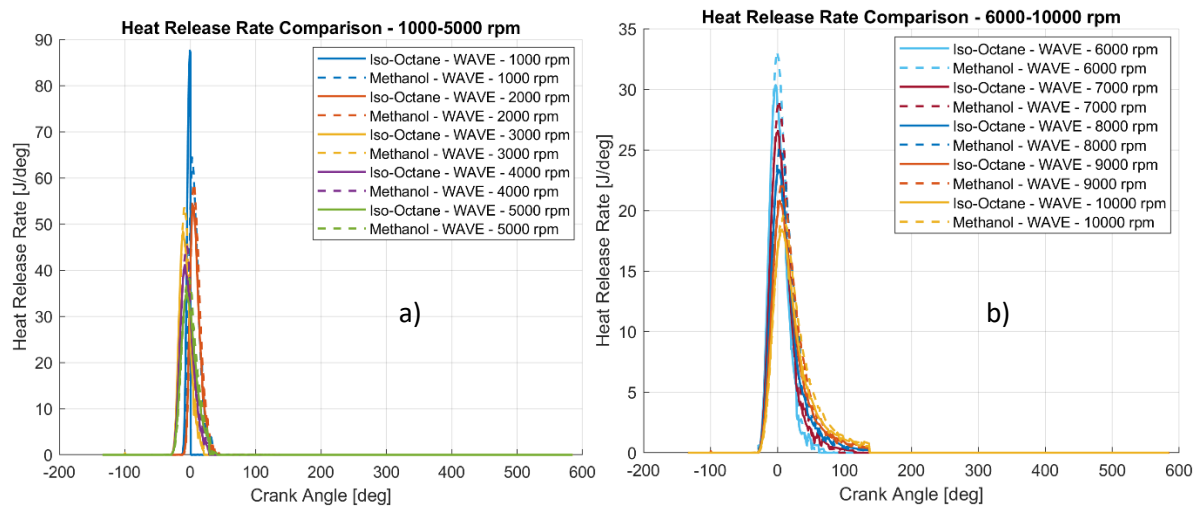


Figure 36: Iso-Octane and Methanol WAVE heat release rate: a) 1000–5000 rpm, b) 6000-10000 rpm

Table 17: Iso-Octane and Methanol WAVE combustion characteristics

RPM	CA50 OCTANE (°ATDC)	CA50 METHANOL (°ATDC)	CA10–90 OCTANE (°CA)	CA10–90 METHANOL (°CA)	PEAK HRR OCTANE (J/°CA)	PEAK HRR METHANOL (J/°CA)
1000	0.19	5.73	419.53	19.02	87.61	64.48
2000	7.30	8.10	21.57	20.77	54.05	58.81
3000	-7.81	-6.98	21.67	21.67	48.40	53.88
4000	-4.73	-3.87	26.84	25.97	41.14	45.15
5000	-1.45	-0.67	30.95	31.72	35.25	38.28
6000	1.88	2.81	38.09	36.23	30.36	33.14
7000	5.92	6.78	45.97	44.24	26.59	28.95
8000	9.08	11.06	57.51	54.54	23.37	25.21
9000	13.60	14.43	73.64	69.45	20.82	22.29
10000	17.93	19.79	96.70	92.06	18.43	19.58

The Forte combustion characteristics of Methanol and Iso-Octane are compared in Figures 37 & 38, with the corresponding combustion phasing and heat-release parameters given in Table 18.

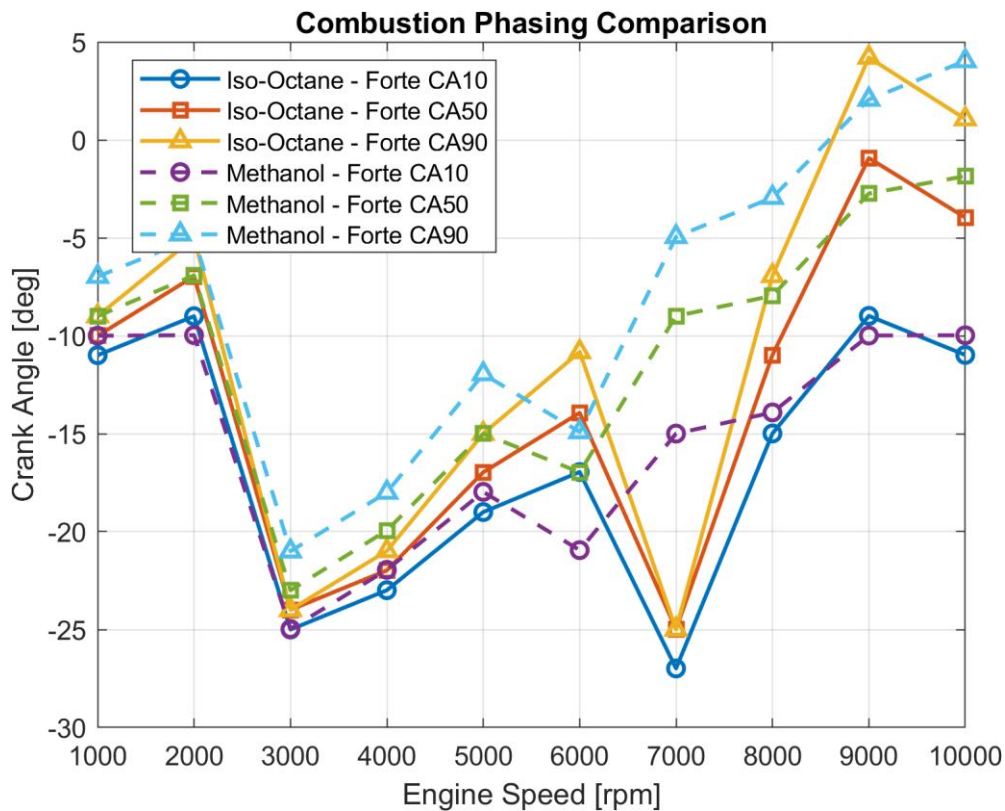


Figure 37: Iso-Octane and Methanol Forte combustion phasing across engine speed

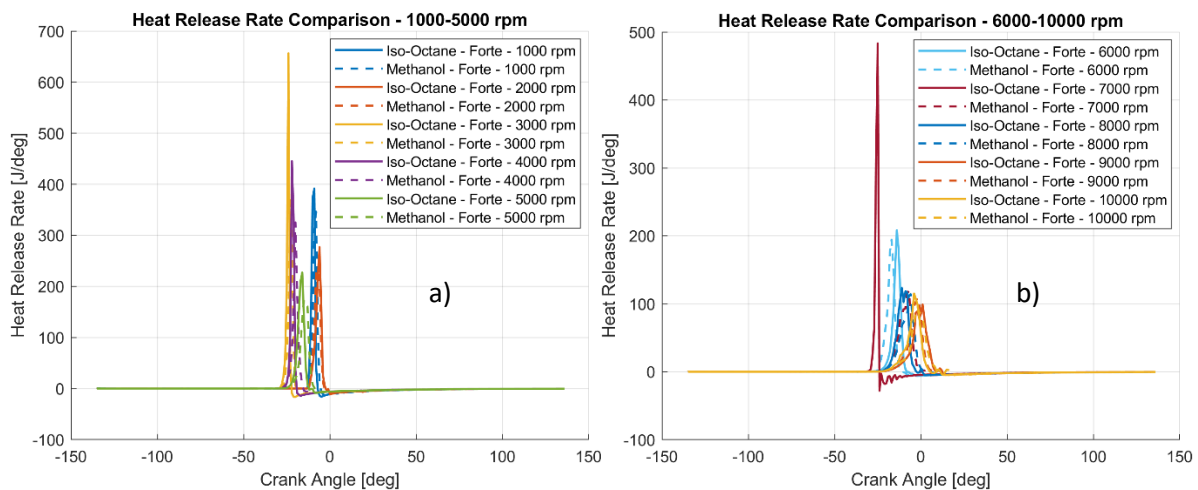


Figure 38: Iso-Octane and Methanol Forte heat release rate: a) 1000–5000 rpm, b) 6000–10000 rpm

Table 18: Iso-Octane and Methanol Forte combustion characteristics

RPM	CA50 OCT. (°ATDC)	CA50 METH. (°ATDC)	CA10–90 OCT. (°CA)	CA10–90 METH. (°CA)	PEAK HRR OCT. (J/°CA)	PEAK HRR METH. (J/°CA)
1000	-10.00	-9.00	2.00	3.02	391.80	346.34
2000	-6.96	-6.91	4.00	4.98	277.12	226.32
3000	-23.99	-23.00	1.01	3.97	656.71	378.21
4000	-21.98	-19.94	2.01	3.95	445.68	325.52
5000	-16.97	-15.00	4.01	6.02	227.50	174.15
6000	-13.93	-16.97	6.13	6.04	208.46	194.52
7000	-24.99	-8.99	2.00	10.05	483.42	119.11
8000	-10.98	-7.96	8.05	10.97	123.42	114.31
9000	-0.92	-2.74	13.19	12.05	99.14	108.68
10000	-3.96	-1.85	12.05	14.01	115.20	95.06

WAVE predicts broadly similar combustion behaviour for methanol and iso-octane at intermediate speeds, with methanol maintaining shorter combustion at high speed. At 10,000 rpm methanol CA50 occurs at 16.18° ATDC compared with 17.93° for iso-octane, while CA10–90 is 92.06°CA compared with 96.70°CA and peak HRR is 19.58 compared with 18.43 J/°CA. Forte produces a more variable fuel response. At 6000 rpm methanol CA50 is –16.97° compared with –13.93° for iso-octane, whereas at 10,000 rpm the respective values are –1.85° and –3.96°. At the latter point, methanol duration reaches 14.01°CA compared with 12.05°CA and peak HRR is 95.06 compared with 115.20 J/°CA.

5.1.2.2 Cylinder Pressure and Temperature

WAVE cylinder pressure and temperature are compared in Figures 39 & 40, with the principal peak values and crank-angle locations summarised in Table 19.

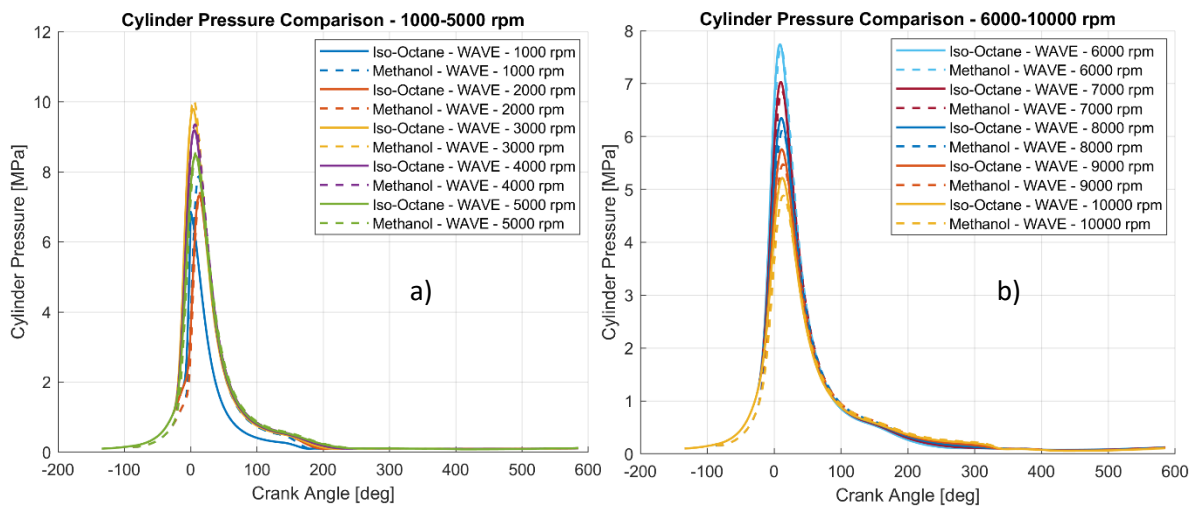


Figure 39: Iso-Octane and Methanol WAVE cylinder pressure: a) 1000–5000 rpm, b) 6000–10000 rpm

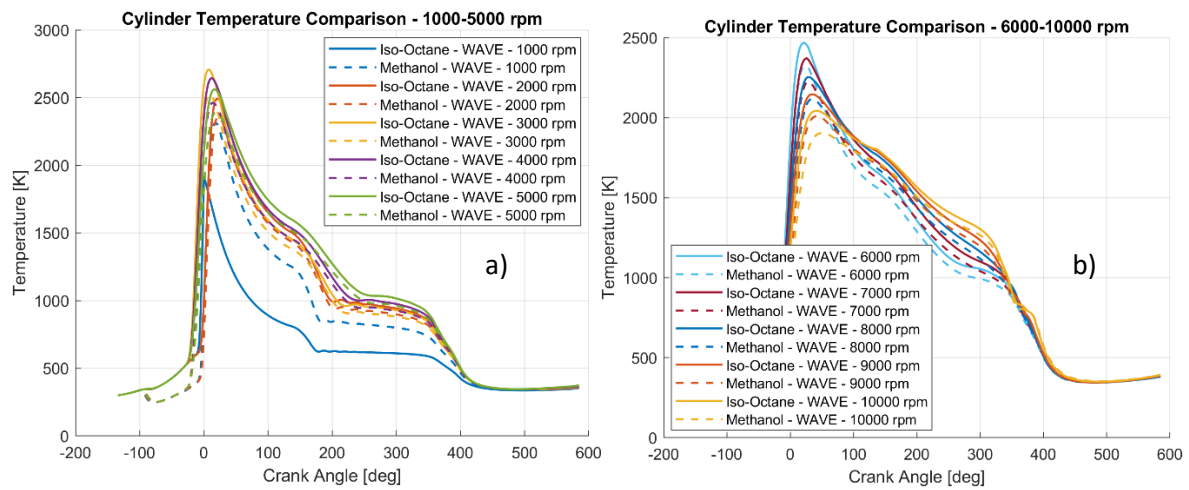


Figure 40: Iso-Octane and Methanol WAVE cylinder temperature: a) 1000–5000 rpm, b) 6000-10000 rpm

Table 19: Iso-Octane and Methanol WAVE peak cylinder pressure and temperature

RPM	P _{MAX} OCT. (MPa)	P _{MAX} METH. (MPa)	P _{MAX} CA OCT./METH. (°ATDC)	T _{MAX} OCT. (K)	T _{MAX} METH. (K)	T _{MAX} CA OCT./METH. (°ATDC)
1000	6.857	7.867	0.19 / 11.82	1916	2315	0.19 / 18.66
2000	7.324	7.401	12.89 / 14.49	2492	2358	20.88 / 22.47
3000	9.791	10.047	3.86 / 4.69	2707	2522	7.19 / 8.85
4000	9.191	9.359	5.65 / 6.52	2643	2462	11.71 / 14.31
5000	8.476	8.571	7.06 / 8.61	2560	2390	16.34 / 18.66
6000	7.743	7.681	8.39 / 10.24	2468	2325	21.39 / 23.25
7000	7.031	6.858	10.25 / 11.99	2370	2227	25.87 / 28.47
8000	6.351	6.124	11.06 / 13.04	2254	2119	28.91 / 34.86
9000	5.760	5.473	11.08 / 13.60	2146	2011	34.52 / 42.05
10000	5.218	4.883	12.34 / 14.20	2043	1903	41.18 / 51.40

Forte cylinder pressure and temperature are compared in Figures 41 & 42, with the principal peak values and crank-angle locations summarised in Table 20.

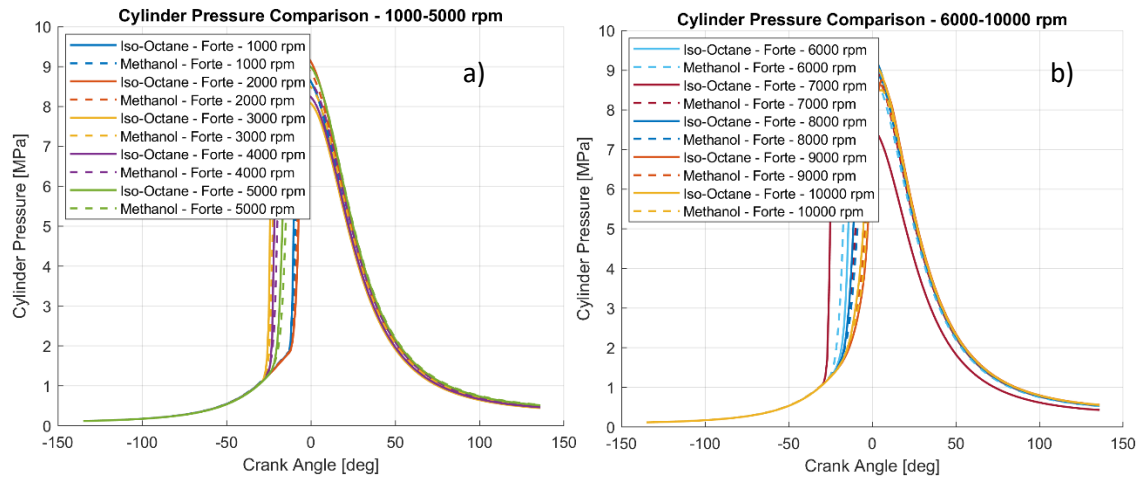


Figure 41: Iso-Octane and Methanol Forte cylinder pressure: a) 1000–5000 rpm, b) 6000-10000 rpm

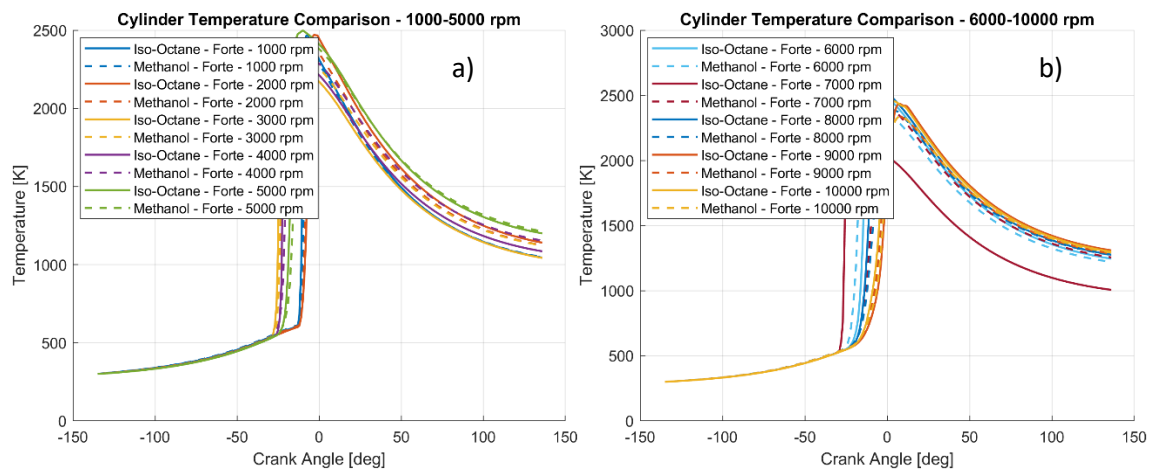


Figure 42: Iso-Octane and Methanol Forte cylinder temperature: a) 1000–5000 rpm, b) 6000-10000 rpm

Table 20: Iso-Octane and Methanol Forte peak cylinder pressure and temperature

RPM	P_{MAX} OCT./METH. (MPa)	P_{MAX} CA OCT./METH. (°ATDC)	T_{MAX} OCT./METH. (K)	T_{MAX} CA OCT./METH. (°ATDC)
1000	8.854 / 8.788	-5.98 / -3.99	2465 / 2387	-7.00 / -5.99
2000	9.189 / 8.909	-0.97 / -0.98	2472 / 2373	-2.98 / -1.96
3000	8.132 / 8.534	-2.96 / -1.95	2389 / 2376	-22.99 / -16.89
4000	8.287 / 8.637	-2.76 / -1.84	2398 / 2380	-19.98 / -12.99
5000	9.031 / 9.002	-1.92 / -1.81	2498 / 2427	-9.94 / -5.99
6000	9.319 / 8.964	-1.99 / -1.92	2526 / 2421	-5.99 / -8.96
7000	7.601 / 9.057	-2.00 / 1.04	2237 / 2410	-24.99 / 2.03
8000	9.351 / 9.009	0.00 / 1.08	2508 / 2396	0.00 / 2.05
9000	8.763 / 8.720	6.18 / 3.03	2434 / 2350	7.04 / 5.06
10000	9.045 / 8.500	3.16 / 5.11	2454 / 2345	4.08 / 8.04

WAVE predicts relatively small pressure and temperature differences between methanol and iso-octane through the lower and intermediate speeds, before methanol becomes lower at high speed. At 10,000 rpm peak pressure reaches 4.88 MPa for methanol compared with 5.22 MPa for iso-octane, while peak temperature is 1903 and 2043 K respectively. Forte also predicts relatively close pressure values through much of the range, including 9.00 MPa for methanol and 9.03 MPa for iso-octane at 5000 rpm. At 10,000 rpm methanol reaches 8.50 MPa compared with 9.04 MPa, while peak temperature is 2345 K compared with 2454 K. Forte also shows a pronounced local change at 7000 rpm, where methanol peak pressure reaches 9.06 MPa while iso-octane falls to 7.60 MPa..

5.1.2.3 Engine performance

The WAVE resulting single-cylinder indicated power is compared in Figure 43, with IMEP, indicated work per cycle and power summarised in Table 21.

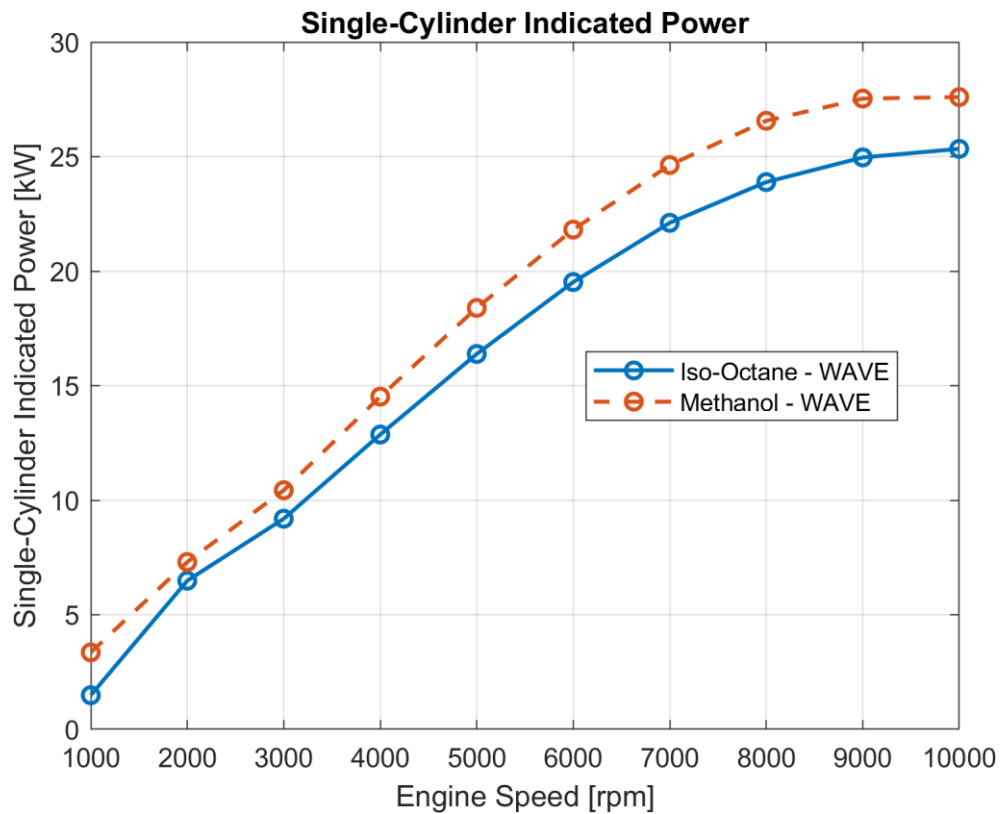


Figure 43: Iso-Octane and Methanol WAVE single-cylinder indicated power across engine speed

Table 21: Iso-Octane and Methanol WAVE single-cylinder indicated performance

RPM	IMEP OCT./METH. (MPA)	WORK/CYCLE OCT./METH. (KJ)	INDICATED POWER OCT./METH. (KW)
1000	0.633 / 1.429	0.0446 / 0.1007	1.487 / 3.358
2000	1.381 / 1.554	0.0973 / 0.1095	6.488 / 7.303
3000	1.304 / 1.481	0.0919 / 0.1044	9.193 / 10.437
4000	1.369 / 1.545	0.0965 / 0.1089	12.867 / 14.523
5000	1.395 / 1.566	0.0983 / 0.1104	16.385 / 18.396
6000	1.385 / 1.548	0.0976 / 0.1091	19.522 / 21.815
7000	1.344 / 1.498	0.0948 / 0.1056	22.110 / 24.632
8000	1.271 / 1.413	0.0896 / 0.0996	23.888 / 26.565
9000	1.181 / 1.302	0.0832 / 0.0918	24.964 / 27.534
10000	1.079 / 1.175	0.0760 / 0.0828	25.338 / 27.599

The Forte resulting single-cylinder indicated power is compared in Figure 44, with IMEP, indicated work per cycle and power summarised in Table 22.

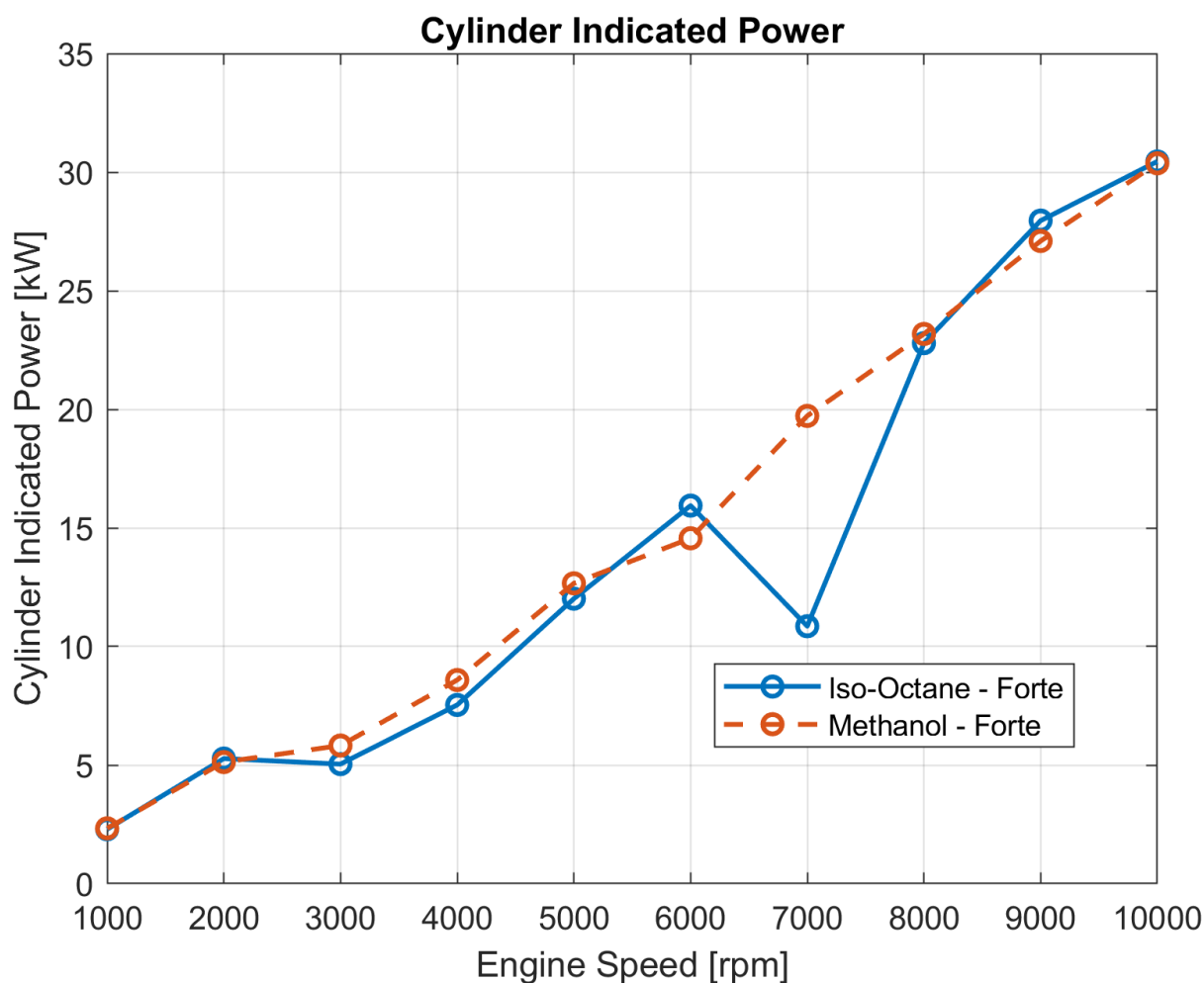


Figure 44: Iso-Octane and Methanol Forte single-cylinder indicated power across engine speed

Table 22: Iso-Octane and Methanol Forte single-cylinder indicated performance

RPM	IMEP OCT./METH. (MPA)	INDICATED WORK OCT./METH. (KJ/CYCLE)	INDICATED POWER OCT./METH. (KW)
1000	0.976 / 0.993	0.2752 / 0.2800	9.17 / 9.33
2000	1.123 / 1.091	0.3165 / 0.3076	21.10 / 20.51
3000	0.715 / 0.826	0.2016 / 0.2330	20.16 / 23.30
4000	0.802 / 0.914	0.2261 / 0.2576	30.14 / 34.35
5000	1.023 / 1.078	0.2884 / 0.3040	48.07 / 50.66
6000	1.131 / 1.033	0.3188 / 0.2913	63.77 / 58.25
7000	0.660 / 1.199	0.1861 / 0.3381	43.42 / 78.89
8000	1.213 / 1.233	0.3419 / 0.3477	91.16 / 92.71
9000	1.322 / 1.282	0.3728 / 0.3613	111.83 / 108.39
10000	1.296 / 1.293	0.3654 / 0.3645	121.80 / 121.49

WAVE predicts greater methanol indicated output through the comparative speed range, with the advantage becoming more apparent at high speed. At 10,000 rpm methanol produces 27.60 kW compared with 25.34 kW for iso-octane, accompanied by greater IMEP and indicated work per cycle. Forte produces a less consistent response: methanol exceeds iso-octane at 3000–5000 rpm and particularly at 7000 rpm, where indicated power reaches 78.89 kW compared with 43.42 kW. The outputs subsequently converge, reaching 121.49 kW for methanol and 121.80 kW for iso-octane at 10,000 rpm, with IMEP values of 1.293 and 1.296 MPa respectively.

5.1.2.4 Fuel Consumption and Efficiency

WAVE fuel consumption and thermal efficiency are compared in Figures 45 and 46, with the corresponding numerical values given in Table 23.

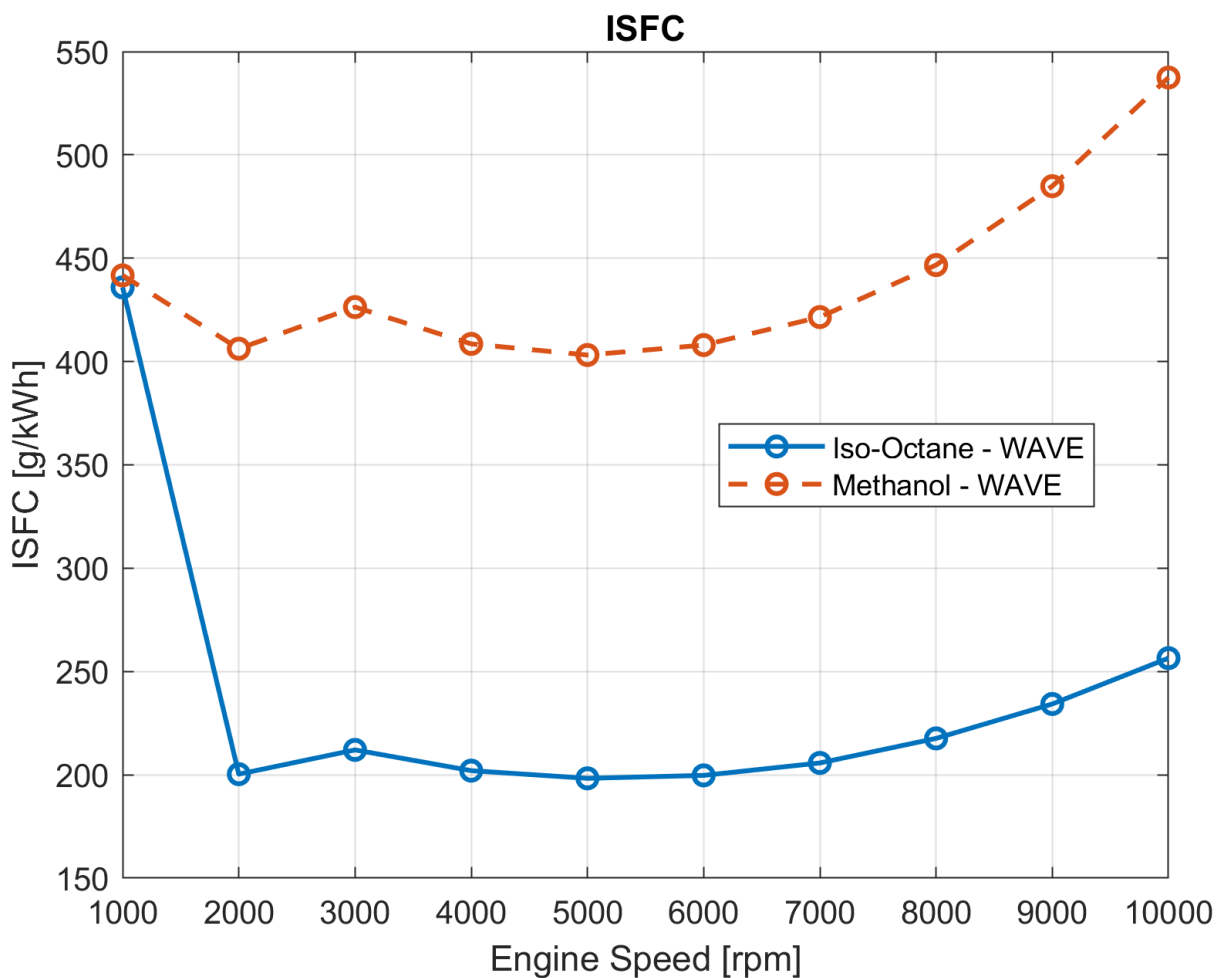


Figure 45: Iso-Octane and Methanol WAVE indicated specific fuel consumption across engine speed

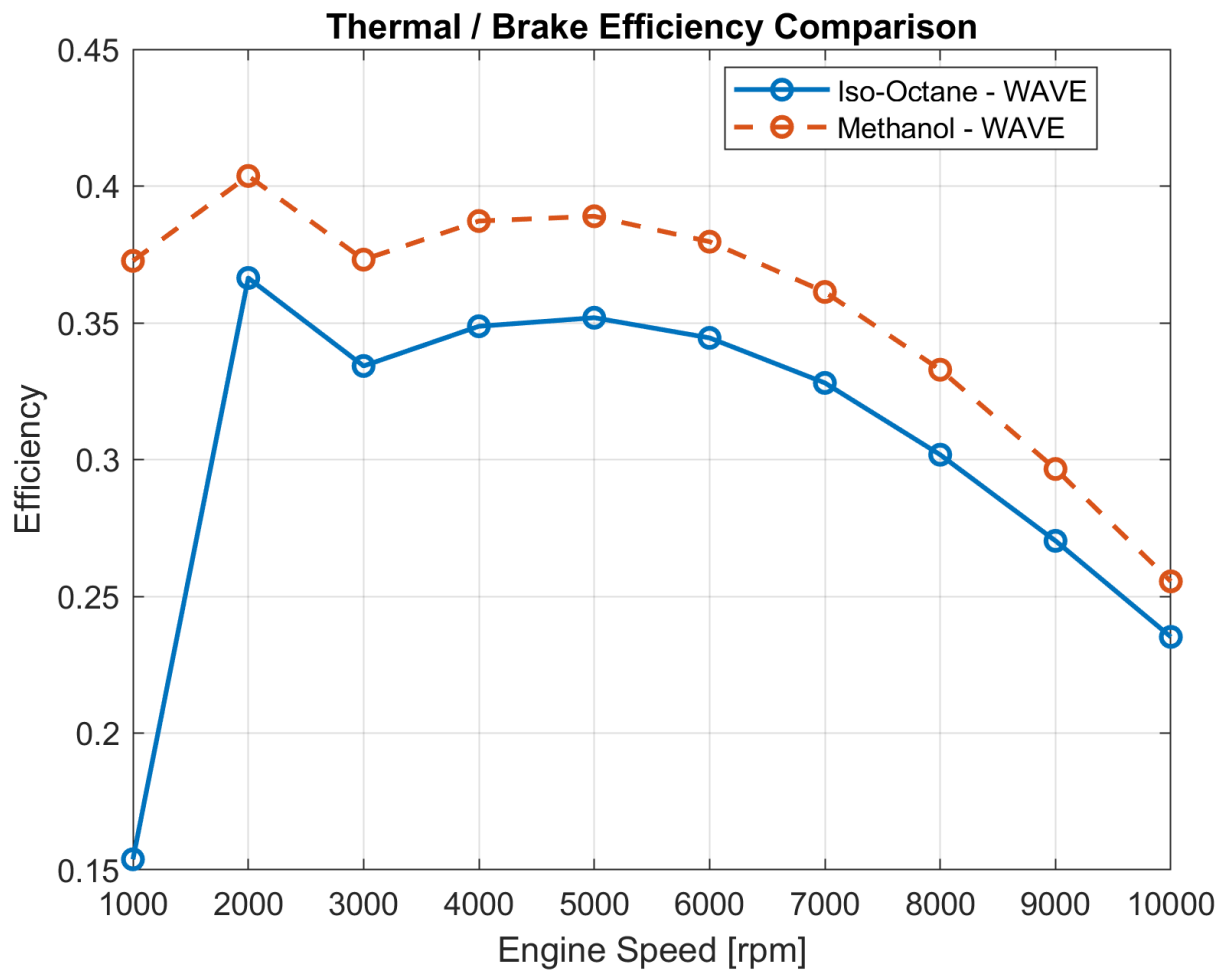


Figure 46: Iso-Octane and Methanol WAVE brake efficiency across engine speed

Table 23: Iso-Octane and Methanol WAVE fuel consumption and efficiency

RPM	FUEL MASS/CYCLE OCT./METH. (MG)	FUEL FLOW OCT./METH. (KG/H)	ISFC OCT./METH. (G/KWH)	BRAKE EFFICIENCY OCT./METH. (%)
1000	5.401 / 12.361	0.648 / 1.483	435.9 / 441.7	15.39 / 37.27
2000	5.415 / 12.361	1.300 / 2.967	200.3 / 406.2	36.64 / 40.37
3000	5.415 / 12.361	1.949 / 4.450	212.1 / 426.4	33.42 / 37.31
4000	5.415 / 12.361	2.599 / 5.933	202.0 / 408.5	34.87 / 38.72
5000	5.415 / 12.361	3.249 / 7.417	198.3 / 403.2	35.19 / 38.90
6000	5.415 / 12.361	3.899 / 8.900	199.7 / 408.0	34.45 / 37.97
7000	5.415 / 12.361	4.549 / 10.383	205.7 / 421.5	32.80 / 36.13
8000	5.415 / 12.361	5.199 / 11.867	217.6 / 446.7	30.18 / 33.28
9000	5.415 / 12.361	5.848 / 13.350	234.3 / 484.9	27.03 / 29.66
10000	5.415 / 12.361	6.498 / 14.833	256.5 / 537.4	23.52 / 25.55

Forte fuel consumption and thermal efficiency are compared in Figures 47 and 48, with the corresponding numerical values given in Table 24.

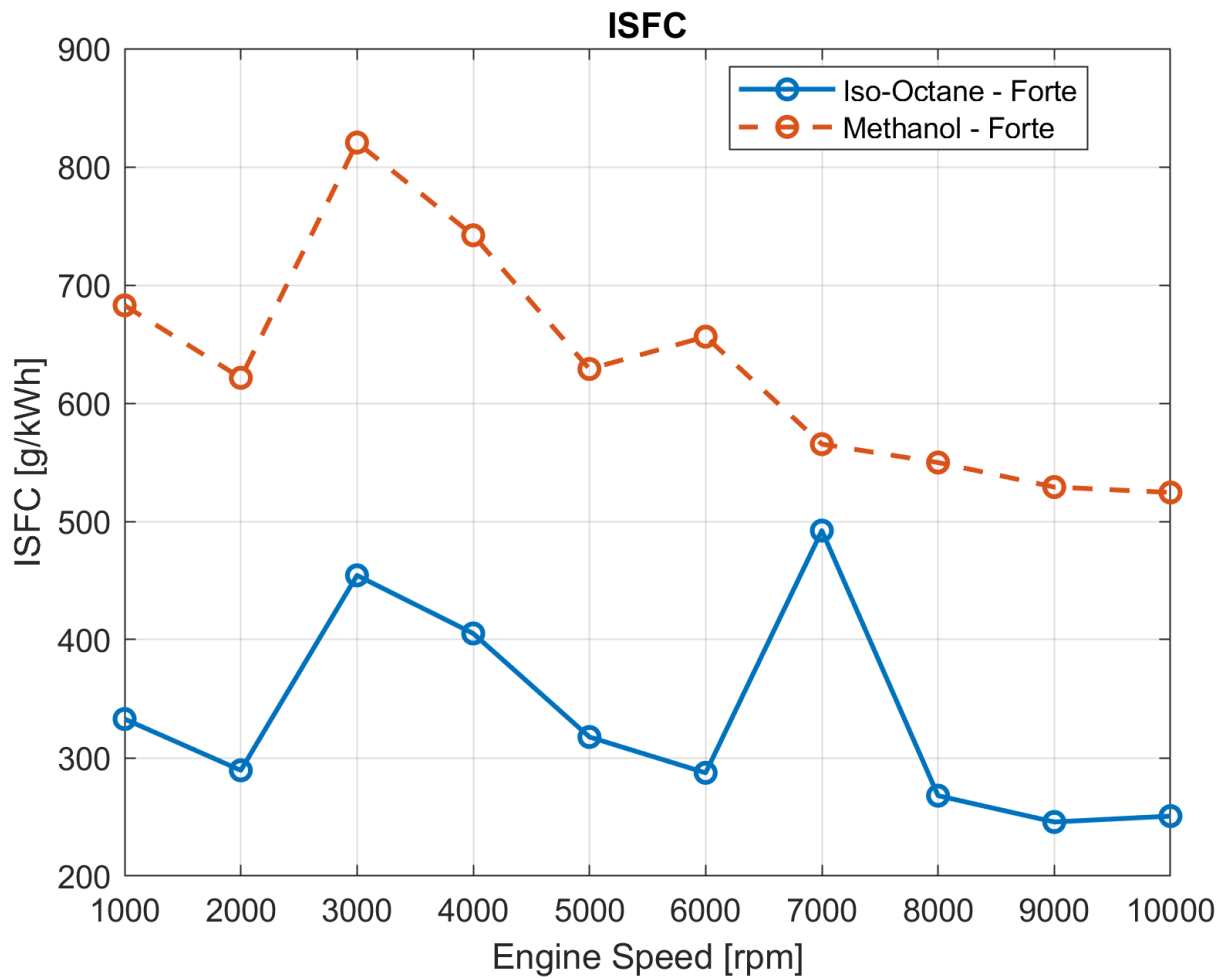


Figure 47: Iso-Octane and Methanol Forte indicated specific fuel consumption across engine speed

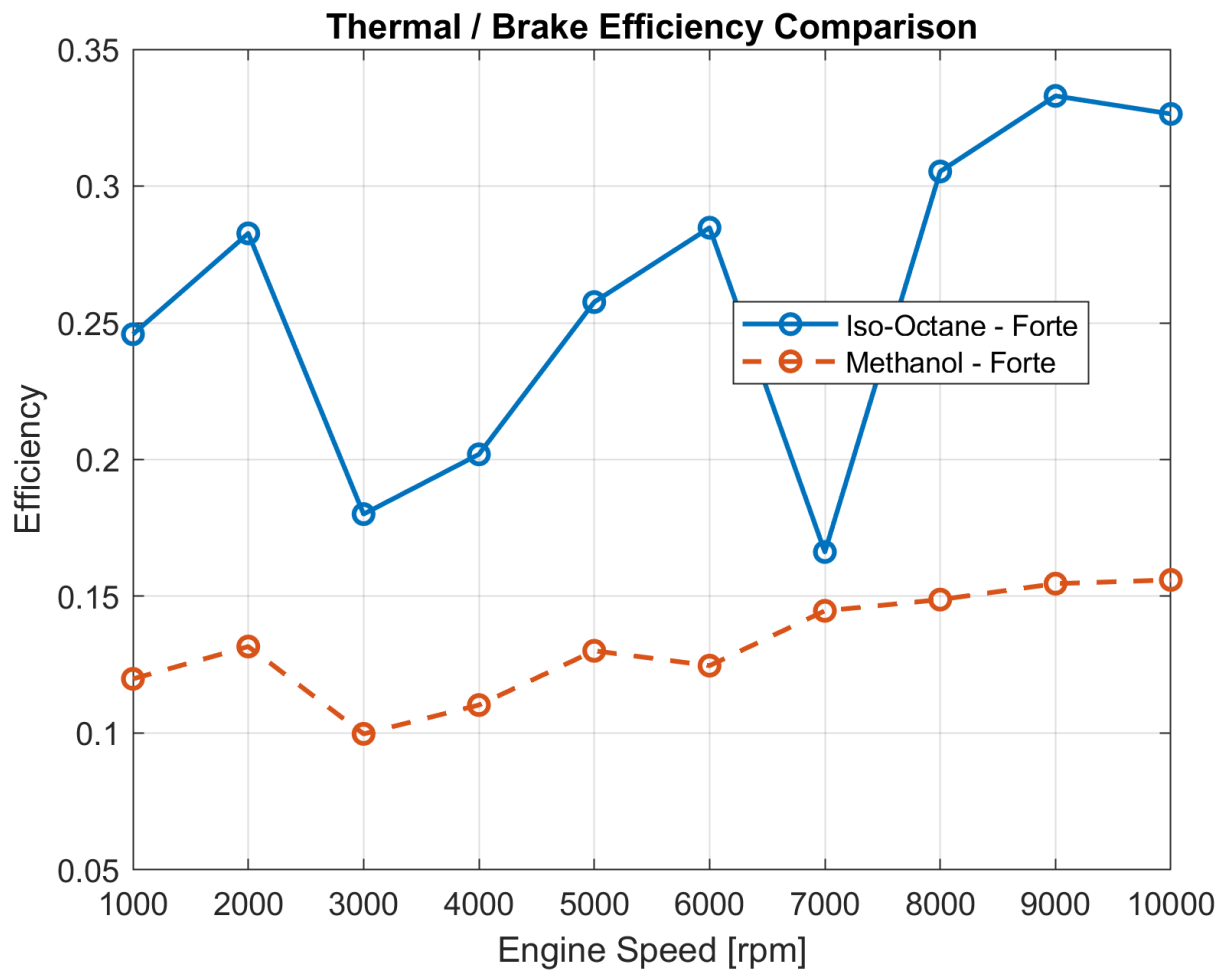


Figure 48: Iso-Octane and Methanol Forte brake efficiency across engine speed

Table 24: Iso-Octane and Methanol Forte fuel consumption and efficiency

RPM	FUEL MASS/CYCLE OCT./METH. (MG)	FUEL FLOW OCT./METH. (KG/H)	ISFC OCT./METH. (G/KWH)	BRAKE EFFICIENCY OCT./METH. (%)
1000	25.445 / 53.100	3.053 / 6.372	332.8 / 682.8	24.58 / 11.98
2000	25.445 / 53.100	6.107 / 12.744	289.4 / 621.5	28.27 / 13.16
3000	25.445 / 53.100	9.160 / 19.116	454.4 / 820.5	18.01 / 9.97
4000	25.445 / 53.100	12.214 / 25.488	405.2 / 742.1	20.19 / 11.03
5000	25.445 / 53.100	15.267 / 31.860	317.6 / 628.9	25.76 / 13.01
6000	25.445 / 53.100	18.321 / 38.232	287.3 / 656.3	28.48 / 12.47
7000	25.445 / 53.100	21.374 / 44.604	492.2 / 565.4	16.62 / 14.47
8000	25.445 / 53.100	24.428 / 50.976	268.0 / 549.8	30.53 / 14.88
9000	25.445 / 53.100	27.481 / 57.348	245.7 / 529.1	33.30 / 15.46
10000	25.445 / 53.100	30.535 / 63.720	250.7 / 524.5	32.64 / 15.60

Methanol requires substantially greater fuel mass in both models because of its lower gravimetric energy content, producing correspondingly greater fuel flow and mass-based ISFC. WAVE nevertheless predicts greater methanol efficiency over the comparative range, reaching 25.55% at 10,000 rpm compared with 23.52% for iso-octane. Forte predicts lower methanol efficiency throughout its 1000–10,000 rpm dataset, beginning at 11.98% compared with 24.58% for iso-octane at 1000 rpm and reaching 15.60% compared with 32.64% at 10,000 rpm. At the latter point, Forte ISFC is 524.5 g/kWh for methanol and 250.7 g/kWh for iso-octane, while the corresponding fuel flows reach 63.72 and 30.53 kg/h.

5.1.3 Octane vs Ethanol

5.1.3.1 Combustion Behaviour

The WAVE combustion behaviour of ethanol and iso-octane is compared in Figures 49 & 50, with the principal combustion parameters summarised in Table 25.

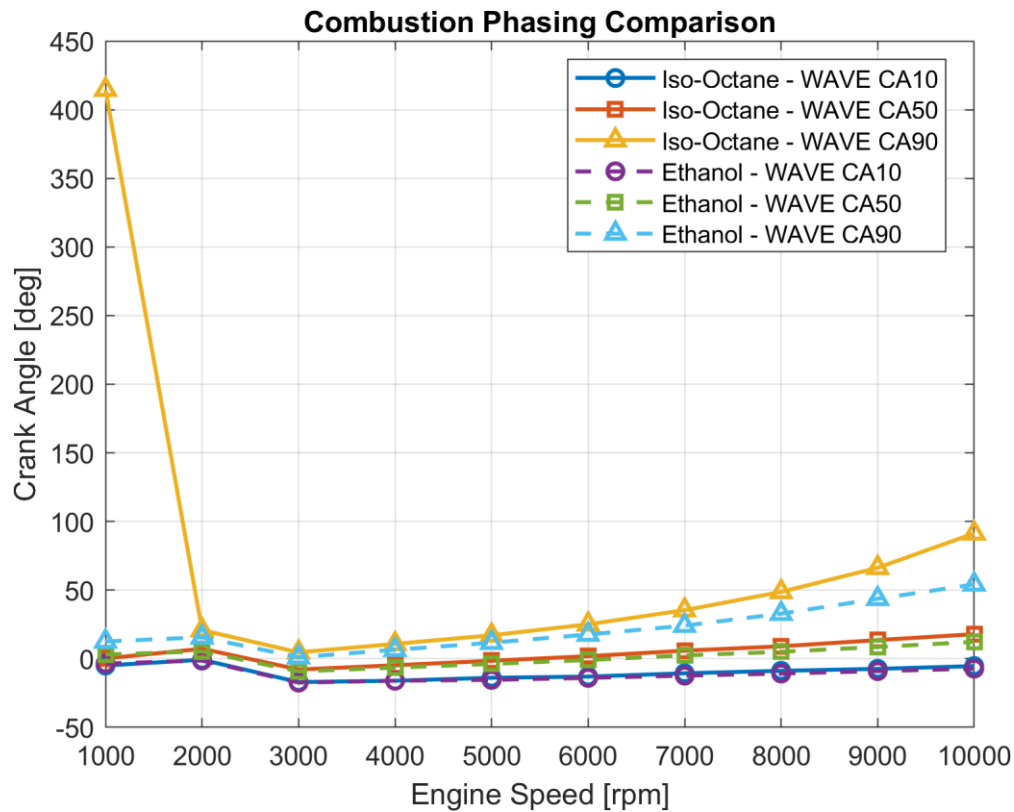


Figure 49: Iso-octane and ethanol WAVE combustion phasing across engine speed

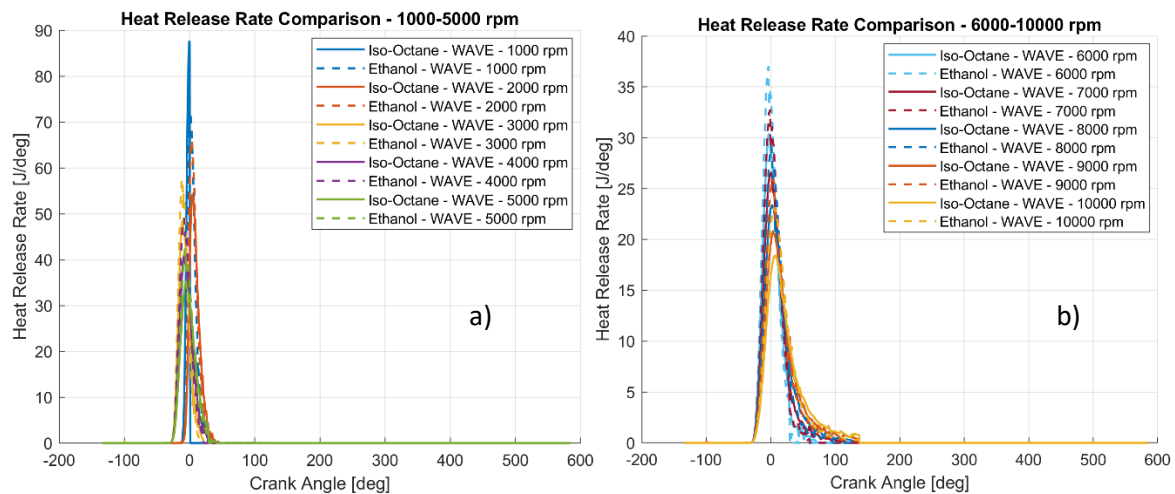


Figure 50: Iso-octane and ethanol WAVE heat release rate: a) 1000–5000 rpm, b) 6000-10000 rpm

Table 25: Iso-octane and ethanol WAVE combustion characteristics

RPM	CA50 OCT. (°ATDC)	CA50 ETH. (°ATDC)	CA10–90 OCT. (°CA)	CA10–90 ETH. (°CA)	PEAK HRR OCT. (J/°CA)	PEAK HRR ETH. (J/°CA)
2000	7.30	5.33	21.57	16.97	54.05	66.07
3000	−7.81	−9.20	21.67	18.52	48.40	57.57
4000	−4.73	−6.47	26.84	22.51	41.14	48.96

5000	-1.45	-3.77	30.95	27.08	35.25	42.31
6000	1.88	-0.90	38.09	31.59	30.36	36.97
7000	5.92	2.45	45.97	36.44	26.59	32.71
8000	9.08	5.11	57.51	43.64	23.37	29.17
9000	13.60	8.57	73.64	52.73	20.82	25.99
10000	17.93	12.34	96.70	61.39	18.43	23.13

The Forte combustion behaviour of ethanol and iso-octane is compared in Figures 51 & 52, with the principal combustion parameters summarised in Table 26.

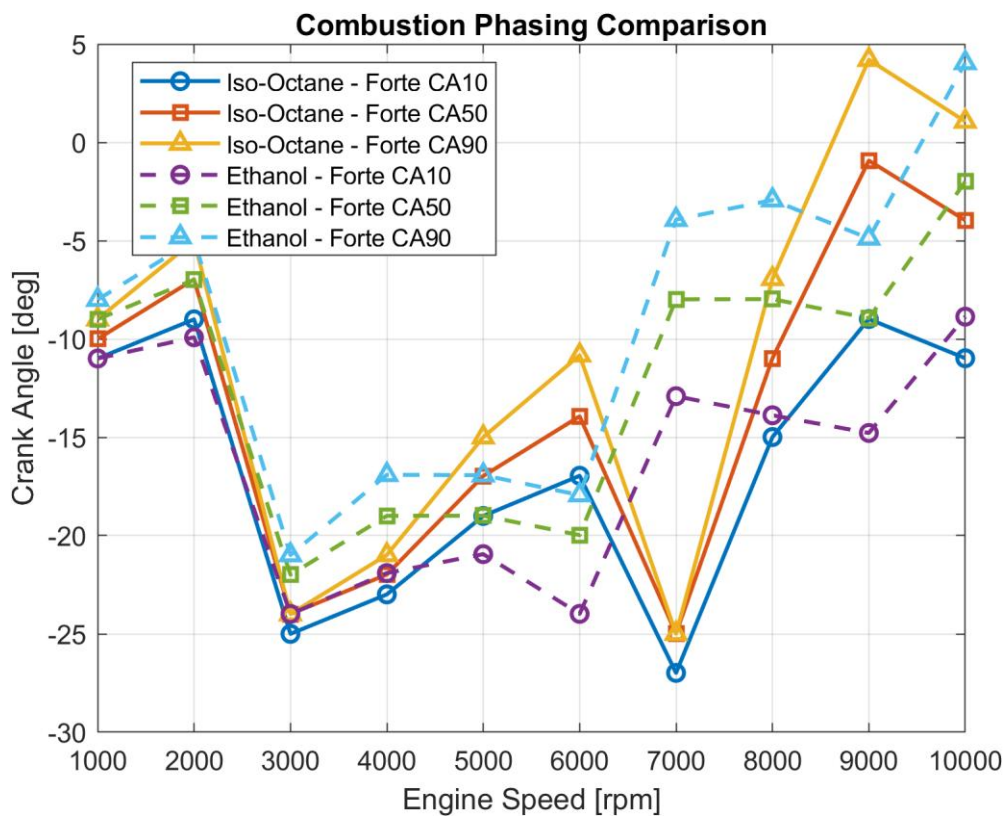


Figure 51: Iso-octane and ethanol Forte combustion phasing across engine speed

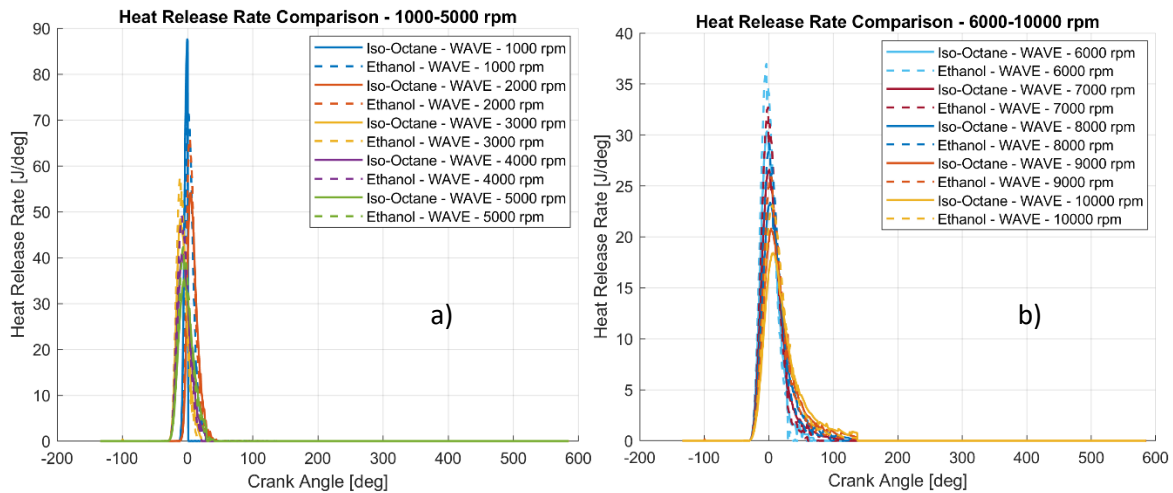


Figure 52: Iso-octane and ethanol Forte heat release rate: a) 1000–5000 rpm, b) 6000–10000 rpm

Table 26: Iso-octane and ethanol Forte combustion characteristics

RPM	CA50 OCT. (°ATDC)	CA50 ETH. (°ATDC)	CA10–90 OCT. (°CA)	CA10–90 ETH. (°CA)	PEAK HRR OCT. (J/°CA)	PEAK HRR ETH. (J/°CA)
1000	-10.00	-8.99	2.00	2.99	391.80	342.08
2000	-6.96	-6.98	4.00	4.96	277.12	227.98
3000	-23.99	-21.99	1.01	3.00	656.71	381.41
4000	-21.98	-19.00	2.01	5.00	445.68	280.60
5000	-16.97	-18.99	4.01	4.02	227.50	252.65
6000	-13.93	-19.98	6.13	6.04	208.46	230.30
7000	-24.99	-7.98	2.00	8.98	483.42	141.46
8000	-10.98	-7.97	8.05	10.95	123.42	103.84
9000	-0.92	-8.95	13.19	9.90	99.14	119.20
10000	-3.96	-1.99	12.05	12.91	115.20	91.83

WAVE predicts consistently earlier and shorter ethanol combustion than iso-octane, with the separation increasing at high speed. At 10,000 rpm ethanol CA50 occurs at 12.34° ATDC compared with 17.93° for iso-octane, while CA10–90 decreases from 96.70 to 61.39°CA and peak HRR increases from 18.43 to 23.13 J/°CA. Forte produces a more variable response. At 6000 rpm ethanol CA50 is –19.98° compared with –13.93° for iso-octane and the corresponding durations are 6.04 and 6.13°CA. By 10,000 rpm ethanol CA50 is –1.99° compared with –3.96°, CA10–90 is 12.91 compared with 12.05°CA, and peak HRR is 91.83 compared with 115.20 J/°CA.

5.1.3.2 Cylinder Pressure and Temperature

The differences in WAVE combustion behaviour are reflected in the cylinder pressure and temperature histories shown in Figures 53, 54, & 55.

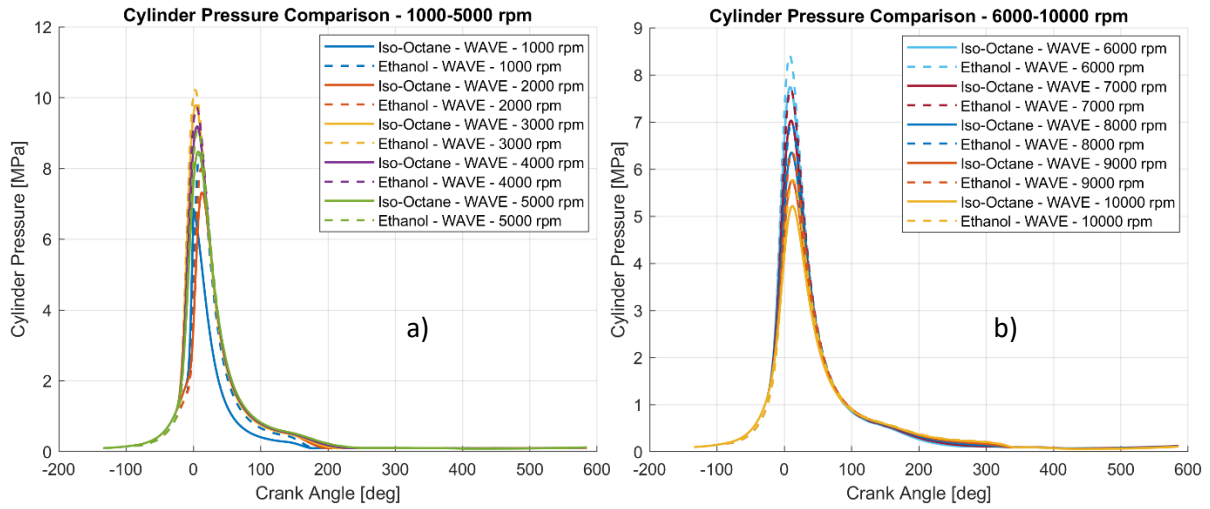


Figure 53: Iso-octane and ethanol WAVE cylinder pressure: a) 1000–5000 rpm, b) 6000-10000 rpm

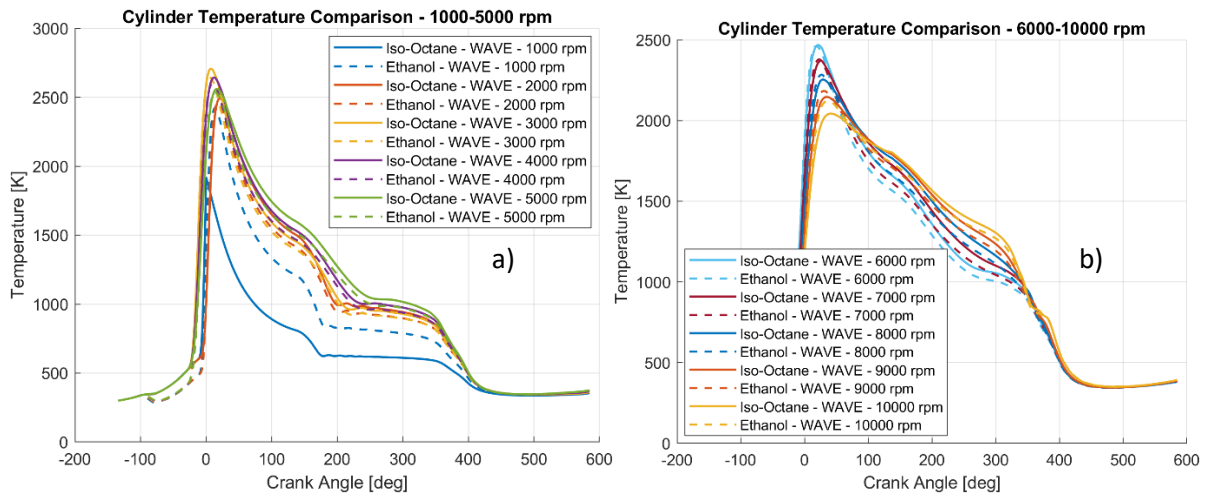


Figure 54: Iso-octane and ethanol WAVE cylinder temperature: a) 1000–5000 rpm, b) 6000-10000 rpm

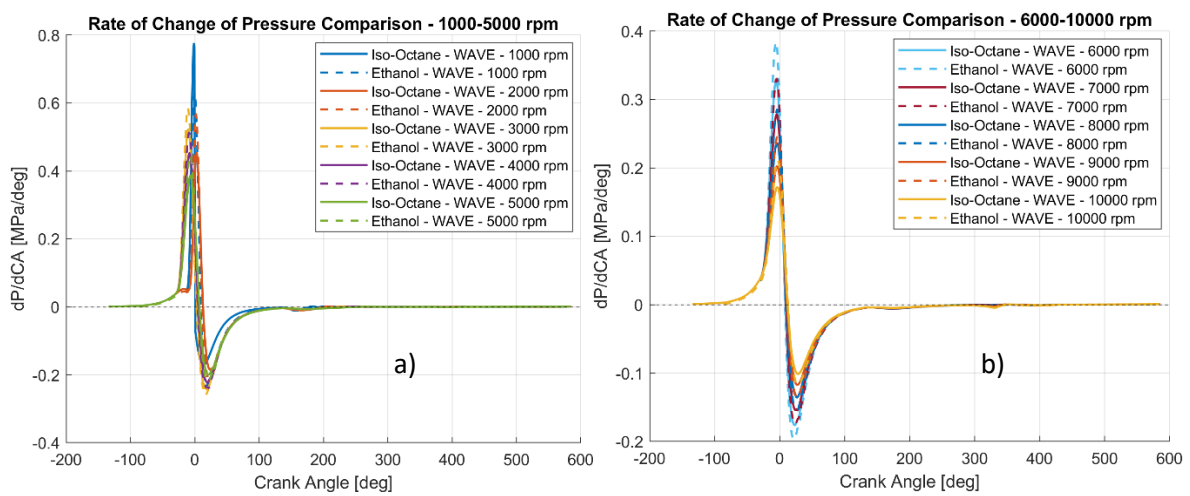


Figure 55: Iso-octane and ethanol WAVE rate of change of cylinder pressure: a) 1000–5000 rpm, b) 6000-10000 rpm

Table 27: Iso-octane and ethanol WAVE cylinder pressure and temperature characteristics

RPM	P _{MAX} OCT. (MPa)	P _{MAX} ETH. (MPa)	MAX. DP/DCA OCT. (MPa/°CA)	MAX. DP/DCA ETH. (MPa/°CA)	T _{MAX} OCT. (K)	T _{MAX} ETH. (K)
2000	7.324	7.998	0.453	0.568	2492	2460
3000	9.791	10.217	0.522	0.586	2707	2644
4000	9.191	9.730	0.453	0.513	2643	2599
5000	8.476	9.090	0.387	0.446	2560	2534
6000	7.743	8.404	0.329	0.384	2468	2461
7000	7.031	7.693	0.279	0.331	2370	2379
8000	6.351	6.993	0.236	0.286	2254	2285
9000	5.760	6.356	0.202	0.245	2146	2183
10000	5.218	5.766	0.172	0.211	2043	2121

The differences in Forte combustion behaviour are reflected in the cylinder pressure and temperature histories shown in Figures 56–58.

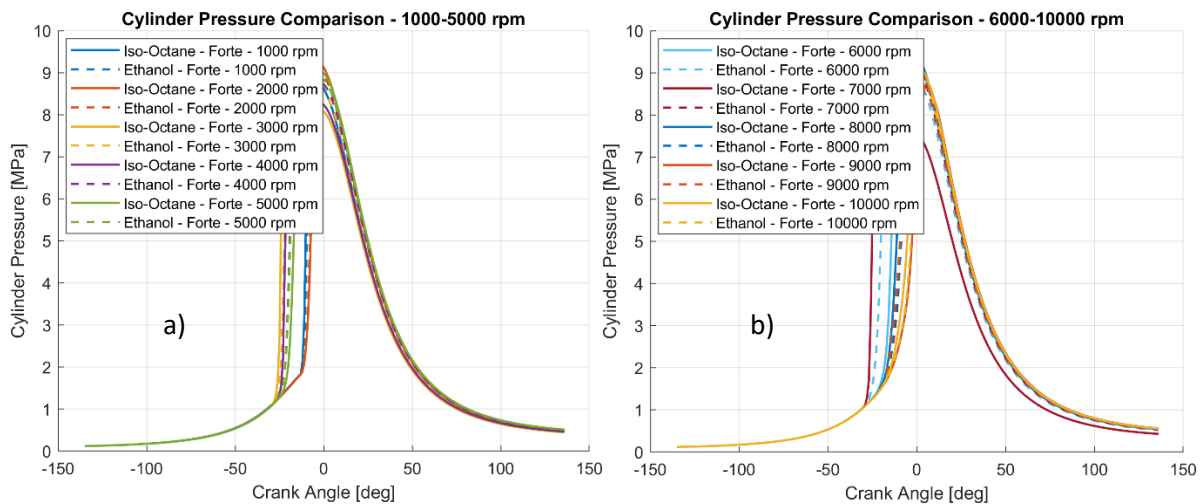


Figure 56: Iso-octane and ethanol Forte cylinder pressure: a) 1000–5000 rpm, b) 6000-10000 rpm

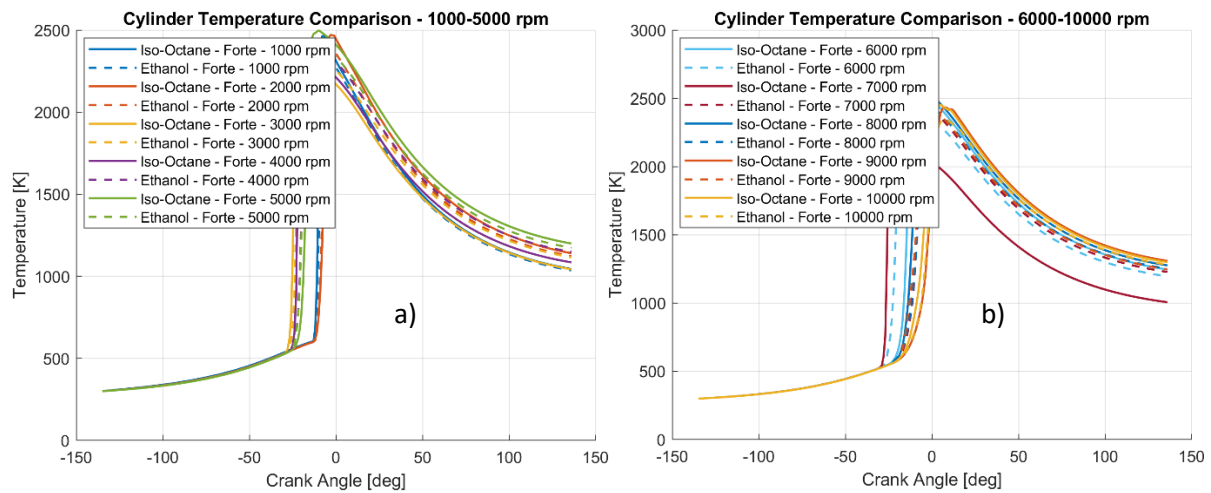


Figure 57: Iso-octane and ethanol Forte cylinder temperature: a) 1000–5000 rpm, b) 6000-10000

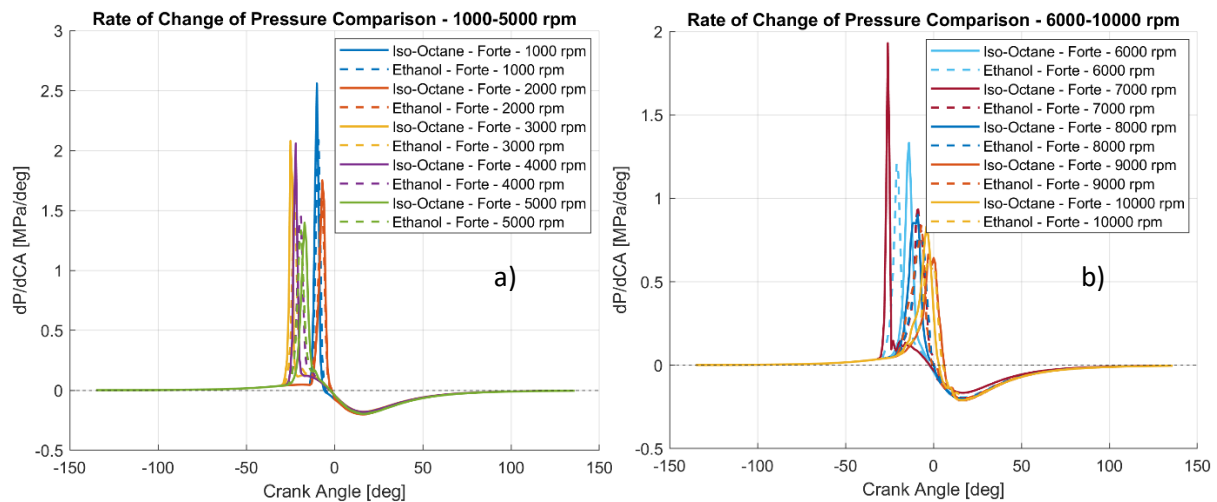


Figure 58: Iso-octane and ethanol Forte rate of change of cylinder pressure: a) 1000–5000 rpm, b) 6000-10000

Table 28: Iso-octane and ethanol Forte cylinder pressure and temperature characteristics

RPM	P _{MAX} OCT./ETH. (MPa)	MAX RATE OF CHANGE OF PRESSURE OCT./ETH. (MPa/°CA)	T _{MAX} OCT./ETH. (K)
1000	8.854 / 8.795	2.560 / 2.086	2465 / 2389
2000	9.189 / 8.974	1.752 / 1.573	2472 / 2387
3000	8.132 / 8.573	2.081 / 1.676	2389 / 2382
4000	8.287 / 8.769	2.060 / 1.449	2398 / 2399
5000	9.031 / 8.874	1.398 / 1.334	2498 / 2410
6000	9.319 / 8.903	1.333 / 1.208	2526 / 2408
7000	7.601 / 9.148	1.930 / 0.959	2237 / 2417
8000	9.351 / 9.074	0.901 / 0.755	2508 / 2402
9000	8.763 / 9.075	0.664 / 0.836	2434 / 2401
10000	9.045 / 8.609	0.833 / 0.634	2454 / 2350

WAVE predicts increasingly greater ethanol pressure and rate of change of pressure at high speed. At 10,000 rpm ethanol reaches 5.77 MPa compared with 5.22 MPa for iso-octane, while maximum rate of change of pressure is 0.211 compared with 0.172 MPa/°CA and peak temperature is 2121 compared with 2043 K. Forte shows a less consistent fuel response. Ethanol peak pressure exceeds iso-octane at 3000–4000 and 7000–9000 rpm, including 9.15 MPa compared with 7.60 MPa at 7000 rpm, but at 10,000 rpm reaches 8.61 MPa compared with 9.04 MPa. Maximum rate of change of pressure is then 0.634 compared with 0.833 MPa/°CA, with peak temperatures of 2350 and 2454 K respectively.

5.1.3.3 Engine performance

The effect of these combustion differences on engine output is shown by the WAVE indicated-power comparison in Figure 59 and the performance parameters in Table 29.

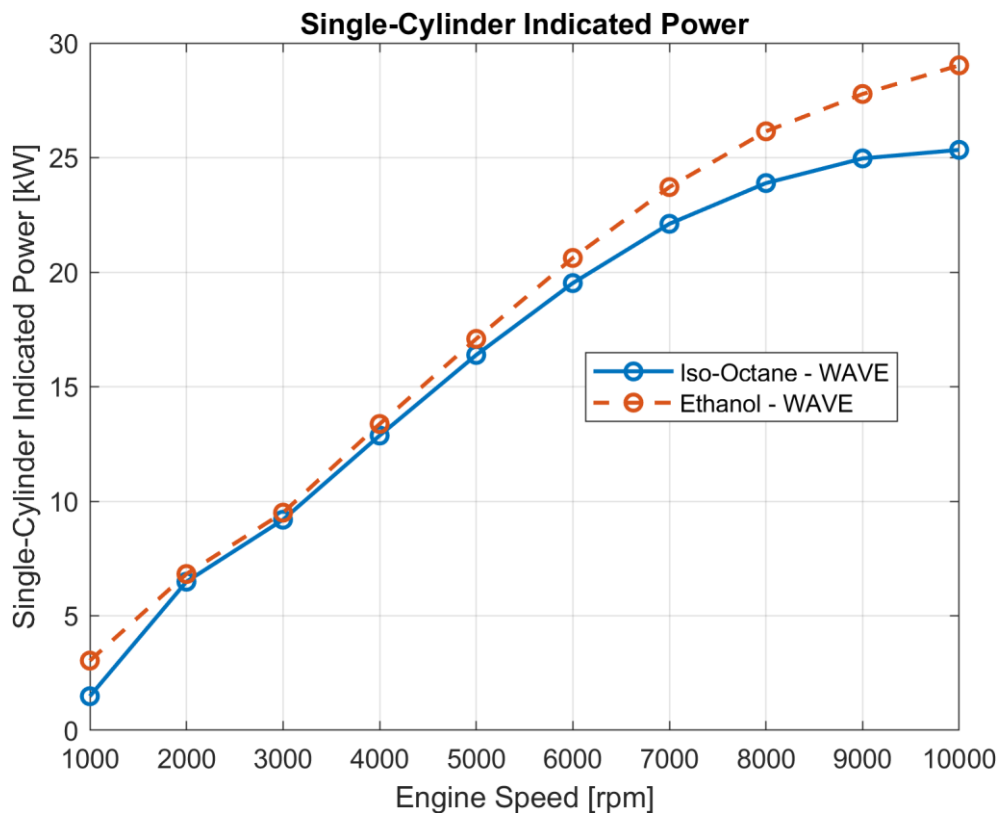


Figure 59: Iso-octane and ethanol WAVE single-cylinder indicated power across engine speed

Table 29: Iso-octane and ethanol WAVE indicated performance

RPM	IMEP OCT. (MPA)	IMEP ETH. (MPA)	WORK OCT. (KJ/CYCLE)	WORK ETH. (KJ/CYCLE)	POWER OCT. (KW)	POWER ETH. (KW)
2000	1.381	1.452	0.0973	0.1023	6.49	6.82
3000	1.304	1.348	0.0919	0.0950	9.19	9.50
4000	1.369	1.423	0.0965	0.1003	12.87	13.37
5000	1.395	1.455	0.0983	0.1025	16.38	17.09
6000	1.385	1.463	0.0976	0.1031	19.52	20.62
7000	1.344	1.442	0.0948	0.1016	22.11	23.71
8000	1.271	1.391	0.0896	0.0981	23.89	26.15
9000	1.181	1.314	0.0832	0.0926	24.96	27.78
10000	1.079	1.236	0.0760	0.0871	25.34	29.03

The effect of these combustion differences on engine output is shown by the Forte indicated-power comparison in Figure 60 and the performance parameters in Table 30.

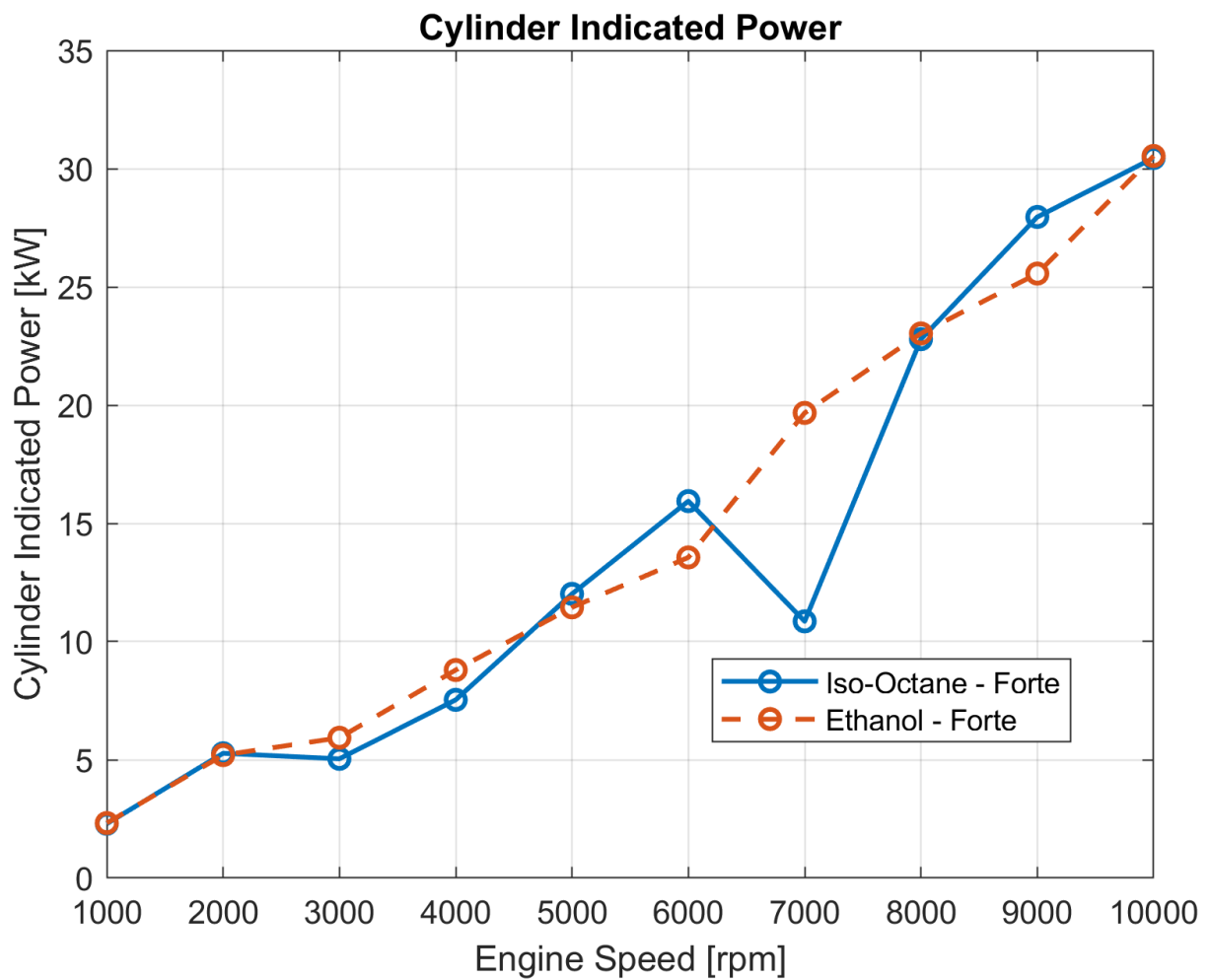


Figure 60: Iso-octane and ethanol Forte single-cylinder indicated power across engine speed

Table 30: Iso-octane and ethanol Forte indicated performance

RPM	IMEP OCT./ETH. (MPa)	INDICATED WORK OCT./ETH. (KJ/CYCLE)	INDICATED POWER OCT./ETH. (KW)
1000	0.976 / 0.990	0.2752 / 0.2790	9.17 / 9.30
2000	1.123 / 1.106	0.3165 / 0.3119	21.10 / 20.80
3000	0.715 / 0.842	0.2016 / 0.2373	20.16 / 23.73
4000	0.802 / 0.936	0.2261 / 0.2639	30.14 / 35.19
5000	1.023 / 0.975	0.2884 / 0.2748	48.07 / 45.79
6000	1.131 / 0.962	0.3188 / 0.2712	63.77 / 54.24
7000	0.660 / 1.196	0.1861 / 0.3372	43.42 / 78.68
8000	1.213 / 1.225	0.3419 / 0.3455	91.16 / 92.13
9000	1.322 / 1.209	0.3728 / 0.3409	111.83 / 102.26
10000	1.296 / 1.300	0.3654 / 0.3664	121.80 / 122.15

WAVE predicts greater ethanol indicated work and power throughout the comparative range, with the difference becoming more pronounced at high speed. At 5000 rpm ethanol indicated power is 17.09 kW compared with 16.38 kW for iso-octane, increasing to 29.03 and 25.34 kW respectively at 10,000 rpm. Forte instead shows a strongly speed-dependent response: ethanol produces greater output at 3000–4000 and 7000–8000 rpm but lower output at several other points. By 10,000 rpm the two fuels produce almost identical indicated output, with ethanol reaching 122.15 kW and 1.300 MPa IMEP compared with 121.80 kW and 1.296 MPa for iso-octane.

5.1.3.4 Fuel Consumption and Efficiency

The WAVE efficiency comparison is shown in Figure 61, while Table 31 includes the corresponding fuel consumption and efficiency parameters.

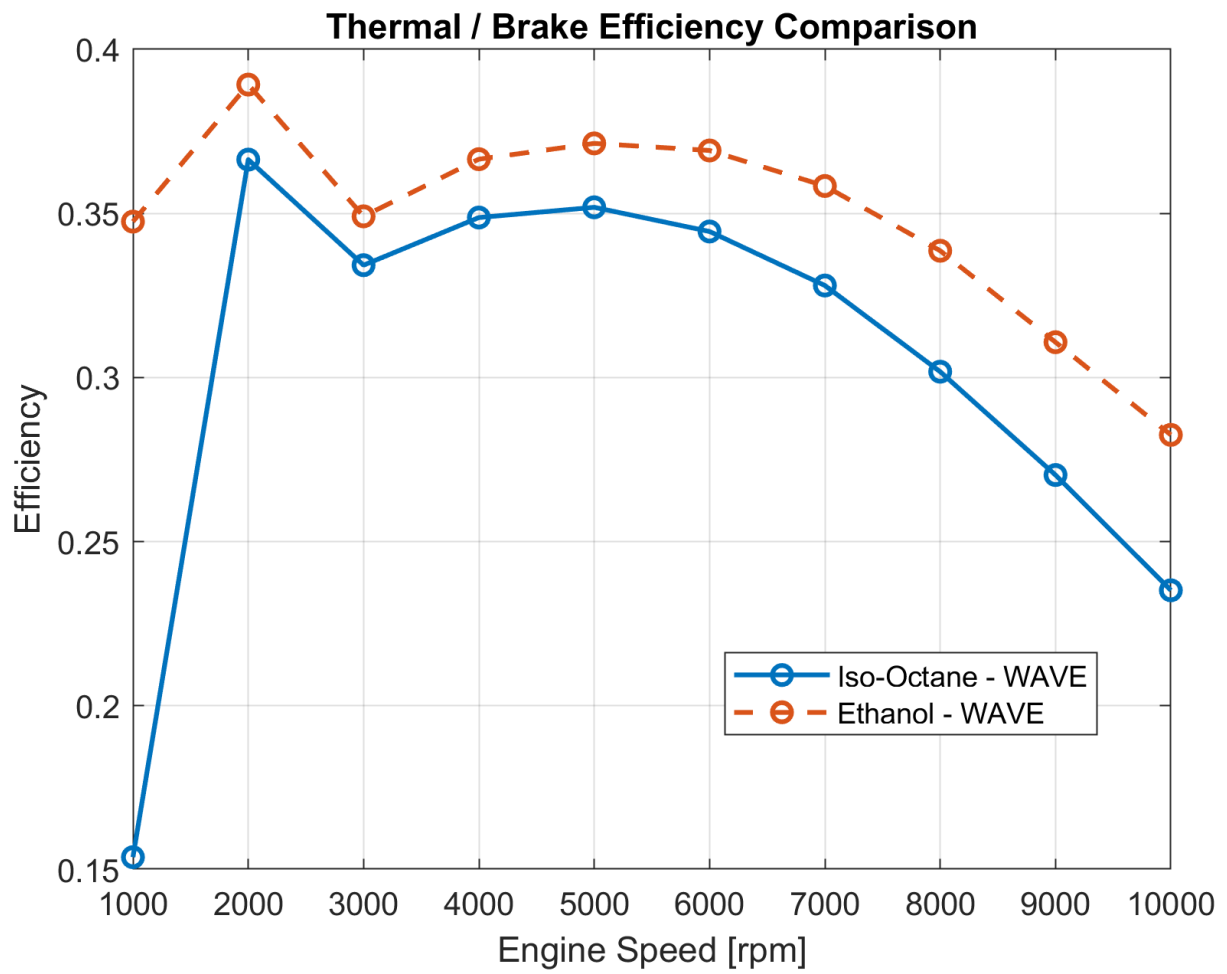


Figure 61: Iso-octane and ethanol WAVE thermal efficiency across engine speed

Table 31: Iso-octane and ethanol WAVE fuel consumption and efficiency

RPM	FUEL/CYCLE OCT. (MG)	FUEL/CYCLE ETH. (MG)	FUEL FLOW OCT. (KG/H)	FUEL FLOW ETH. (KG/H)	ISFC OCT. (G/KWH)	ISFC ETH. (G/KWH)	HTH OCT. (%)	HTH ETH. (%)
2000	5.415	8.894	1.300	2.135	200.3	312.9	36.64	38.92
3000	5.415	8.894	1.949	3.202	212.1	337.0	33.42	34.91
4000	5.415	8.894	2.599	4.269	202.0	319.3	34.87	36.65
5000	5.415	8.894	3.249	5.337	198.3	312.3	35.19	37.13
6000	5.415	8.894	3.899	6.404	199.7	310.6	34.45	36.92
7000	5.415	8.894	4.549	7.471	205.7	315.1	32.80	35.84
8000	5.415	8.894	5.199	8.539	217.6	326.5	30.18	33.86
9000	5.415	8.894	5.848	9.606	234.3	345.8	27.03	31.08
10000	5.415	8.894	6.498	10.673	256.5	367.7	23.52	28.25

The Forte efficiency comparison is shown in Figure 62, while Table 32 includes the corresponding fuel consumption and efficiency parameters.

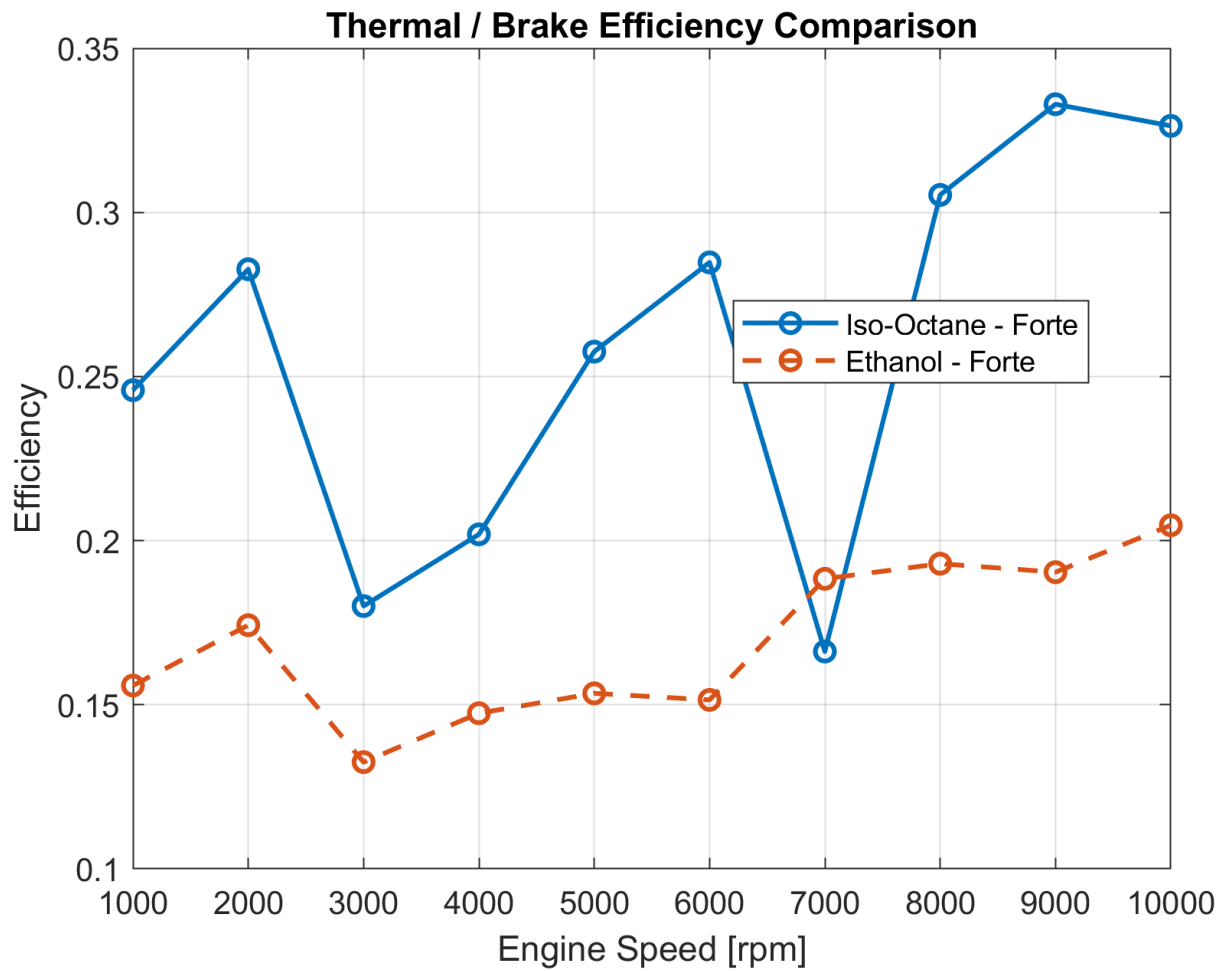


Figure 62: Iso-octane and ethanol Forte thermal efficiency across engine speed

Table 32: Iso-octane and ethanol Forte fuel consumption and efficiency

RPM	FUEL MASS/CYCLE OCT./ETH. (MG)	FUEL FLOW OCT./ETH. (KG/H)	ISFC OCT./ETH. (G/KWH)	BRAKE EFFICIENCY OCT./ETH. (%)
1000	25.445 / 40.687	3.053 / 4.882	332.8 / 525.0	24.58 / 15.59
2000	25.445 / 40.687	6.107 / 9.765	289.4 / 469.6	28.27 / 17.42
3000	25.445 / 40.687	9.160 / 14.647	454.4 / 617.4	18.01 / 13.25
4000	25.445 / 40.687	12.214 / 19.530	405.2 / 555.0	20.19 / 14.74
5000	25.445 / 40.687	15.267 / 24.412	317.6 / 533.1	25.76 / 15.35
6000	25.445 / 40.687	18.321 / 29.295	287.3 / 540.1	28.48 / 15.15
7000	25.445 / 40.687	21.374 / 34.177	492.2 / 434.4	16.62 / 18.84
8000	25.445 / 40.687	24.428 / 39.059	268.0 / 424.0	30.53 / 19.30
9000	25.445 / 40.687	27.481 / 43.942	245.7 / 429.7	33.30 / 19.04
10000	25.445 / 40.687	30.535 / 48.824	250.7 / 399.7	32.64 / 20.47

Ethanol requires greater fuel mass per cycle in both models because of its lower gravimetric energy content, producing correspondingly greater fuel flow and mass-based ISFC. WAVE nevertheless predicts consistently greater ethanol thermal efficiency, reaching 28.25% compared with 23.52% for iso-octane at 10,000 rpm. Forte predicts lower ethanol efficiency across the investigated range, beginning at 15.59% compared with 24.58% at 1000 rpm and reaching 20.47% compared with 32.64% at 10,000 rpm. At the latter point, ethanol ISFC is 399.7 g/kWh compared with 250.7 g/kWh for iso-octane, despite the two fuels producing almost identical indicated output.

5.1.4 Octane vs Hydrogen

5.1.4.1 Combustion Behaviour

The WAVE combustion behaviour of hydrogen and iso-octane is compared in Figures 63–64, with the principal combustion parameters summarised in Table 33.

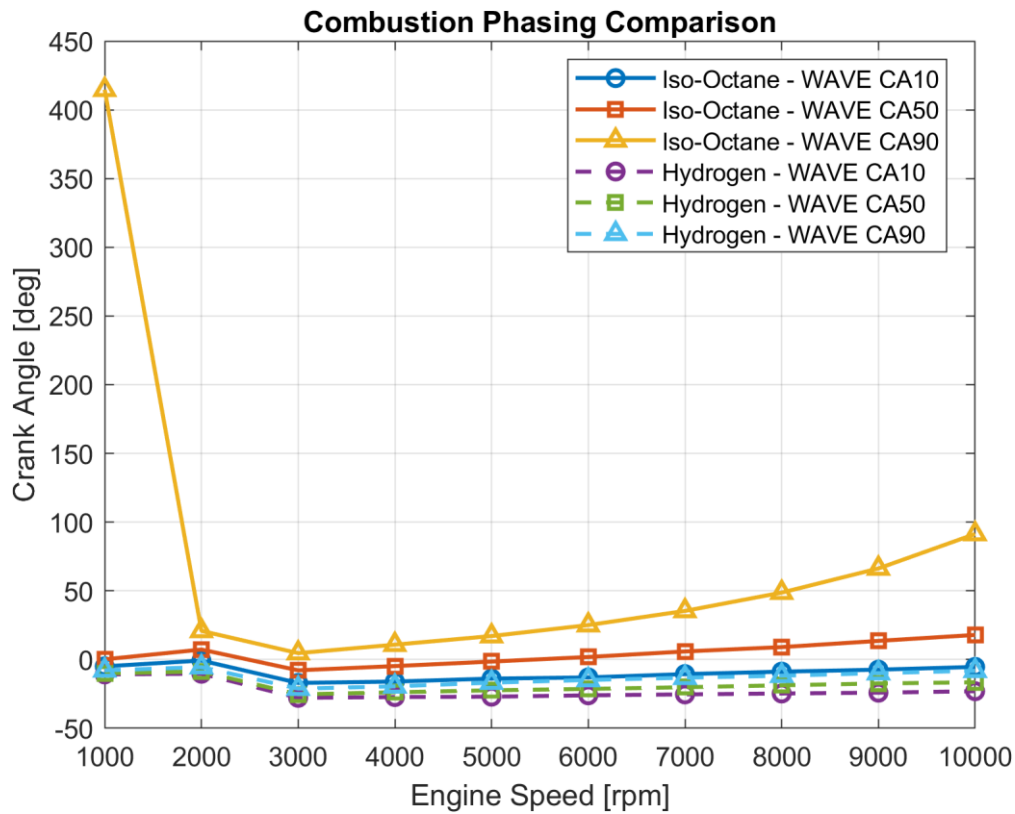


Figure 63: Iso-octane and hydrogen WAVE combustion phasing across engine speed

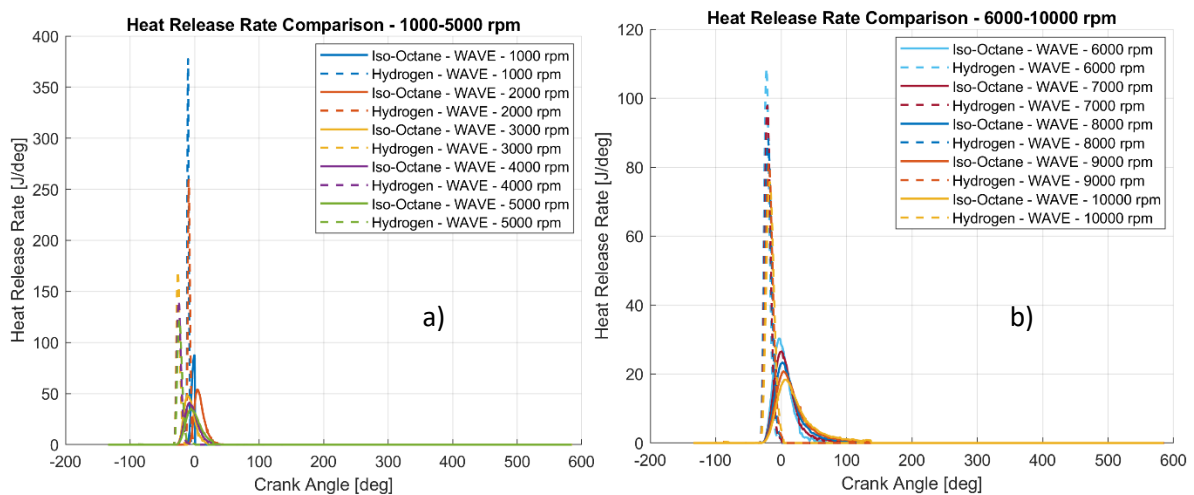


Figure 64: Iso-octane and hydrogen WAVE heat release rate: a) 1000–5000 rpm, b) 6000–10000 rpm

Table 33: Iso-octane and hydrogen WAVE combustion characteristics

RPM	CA50 OCT. (°ATDC)	CA50 H₂ (°ATDC)	CA10–90 OCT. (°CA)	CA10–90 H₂ (°CA)	PEAK HRR OCT. (J/°CA)	PEAK HRR H₂ (J/°CA)
2000	7.30	−8.70	21.57	4.80	54.05	259.40
3000	−7.81	−25.32	21.67	6.67	48.40	170.47
4000	−4.73	−23.79	26.84	7.80	41.14	141.71
5000	−1.45	−22.35	30.95	10.22	35.25	122.71
6000	1.88	−21.35	38.09	11.16	30.36	108.74
7000	5.92	−20.12	45.97	12.15	26.59	97.84
8000	9.08	−18.70	57.51	12.90	23.37	88.21
9000	13.60	−17.38	73.64	14.24	20.82	81.28
10000	17.93	−16.50	96.70	14.89	18.43	75.79

The Forte combustion behaviour of hydrogen and iso-octane is compared in Figures 65–66, with the principal combustion parameters summarised in Table 34.

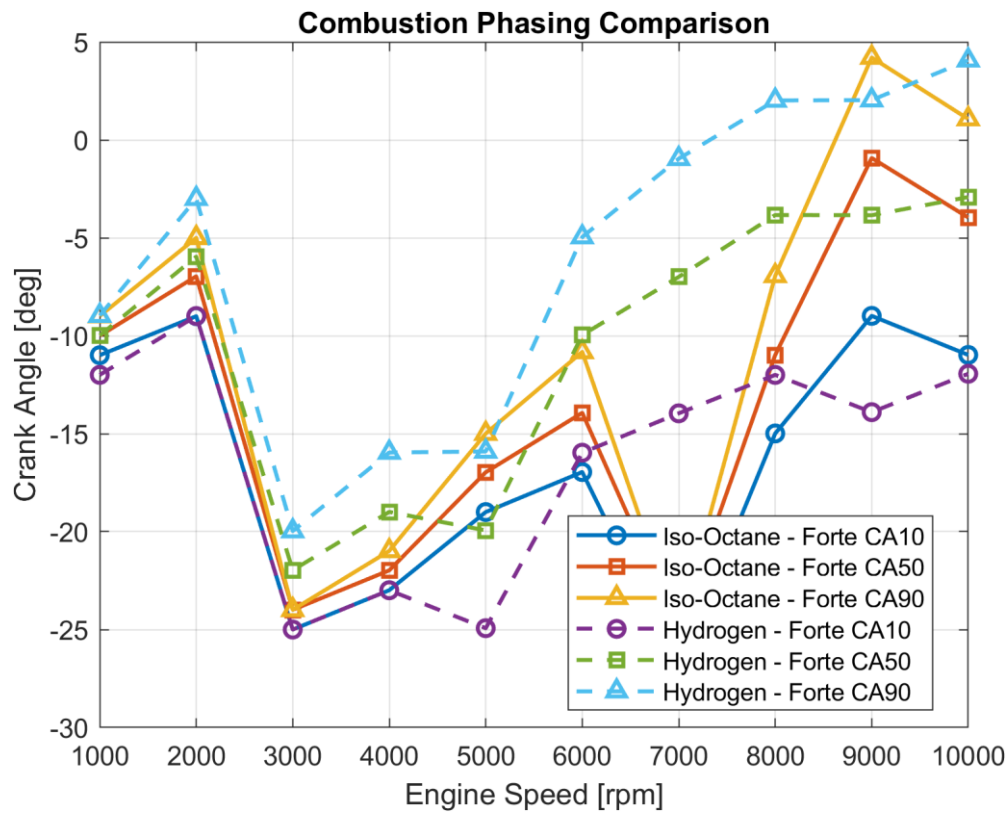


Figure 65: Iso-octane and hydrogen Forte combustion phasing across engine speed

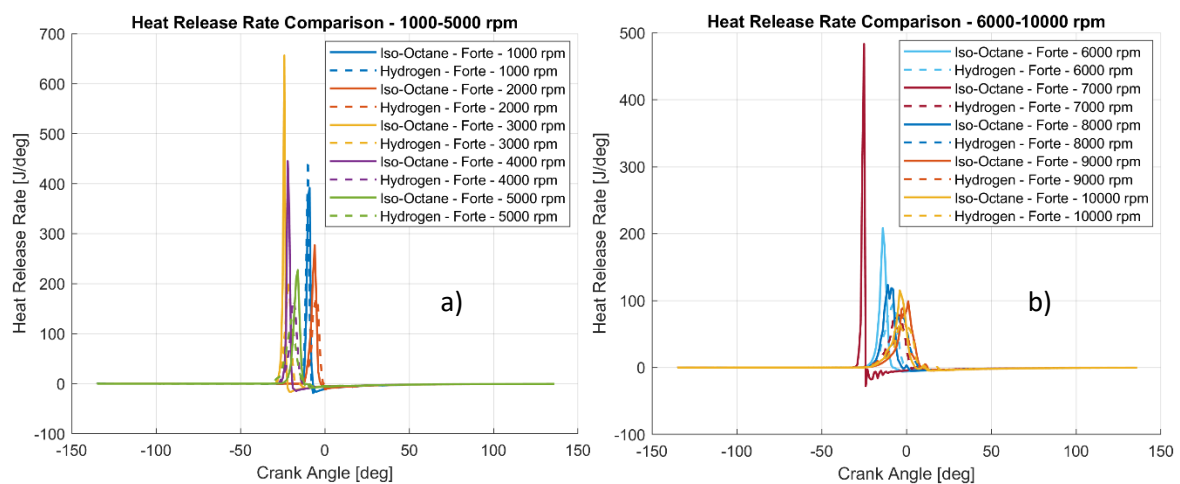


Figure 66: Iso-octane and hydrogen Forte heat release rate: a) 1000–5000 rpm, b) 6000–10000 rpm

Table 34: Iso-octane and hydrogen Forte combustion characteristics

RPM	CA50 OCT. (°ATDC)	CA50 H₂ (°ATDC)	CA10–90 OCT. (°CA)	CA10–90 H₂ (°CA)	PEAK HRR OCT. (J/°CA)	PEAK HRR H₂ (J/°CA)
1000	-10.00	-9.98	2.00	3.03	391.80	446.13
2000	-6.96	-5.96	4.00	5.99	277.12	170.05
3000	-23.99	-21.97	1.01	5.03	656.71	204.10
4000	-21.98	-18.99	2.01	7.03	445.68	166.25
5000	-16.97	-19.94	4.01	9.02	227.50	128.81
6000	-13.93	-9.96	6.13	11.01	208.46	94.76
7000	-24.99	-6.99	2.00	13.00	483.42	78.87
8000	-10.98	-3.83	8.05	14.01	123.42	76.89
9000	-0.92	-3.84	13.19	15.93	99.14	80.15
10000	-3.96	-2.92	12.05	16.01	115.20	69.12

WAVE predicts substantially earlier and shorter hydrogen combustion than iso-octane throughout the comparative range, consistent with hydrogen's much greater laminar burning velocity. At 10,000 rpm hydrogen CA50 occurs at -16.50° ATDC compared with 17.93° for iso-octane, while CA10–90 is 14.89 compared with 96.70°CA and peak HRR is 75.79 compared with $18.43 \text{ J/}^\circ\text{CA}$. Forte produces a different fuel response. At 5000 rpm hydrogen CA50 is -19.94° compared with -16.97° for iso-octane, while at 10,000 rpm the values are -2.92° and -3.96° . Hydrogen duration then reaches 16.01°CA compared with 12.05°CA , with peak HRR of 69.12 compared with $115.20 \text{ J/}^\circ\text{CA}$.

5.1.4.2 Cylinder Pressure and Temperature

The effect of the different combustion rates on cylinder pressure and temperature for WAVE is shown in Figures 67–69, with the principal values summarised in Table 35.

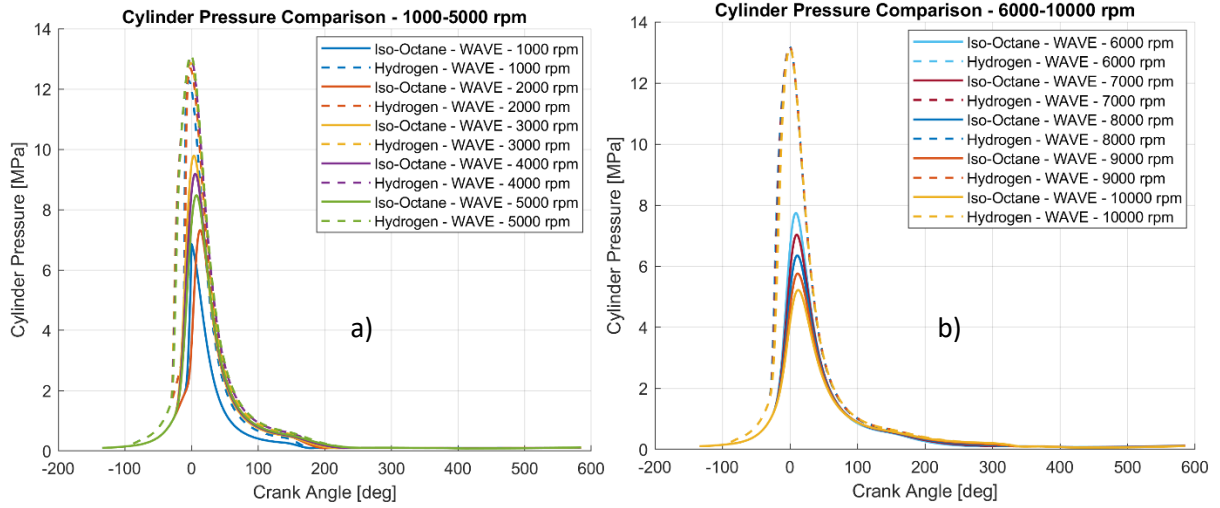


Figure 67: Iso-octane and hydrogen WAVE cylinder pressure: a) 1000–5000 rpm, b) 6000-10000 rpm

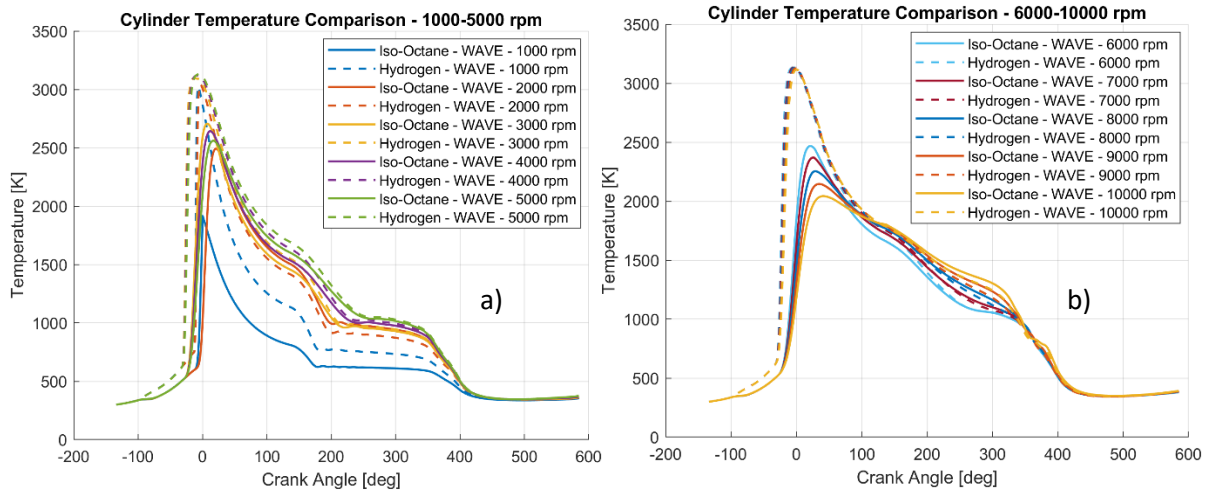


Figure 68: Iso-octane and hydrogen WAVE cylinder temperature: a) 1000–5000 rpm, b) 6000-10000 rpm

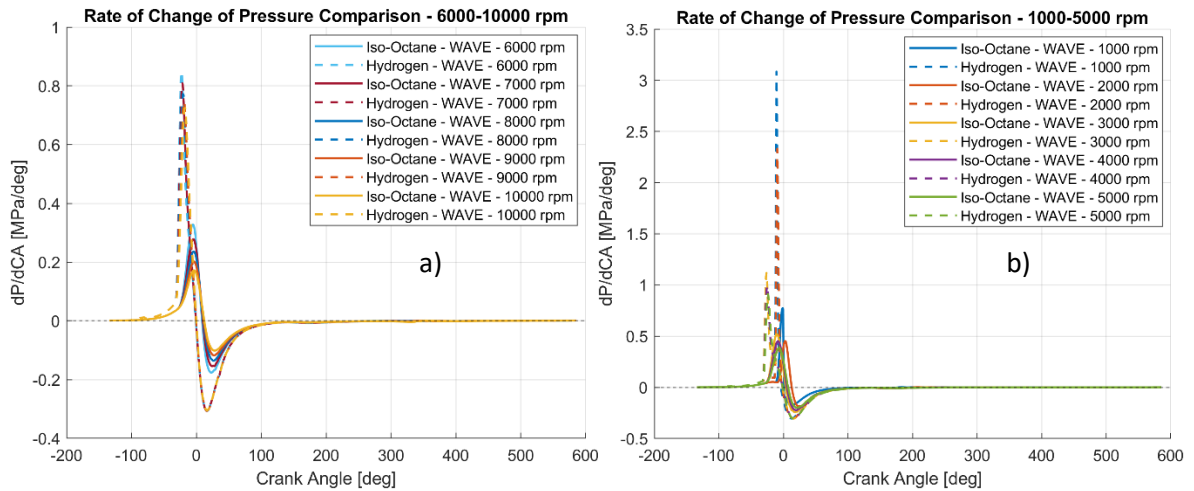


Figure 69: Iso-octane and hydrogen WAVE rate of change of cylinder pressure: a) 1000–5000 rpm, b) 6000-10000 rpm

Table 35: Iso-octane and hydrogen WAVE cylinder pressure and temperature characteristics

RPM	PMAX OCT. (MPa)	PMAX H ₂ (MPa)	MAX. DP/DCA OCT. (MPa/°CA)	MAX. DP/DCA H ₂ (MPa/°CA)	TMAX OCT. (K)	TMAX H ₂ (K)
2000	7.324	12.689	0.453	2.332	2492	3043
3000	9.791	12.864	0.522	1.119	2707	3094
4000	9.191	13.023	0.453	0.993	2643	3121
5000	8.476	13.118	0.387	0.912	2560	3130
6000	7.743	13.170	0.329	0.849	2468	3135
7000	7.031	13.187	0.279	0.809	2370	3135
8000	6.351	13.193	0.236	0.773	2254	3133
9000	5.760	13.178	0.202	0.747	2146	3127
10000	5.218	13.133	0.172	0.725	2043	3115

The effect of the different combustion rates on cylinder pressure and temperature for Forte is shown in Figures 70–72, with the principal values summarised in Table 36.

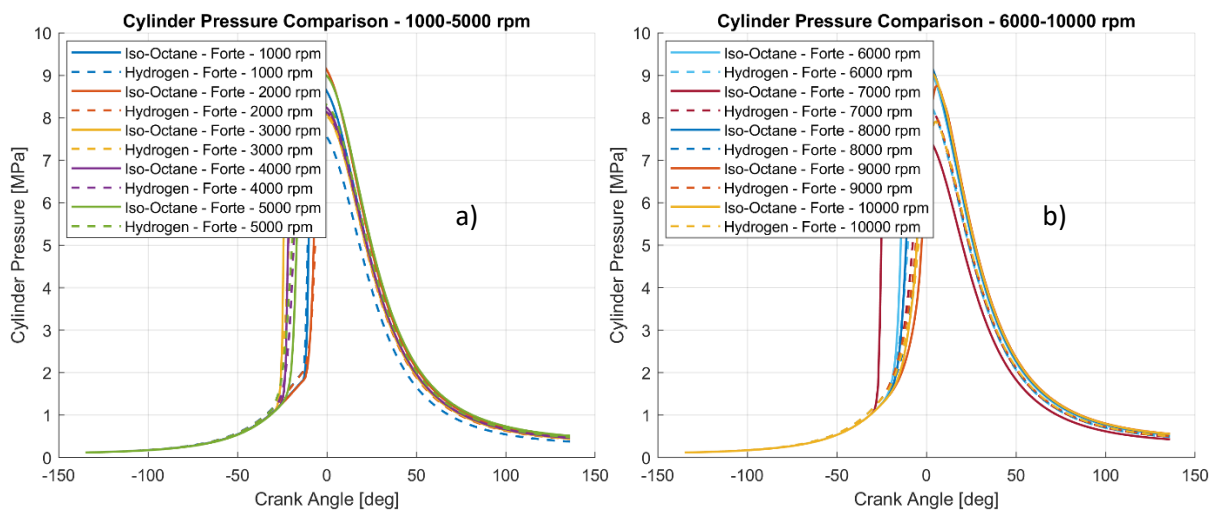


Figure 70: Iso-octane and hydrogen Forte cylinder pressure: a) 1000–5000 rpm, b) 6000–10000 rpm

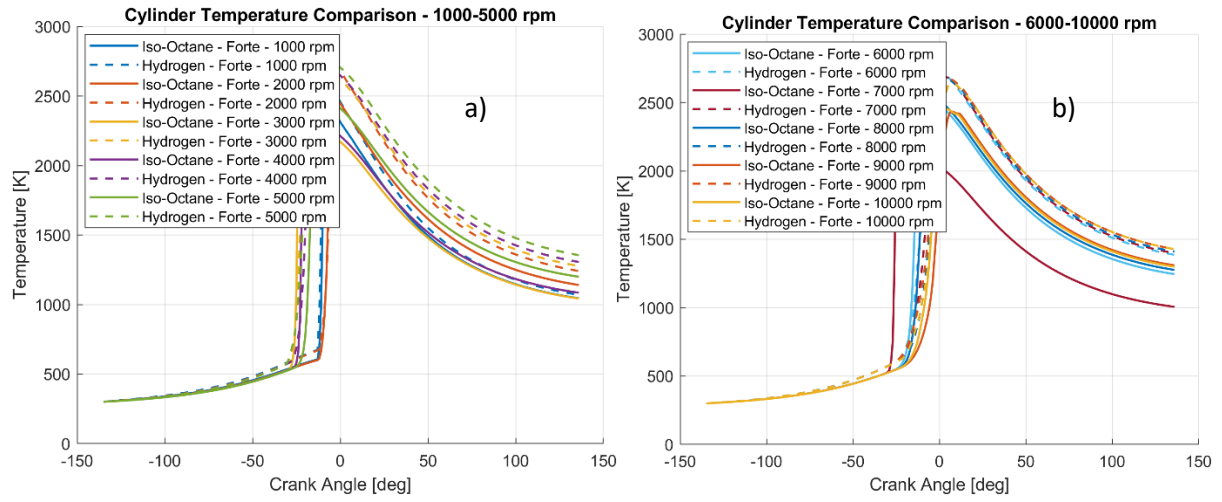


Figure 71: Iso-octane and hydrogen Forte cylinder temperature: a) 1000–5000 rpm, b) 6000-10000 rpm

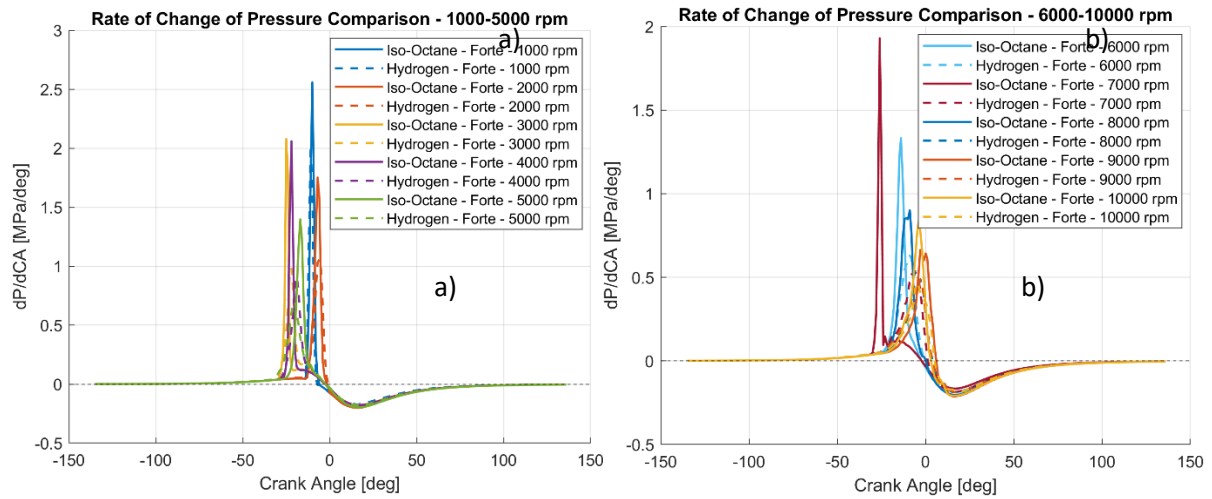


Figure 72: Iso-octane and hydrogen Forte rate of change of cylinder pressure: a) 1000–5000 rpm, b) 6000-10000 rpm

Table 36: Iso-octane and hydrogen Forte cylinder pressure and temperature characteristics

RPM	P_{MAX} OCT./H₂ (MPa)	MAX RATE OF CHANGE OF PRESSURE OCT./H₂ (MPa/°CA)	T_{MAX} OCT./H₂ (K)
1000	8.854 / 7.760	2.560 / 2.024	2465 / 2659
2000	9.189 / 8.237	1.752 / 1.056	2472 / 2684
3000	8.132 / 8.076	2.081 / 0.979	2389 / 2719
4000	8.287 / 8.165	2.060 / 0.911	2398 / 2723
5000	9.031 / 8.331	1.398 / 0.694	2498 / 2758
6000	9.319 / 8.367	1.333 / 0.630	2526 / 2727
7000	7.601 / 8.284	1.930 / 0.532	2237 / 2698
8000	9.351 / 8.028	0.901 / 0.545	2508 / 2663
9000	8.763 / 8.057	0.664 / 0.489	2434 / 2672
10000	9.045 / 7.911	0.833 / 0.469	2454 / 2641

WAVE predicts substantially greater hydrogen pressure, temperature and rate of change of pressure through the speed range. At 10,000 rpm hydrogen reaches 13.13 MPa and 3115 K compared with 5.22 MPa and 2043 K for iso-octane, while maximum rate of change of pressure is 0.725 compared with 0.172 MPa/°CA. Forte also predicts consistently greater hydrogen peak temperature, including 2758 K at 5000 rpm and 2641 K at 10,000 rpm compared with 2498 and 2454 K for iso-octane. Its pressure response differs, with hydrogen reaching 7.91 MPa at 10,000 rpm compared with 9.04 MPa for iso-octane, while maximum rate of change of pressure is 0.469 compared with 0.833 MPa/°CA.

5.1.4.3 Engine performance

The influence of the different combustion characteristics on engine output for WAVE is shown by the single-cylinder indicated-power comparison in Figure 73, with the corresponding performance parameters summarised in Table 37.

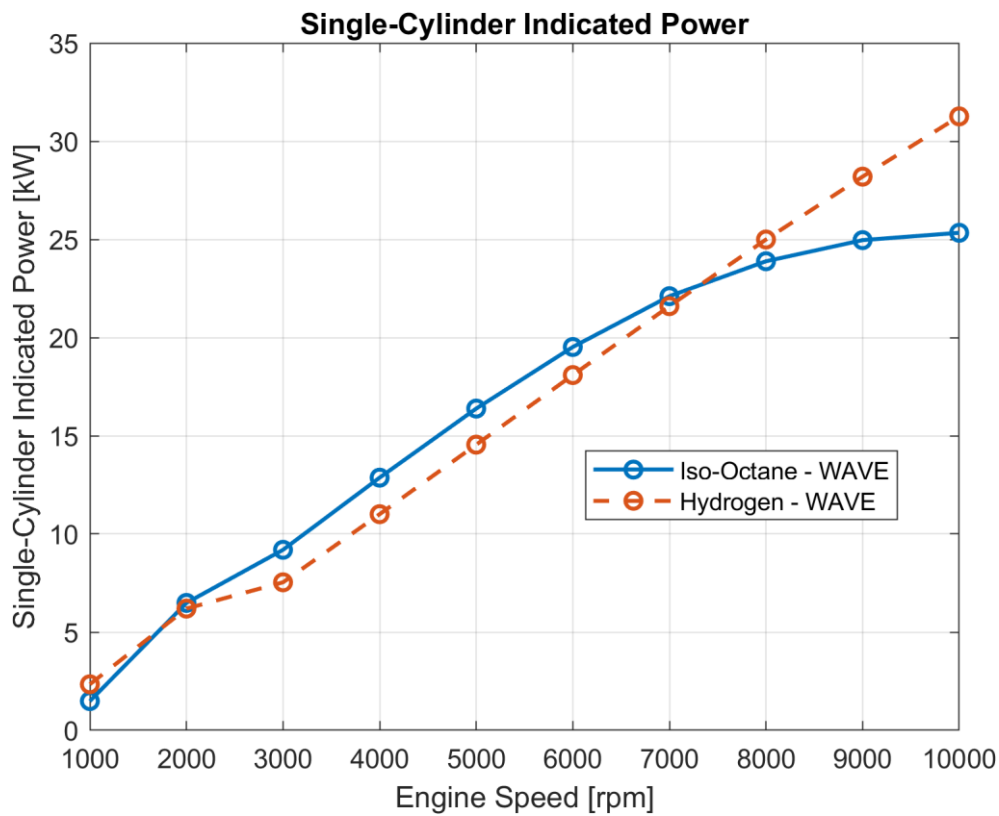


Figure 73: Iso-octane and hydrogen WAVE single-cylinder indicated power across engine speed

Table 37: Iso-octane and hydrogen WAVE indicated performance

RPM	IMEP OCT. (MPA)	IMEP H ₂ (MPA)	WORK OCT. (KJ/CYCLE)	WORK H ₂ (KJ/CYCLE)	POWER OCT. (KW)	POWER H ₂ (KW)
2000	1.381	1.319	0.0973	0.0930	6.49	6.20
3000	1.304	1.069	0.0919	0.0754	9.19	7.54
4000	1.369	1.171	0.0965	0.0826	12.87	11.01
5000	1.395	1.238	0.0983	0.0873	16.38	14.55
6000	1.385	1.283	0.0976	0.0905	19.52	18.09
7000	1.344	1.314	0.0948	0.0926	22.11	21.60
8000	1.271	1.330	0.0896	0.0937	23.89	25.00
9000	1.181	1.334	0.0832	0.0940	24.96	28.20
10000	1.079	1.331	0.0760	0.0938	25.34	31.27

The influence of the different combustion characteristics on engine output for Forte is shown by the single-cylinder indicated-power comparison in Figure 74, with the corresponding performance parameters summarised in Table 38.

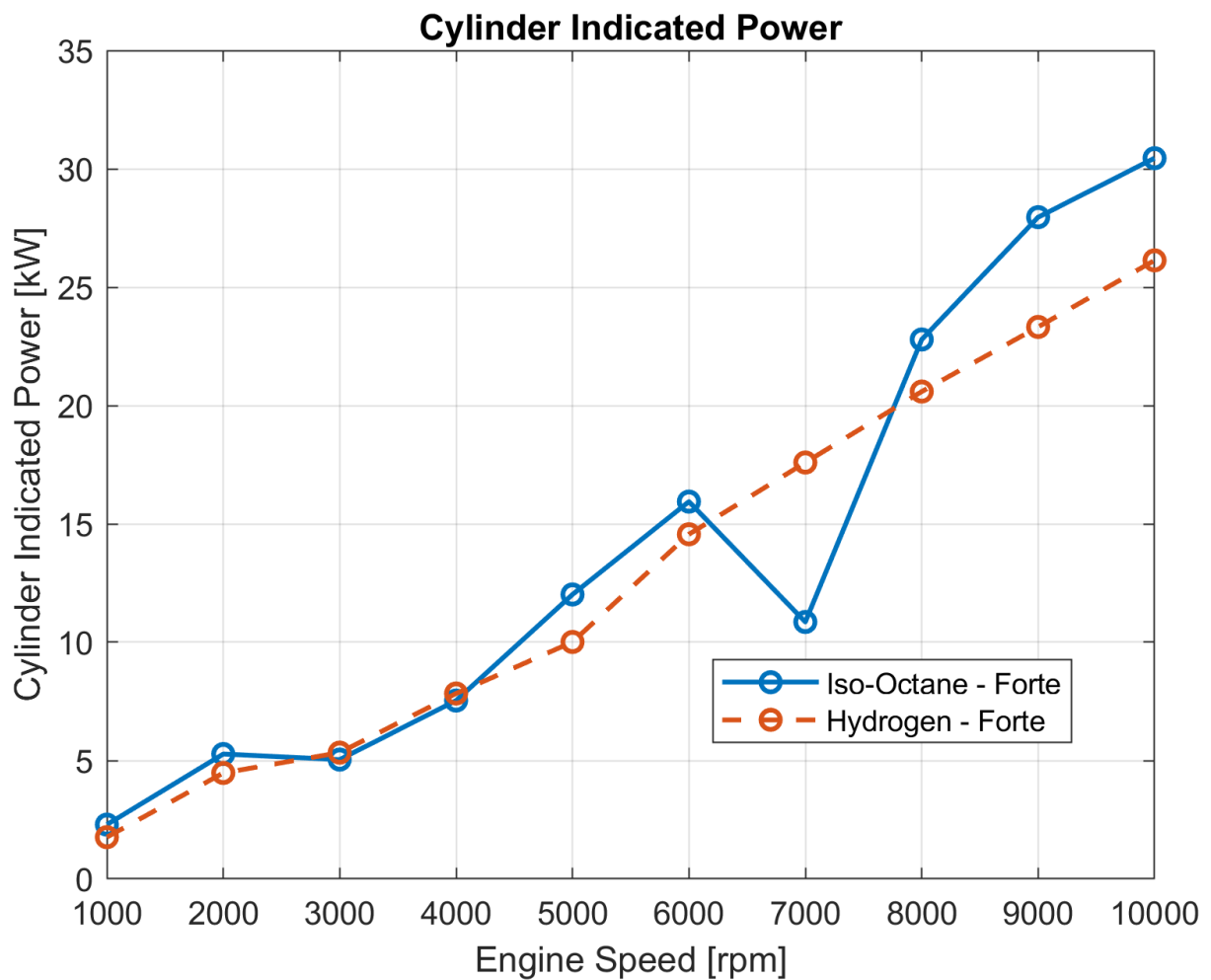


Figure 74: Iso-octane and hydrogen Forte single-cylinder indicated power across engine speed

Table 38: Iso-octane and hydrogen Forte indicated performance

RPM	IMEP OCT./H₂ (MPa)	INDICATED WORK OCT./H₂ (KJ/CYCLE)	INDICATED POWER OCT./H₂ (KW)
1000	0.976 / 0.750	0.2752 / 0.2114	9.17 / 7.05
2000	1.123 / 0.953	0.3165 / 0.2686	21.10 / 17.91
3000	0.715 / 0.757	0.2016 / 0.2134	20.16 / 21.34
4000	0.802 / 0.834	0.2261 / 0.2350	30.14 / 31.33
5000	1.023 / 0.852	0.2884 / 0.2403	48.07 / 40.04
6000	1.131 / 1.033	0.3188 / 0.2912	63.77 / 58.24
7000	0.660 / 1.070	0.1861 / 0.3016	43.42 / 70.37
8000	1.213 / 1.096	0.3419 / 0.3089	91.16 / 82.37
9000	1.322 / 1.103	0.3728 / 0.3110	111.83 / 93.28
10000	1.296 / 1.112	0.3654 / 0.3136	121.80 / 104.54

WAVE predicts lower hydrogen indicated output through much of the intermediate-speed range before the relationship reverses at high speed. At 5000 rpm hydrogen produces 14.55 kW compared with 16.38 kW for iso-octane, whereas at 10,000 rpm the values are 31.27 and 25.34 kW respectively, with hydrogen indicated work reaching 0.0938 kJ/cycle compared with 0.0760 kJ/cycle. Forte also gives a speed-dependent response, with hydrogen exceeding iso-octane at 3000–4000 and 7000 rpm but producing lower output at the upper speeds. At 10,000 rpm hydrogen reaches 104.54 kW and 1.112 MPa IMEP compared with 121.80 kW and 1.296 MPa for iso-octane.

5.1.4.4 Fuel Consumption and Efficiency

The effect of fuel type on energy conversion for WAVE is shown by the thermal-efficiency and heat-transfer comparisons in Figures 75 and 76, with the corresponding fuel-consumption parameters summarised in Table 39.

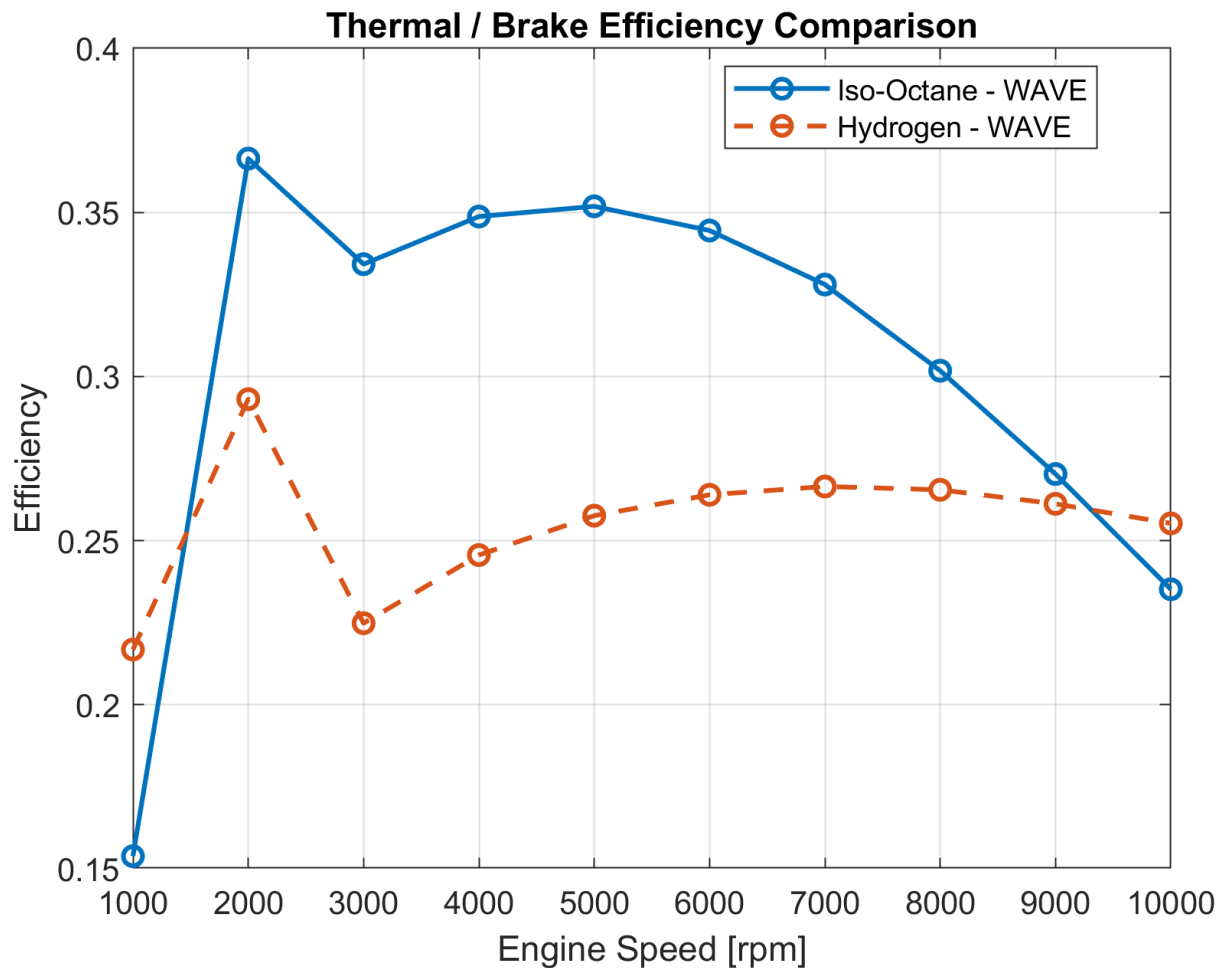


Figure 75: Iso-octane and hydrogen WAVE thermal efficiency across engine speed

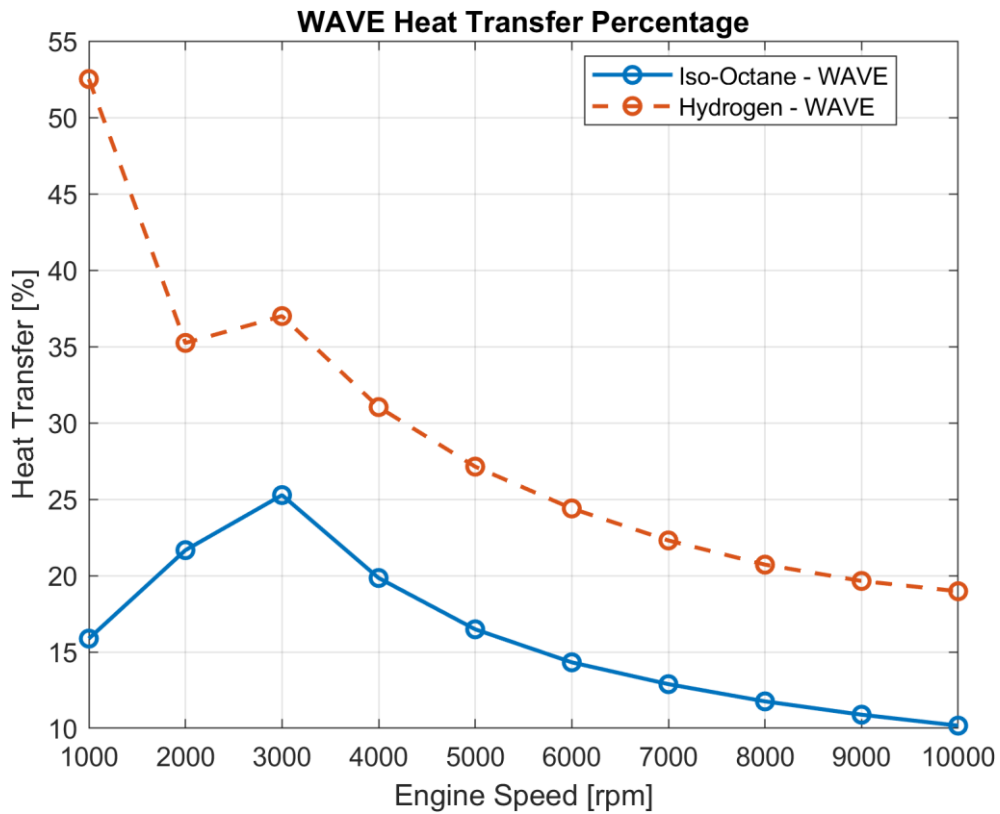


Figure 76: Iso-octane and hydrogen WAVE heat-transfer percentage across engine speed

Table 39: Iso-octane and hydrogen WAVE fuel consumption and efficiency

RPM	FUEL/CYCLE OCT. (MG)	FUEL/CYCLE H ₂ (MG)	FUEL FLOW OCT. (KG/H)	FUEL FLOW H ₂ (KG/H)	ISFC OCT. (G/KWH)	ISFC H ₂ (G/KWH)	HTH OCT. (%)	HTH H ₂ (%)
2000	5.415	2.319	1.300	0.557	200.3	89.8	36.64	29.31
3000	5.415	2.319	1.949	0.835	212.1	110.8	33.42	22.48
4000	5.415	2.319	2.599	1.113	202.0	101.1	34.87	24.56
5000	5.415	2.319	3.249	1.392	198.3	95.7	35.19	25.76
6000	5.415	2.319	3.899	1.670	199.7	92.3	34.45	26.40
7000	5.415	2.319	4.549	1.948	205.7	90.2	32.80	26.65
8000	5.415	2.319	5.199	2.227	217.6	89.1	30.18	26.55
9000	5.415	2.319	5.848	2.505	234.3	88.8	27.03	26.12

10000	5.415	2.319	6.498	2.783	256.5	89.0	23.52	25.53
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The effect of fuel type on energy conversion for Forte is shown by the thermal-efficiency and heat-transfer comparisons in Figures 77 and 78, with the corresponding fuel-consumption parameters summarised in Table 40.

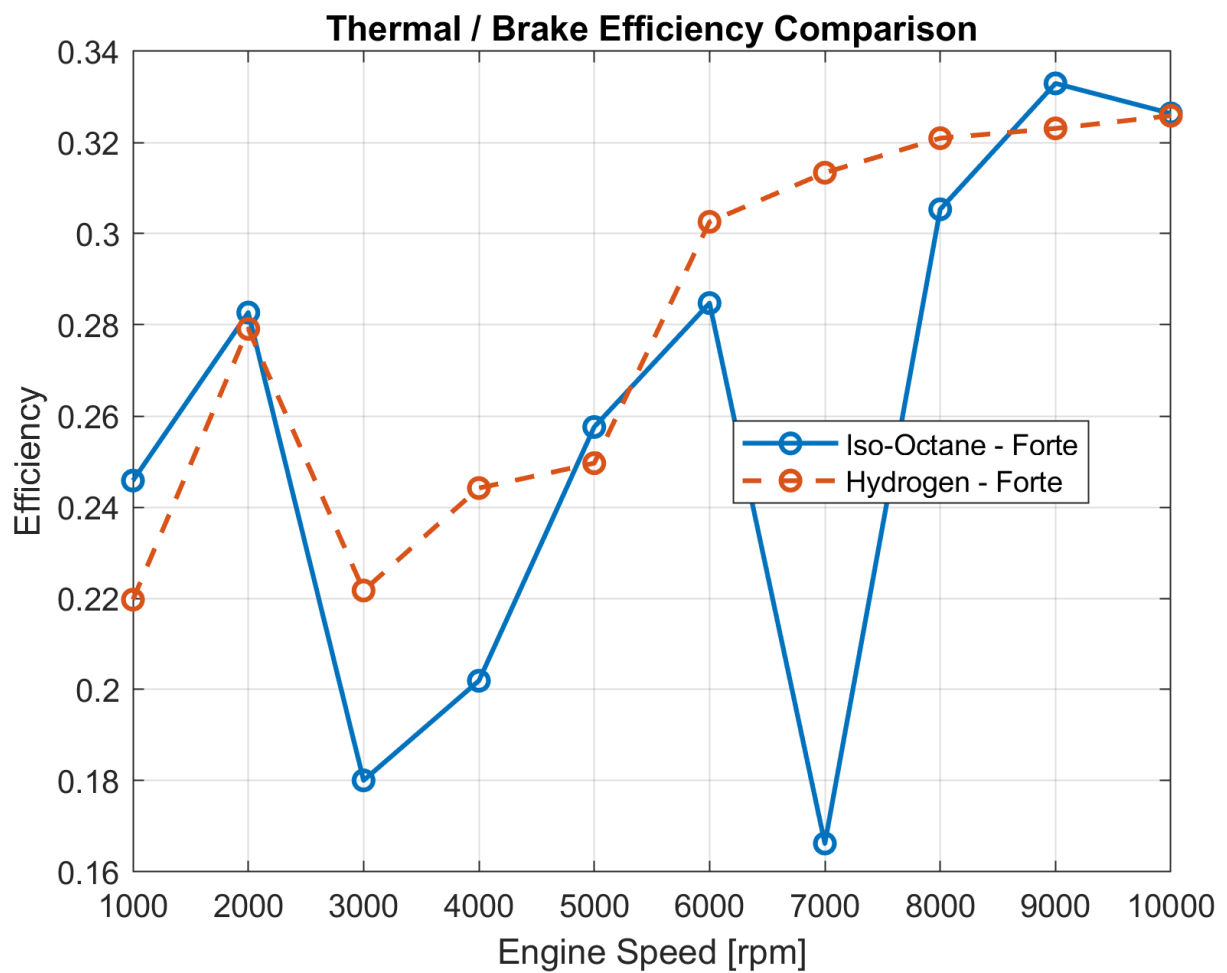


Figure 77: Iso-octane and hydrogen Forte thermal efficiency across engine speed

Table 40: Iso-octane and hydrogen Forte fuel consumption and efficiency

RPM	FUEL MASS/CYCLE OCT./H₂ (MG)	FUEL FLOW OCT./H₂ (KG/H)	ISFC OCT./H₂ (G/KWH)	BRAKE EFFICIENCY OCT./H₂ (%)
1000	25.445 / 8.020	3.053 / 0.962	332.8 / 136.5	24.58 / 21.97
2000	25.445 / 8.020	6.107 / 1.925	289.4 / 107.5	28.27 / 27.90
3000	25.445 / 8.020	9.160 / 2.887	454.4 / 135.3	18.01 / 22.16
4000	25.445 / 8.020	12.214 / 3.849	405.2 / 122.9	20.19 / 24.41
5000	25.445 / 8.020	15.267 / 4.812	317.6 / 120.2	25.76 / 24.96
6000	25.445 / 8.020	18.321 / 5.774	287.3 / 99.1	28.48 / 30.25
7000	25.445 / 8.020	21.374 / 6.737	492.2 / 95.7	16.62 / 31.30
8000	25.445 / 8.020	24.428 / 7.699	268.0 / 93.5	30.53 / 32.06
9000	25.445 / 8.020	27.481 / 8.661	245.7 / 92.9	33.30 / 32.27
10000	25.445 / 8.020	30.535 / 9.624	250.7 / 92.1	32.64 / 32.59

Both models predict substantially lower hydrogen fuel mass and mass-based ISFC because of its much greater gravimetric energy content. In WAVE, hydrogen efficiency remains below iso-octane through much of the intermediate-speed range, including 25.76% compared with 35.19% at 5000 rpm, before reaching 25.53% compared with 23.52% at 10,000 rpm. Forte predicts closer efficiencies at several speeds: hydrogen reaches 21.97% compared with 24.58% at 1000 rpm, 24.96% compared with 25.76% at 5000 rpm and 32.59% compared with 32.64% at 10,000 rpm. At the latter point, hydrogen ISFC is 92.1 g/kWh compared with 250.7 g/kWh for iso-octane.

5.2 Comparison of 1D WAVE and 3D Forte Results

5.2.1 Combustion Model Response

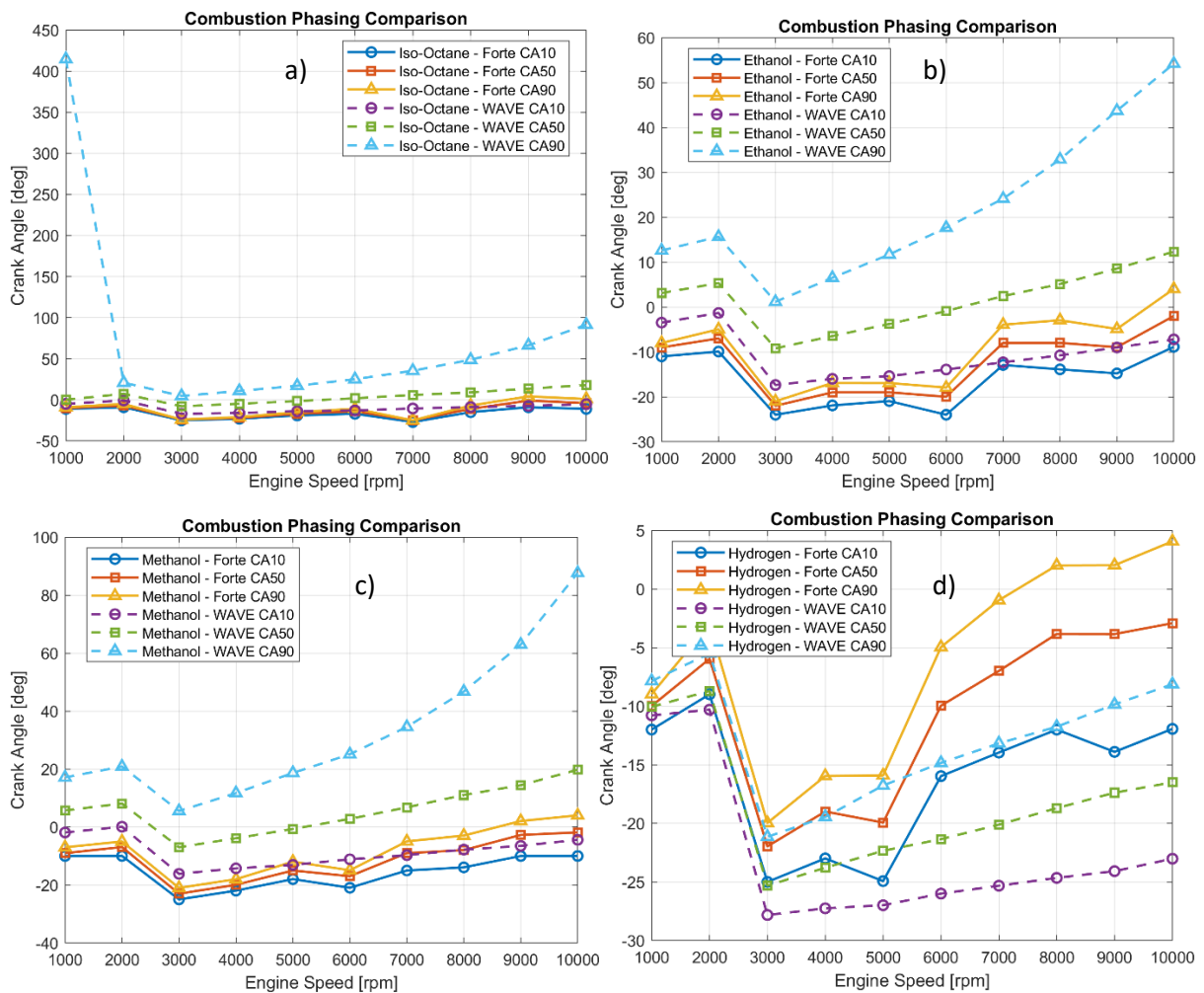


Figure 78: Combustion phasing predicted by WAVE and Forte for (a) iso-octane, (b) ethanol, (c) methanol and (d) hydrogen over 1000–10,000 rpm

Table 41: WAVE and Forte combustion phasing and duration at selected engine speeds

FUEL	RPM	WAVE CA50	FORTE CA50	WAVE CA10–90	FORTE CA10–90	WAVE CA50–90	FORTE CA50–90
ISO- OCTANE	3000	-7.81	-23.99	21.67	1.01	12.50	~0.00*
	6000	1.88	-13.93	38.09	6.13	23.22	3.11
METHANOL	10000	17.93	-3.96	96.70	12.05	73.44	5.03
	3000	-6.98	-23.00	21.67	3.97	12.50	2.00
	6000	2.81	-16.97	36.23	6.04	22.29	2.06
ETHANOL	10000	19.79	-1.85	92.06	14.01	67.87	5.88
	3000	-9.20	-21.99	18.52	3.00	10.37	1.01
	6000	-0.90	-19.97	31.59	6.04	18.58	2.04
HYDROGEN	10000	12.34	-1.99	61.39	12.91	41.85	6.04
	3000	-25.32	-21.97	6.67	5.03	4.17	2.00
	6000	-21.35	-9.96	11.16	11.01	6.51	5.00
	10000	-16.50	-2.92	14.89	16.01	8.38	6.99

Figure 78 shows that the WAVE–Forte combustion disagreement is strongly fuel-dependent. For iso-octane, methanol and ethanol, WAVE combustion increasingly lengthens with speed, predominantly through the latter burn period, whereas Forte retains shorter, more concentrated combustion. As quantified in Table 41, at 10,000 rpm WAVE/Forte CA10–90 durations are 96.70/12.05°CA for iso-octane, 92.06/14.01°CA for methanol and 61.39/12.91°CA for ethanol. Hydrogen differs fundamentally: durations converge to 14.89/16.01°CA, but CA50 remains substantially earlier in WAVE at –16.50° compared with –2.92° ATDC in Forte.

This behaviour is consistent with the different flame-development formulations used by the two models. In WAVE, the SI Predictive burning rate depends on turbulent flame speed and flame-front area, with the Burning Area Decay Exponent predominantly affecting the latter combustion period (Peng, 2023). The increasing CA50–90 separation for the liquid fuels is therefore consistent with sensitivity in WAVE's late-stage flame-area treatment, although it does not independently prove that this parameter alone causes the disagreement. Fuel dependence is also expected because laminar

burning velocity is explicitly fuel-specific and varies with mixture condition, temperature and pressure (Gülder, 1984). In Forte, DPIK kernel growth depends on spark-energy deposition and turbulence before transition to G-equation flame propagation (Tan and Reitz, 2006), while G-equation flame-development parameters have been shown to alter combustion predictions as engine speed changes (Liu, Szybist and Dumitrescu, 2018). Together with the Forte sensitivity studies in Section 5.1, the results therefore indicate that the WAVE–Forte disagreement arises from different fuel- and speed-dependent flame-development responses, compounded by Forte configuration sensitivity, rather than dimensionality alone.

5.2.2 Cylinder Pressure Response

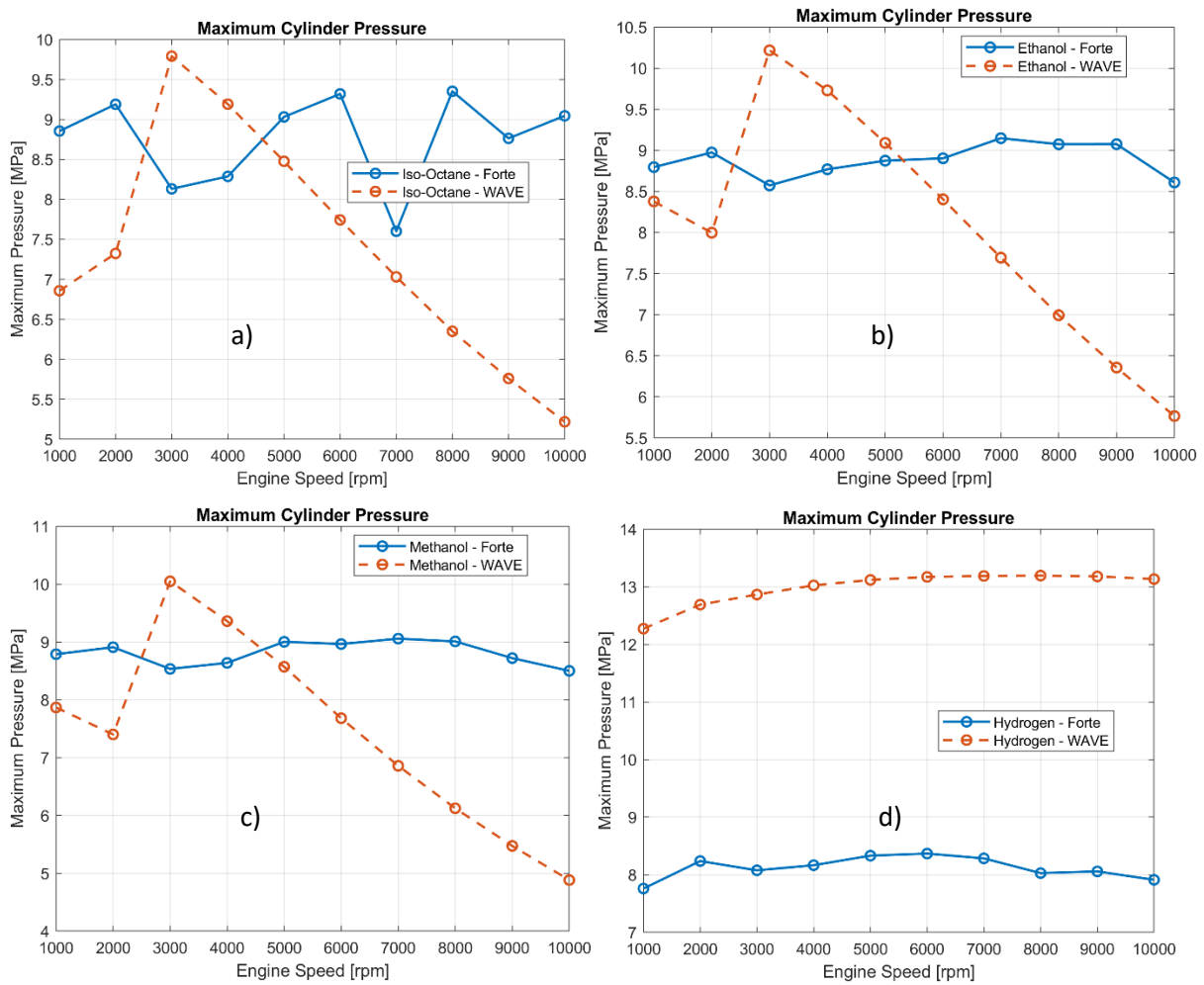


Figure 79: Peak cylinder pressure predicted by WAVE and Forte for (a) iso-octane, (b) ethanol, (c) methanol and (d) hydrogen over 1000–10,000 rpm

Table 42: WAVE and Forte peak cylinder pressure and maximum rate of change of pressure at selected engine speeds

FUEL	SPEED (RPM)	WAVE P _{MAX} (MPa)	FORTE P _{MAX} (MPa)	WAVE P _{MAX} CA (°ATDC)	FORTE P _{MAX} CA (°ATDC)	WAVE MAX DP/DCA (MPa/°CA)	FORTE MAX DP/DCA (MPa/°CA)
ISO- OCTANE	3000	9.79	8.13	3.86	-2.96	0.522	2.081
	6000	7.74	9.32	8.39	-1.99	0.329	1.333
	10000	5.22	9.04	12.34	3.16	0.172	0.833
METHANOL	3000	10.05	8.53	4.69	-1.95	0.554	1.648
	6000	7.68	8.96	10.24	-1.92	0.336	1.116
	10000	4.88	8.50	14.20	5.11	0.173	0.665
ETHANOL	3000	10.22	8.57	2.65	-2.99	0.586	1.676
	6000	8.40	8.90	8.39	-1.90	0.384	1.208
	10000	5.77	8.61	12.34	5.10	0.211	0.634
HYDROGEN	3000	12.86	8.08	-1.14	-2.00	1.119	0.979
	6000	13.17	8.37	-0.90	0.18	0.849	0.630
	10000	13.13	7.91	0.25	5.04	0.725	0.469

Figure 79 shows that the WAVE–Forte pressure disagreement changes with both fuel and engine speed rather than forming a consistent model offset. At 3000 rpm, WAVE predicts greater peak pressure than Forte for all four fuels, including 9.79/8.13 MPa for iso-octane and 10.22/8.57 MPa for ethanol. For the liquid fuels this relationship subsequently reverses as speed increases. At 10,000 rpm, WAVE/Forte peak pressures are 5.22/9.04 MPa for iso-octane, 4.88/8.50 MPa for methanol and 5.77/8.61 MPa for ethanol (Table 42). WAVE peak-pressure timing also moves progressively further ATDC, reaching 12.34°, 14.20° and 12.34° ATDC respectively.

Hydrogen behaves differently. WAVE remains substantially higher than Forte across the selected speeds, predicting 13.13 MPa compared with 7.91 MPa at 10,000 rpm. The associated maximum rate of change of pressure is also higher in WAVE for hydrogen, 0.725 against 0.469 MPa/°CA, whereas Forte remains substantially higher for the liquid fuels. These pressure trends follow the

combustion behaviour identified in Section 6.2.1: delayed and prolonged WAVE liquid-fuel combustion increasingly shifts the pressure response into expansion, while the much earlier WAVE hydrogen combustion produces the opposite pressure relationship.

5.2.3 Thermodynamic and Heat-Transfer Response

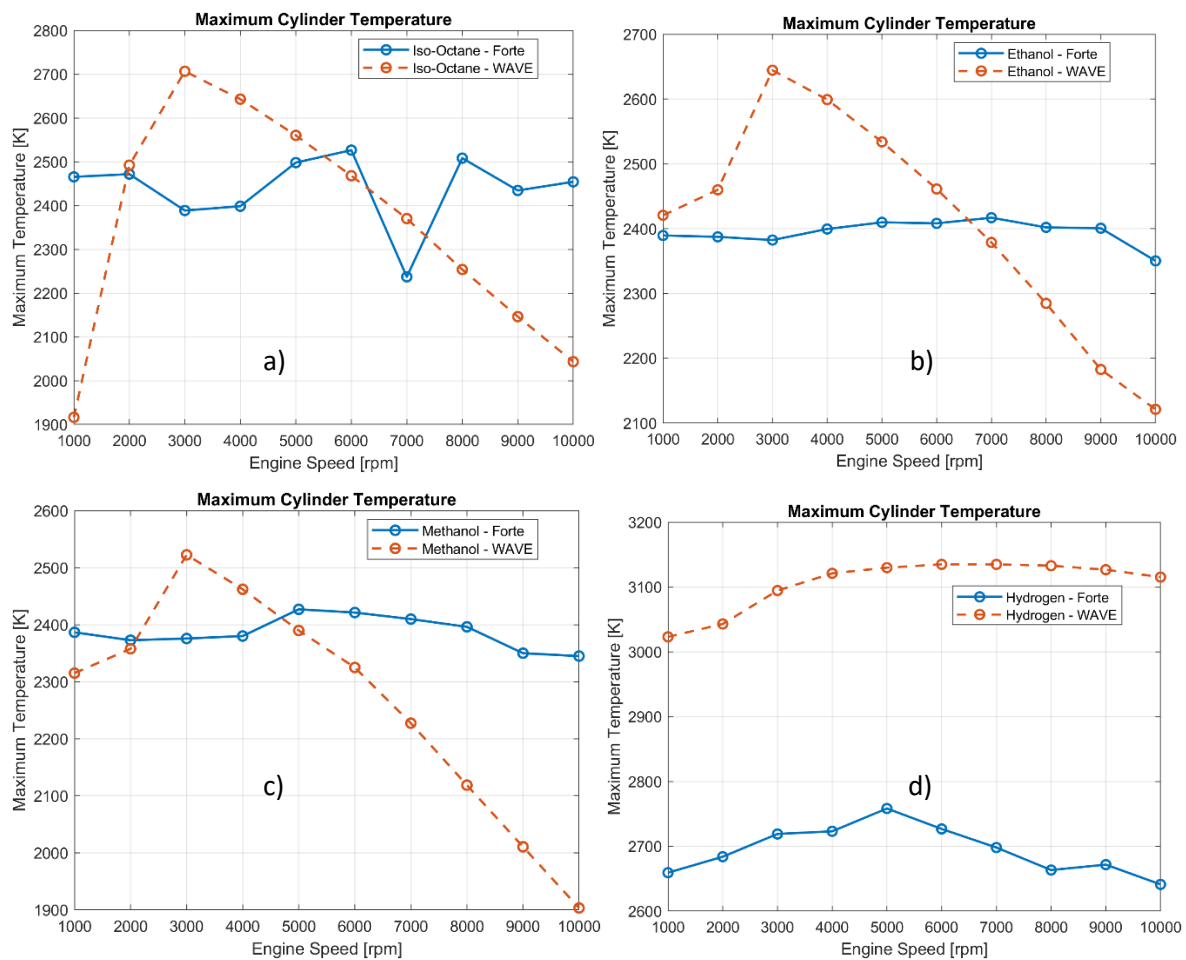


Figure 80: Maximum cylinder temperature predicted by WAVE and Forte for (a) iso-octane, (b) ethanol, (c) methanol and (d) hydrogen over 1000–10,000 rpm

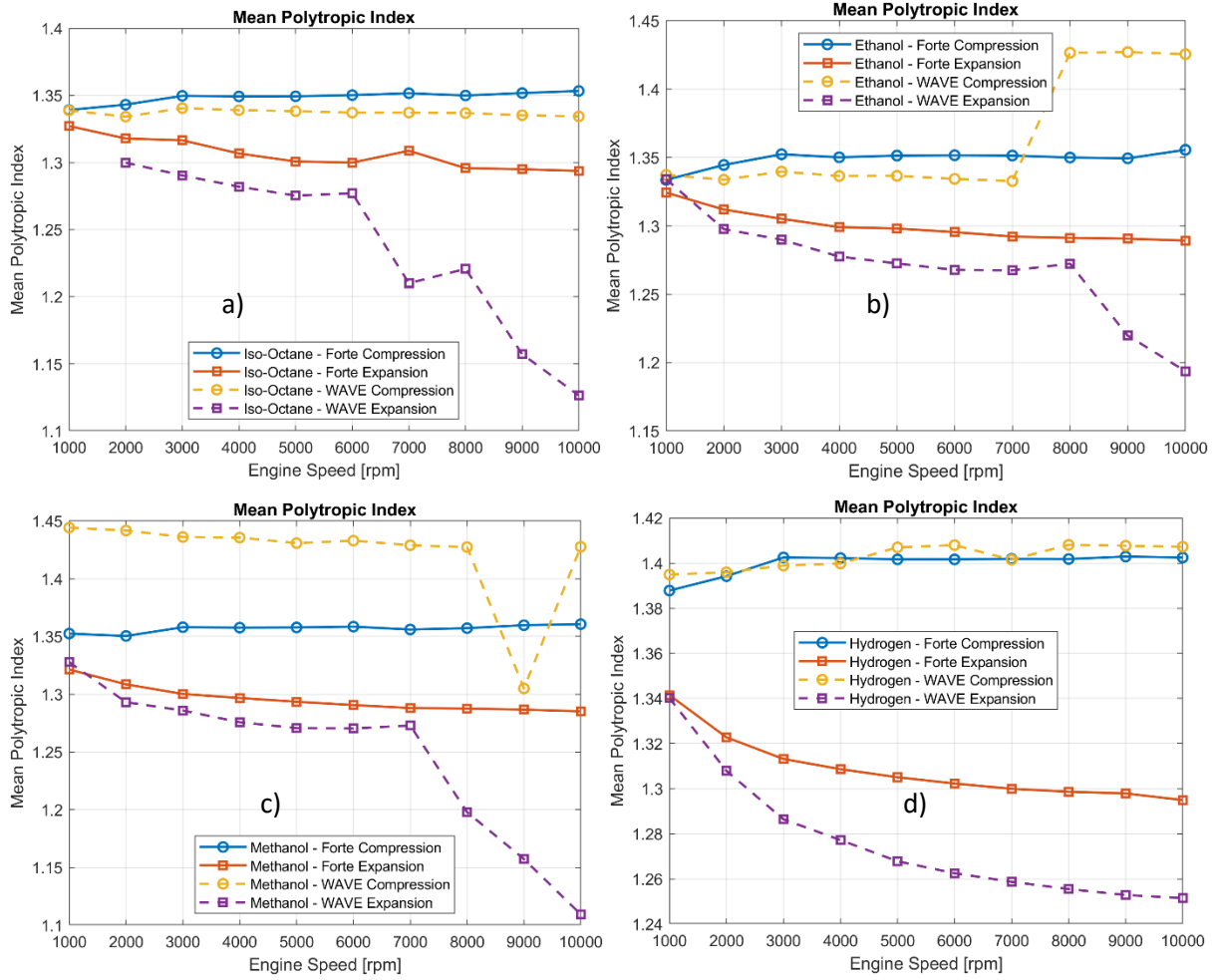


Figure 81: Compression and expansion polytropic indices calculated from WAVE and Forte cylinder-pressure histories for (a) iso-octane, (b) ethanol, (c) methanol and (d) hydrogen over 1000–10,000 rpm

Table 43: WAVE and Forte maximum cylinder temperature and compression and expansion polytropic indices at selected engine speeds

FUEL	SPEED (RPM)	WAVE TMAX (K)	FORTE TMAX (K)	WAVE COMPRESSION N	FORTE COMPRESSION N	WAVE EXPANSION N	FORTE EXPANSION N
ISO-OCTANE	3000	2706.60	2388.60	1.3229	1.3440	1.2819	1.3081
	6000	2468.05	2526.48	1.3207	1.3427	1.2724	1.2935
	10000	2043.30	2454.19	1.3181	1.3429	1.1289	1.2904
METHANOL	3000	2522.47	2375.73	1.1309	1.3515	1.2779	1.2933
	6000	2324.96	2421.29	1.1379	1.3506	1.2642	1.2842
	10000	1903.24	2344.90	1.1468	1.3500	1.1094	1.2824
ETHANOL	3000	2644.36	2382.34	1.2208	1.3446	1.2805	1.2977
	6000	2461.21	2408.13	1.2189	1.3439	1.2592	1.2886
	10000	2121.14	2350.21	1.2284	1.3431	1.1891	1.2862
HYDROGEN	3000	3094.47	2718.96	1.7343	1.3970	1.2763	1.3058
	6000	3134.87	2726.70	1.7737	1.3953	1.2513	1.2947
	10000	3115.23	2641.11	1.7649	1.3955	1.2408	1.2909

Figure 80 shows that the temperature disagreement follows the fuel-dependent reversal already observed in combustion and cylinder pressure. At 3000 rpm WAVE predicts higher maximum temperature for all four fuels, but this relationship reverses at high speed for the liquid fuels. At 10,000 rpm, WAVE/Forte temperatures are 2043/2454 K for iso-octane, 1903/2345 K for methanol and 2121/2350 K for ethanol. Hydrogen remains fundamentally different, with WAVE predicting 3115 K compared with 2641 K in Forte (Table 43), consistent with its substantially earlier WAVE combustion identified in Section 6.2.1.

The polytropic indices in Figure 81 show that the thermodynamic disagreement is not restricted to the combustion and expansion periods. Forte compression indices remain comparatively stable with speed for each fuel, whereas WAVE exhibits strong fuel dependence, ranging at 10,000 rpm from 1.147 for methanol to 1.765 for hydrogen. Expansion behaviour also separates increasingly for the

liquid fuels at high speed: iso-octane reaches 1.129 in WAVE compared with 1.290 in Forte, while methanol reaches 1.109/1.282. These low WAVE expansion indices coincide with the prolonged high-speed combustion identified in Section 6.2.1, where continuing heat release during expansion can sustain cylinder pressure as volume increases. Consequently, the expansion-index difference cannot be interpreted as a heat-transfer difference alone. The compression-index differences additionally indicate that model thermodynamic treatment before combustion also contributes to the WAVE–Forte disagreement, particularly for methanol and hydrogen.

5.2.4 Hydrogen

Table 44: Difference between WAVE and Forte combustion and thermodynamic predictions for each fuel at 10,000 rpm ($\Delta = \text{WAVE} - \text{Forte}$).

FUEL	ΔCA_{50} (°CA)	ΔCA_{10-90} (°CA)	$\Delta \text{P}_{\text{MAX}}$ (MPa)	$\Delta \text{T}_{\text{MAX}}$ (K)	$\Delta \text{EXPANSION N}$
ISO-OCTANE	+21.89	+84.66	−3.83	−411	−0.162
METHANOL	+21.64	+78.05	−3.62	−442	−0.173
ETHANOL	+14.33	+48.48	−2.84	−229	−0.097
HYDROGEN	−13.59	−1.12	+5.22	+474	−0.050

Table 44 demonstrates that fuel changes both the magnitude and direction of the WAVE–Forte disagreement. At 10,000 rpm, iso-octane and methanol produce the greatest combustion-duration differences, with WAVE CA10–90 exceeding Forte by 84.66 and 78.05°CA respectively. Ethanol follows the same response but with a smaller 48.48°CA difference. These increasingly prolonged WAVE liquid-fuel predictions correspond with lower peak pressure and temperature: the WAVE–Forte peak-pressure differences are −3.83, −3.62 and −2.84 MPa for iso-octane, methanol and ethanol respectively.

Hydrogen reverses this relationship. Its WAVE and Forte combustion durations differ by only −1.12°CA, but WAVE CA50 occurs 13.59°CA earlier. WAVE consequently predicts 5.22 MPa greater peak pressure and 474 K greater maximum temperature. This reversal confirms that the model disagreement is not a fixed WAVE or Forte bias. Instead, the two combustion formulations respond differently to fuel, with the resulting differences in combustion phasing and duration propagating into pressure, temperature and expansion behaviour. Together with the Forte sensitivities identified

in Section 5.1, the comparison indicates that predicted fuel effects depend on both combustion-model formulation and model configuration, rather than dimensionality alone

6 Model Limitations and Critical Evaluation

The results are subject to limitations in both modelling approaches. Most importantly, experimental combustion and cylinder-pressure data were unavailable; therefore, agreement between WAVE and Forte cannot establish the accuracy of either model.

6.1 WAVE

The WAVE SI Predictive model required parameters obtained or calculated from several literature sources. Although individually justified, combining parameters derived under different experimental conditions may reduce consistency compared with a complete fuel-specific parameter set from a single validated source. Valve timing was also unavailable and was therefore assumed by centring the measured lift profiles on their respective gas-exchange strokes. This introduces uncertainty in cylinder filling and residual-gas conditions.

6.2 Forte

The intended local mesh-refinement strategy could not be implemented, so representative nominal mesh sizes were used instead. These do not correspond to uniform global cell dimensions, as cell size varies throughout the moving mesh. Furthermore, full mesh independence was not achieved, with the 2 mm mesh selected based on physical plausibility rather than experimental validation.

The Forte combustion-chamber geometry was also simplified, limiting the accuracy with which local flow, turbulence and flame development could be resolved. Consequently, Forte's greater spatial resolution should not itself be interpreted as greater accuracy without validated geometry, mesh refinement and appropriate fuel-specific model parameters.

7 Engineering Implications

The modelling approach used in this study reflects an important stage of engine development, where simulation may be required before a physical prototype and corresponding validation data are available. Under these conditions, WAVE is advantageous for early design and concept screening because its lower computational requirement enables rapid comparison across fuels and operating conditions. Forte provides greater spatial detail of in-cylinder flow and combustion, but requires greater computational resources, geometric definition and model setup.

The disagreement observed between WAVE and Forte demonstrates that greater model complexity cannot itself be taken as evidence of greater accuracy, particularly when validated setup parameters are unavailable. Model selection should therefore depend on the required level of detail and stage of development: WAVE may be sufficient for initial comparative studies, while 3D CFD becomes more valuable where local combustion, turbulence or chamber geometry must be investigated. As prototype data become available, both models can then be validated and refined before being relied upon for detailed design decisions.

8 Conclusions

This study compared Ricardo WAVE and ANSYS Forte predictions for a Suzuki GSX1100F spark-ignition engine operating with iso-octane, methanol, ethanol and hydrogen. The results show that disagreement between the 1D and 3D approaches is strongly dependent on both engine speed and fuel.

For the liquid fuels, WAVE generally predicted increasingly delayed and prolonged combustion as engine speed increased, while Forte predicted much shorter combustion durations. For iso-octane, for example, WAVE CA_{10–90} increased from 21.67°CA at 3000 rpm to 96.70°CA at 10,000 rpm, whereas Forte increased from 1.01°CA to 12.05°CA over the same range. This difference in combustion behaviour was reflected in the pressure and temperature predictions, with Forte generally predicting higher values for the liquid fuels at high engine speed.

Hydrogen showed a different response, with WAVE predicting substantially higher peak pressure than Forte at high speed, reaching 13.13 MPa compared with 7.91 MPa at 10,000 rpm. This reversal demonstrates that the WAVE–Forte disagreement is not a fixed bias between 1D and 3D modelling but depends on how each combustion model responds to fuel properties and engine speed.

Overall, WAVE provides a computationally efficient approach for broad engine studies, while Forte offers greater spatial detail but with increased sensitivity to model configuration. **Without experimental validation or known and validated setup parameters appropriate to each fuel and combustion model, neither approach can be considered definitively more accurate.** Their predictions should therefore be interpreted in relation to the assumptions, model parameters and limitations of each simulation approach. **Further work should implement the proposed Forte mesh-refinement strategy presented in Appendix A and validate the resulting model setup against experimental engine data, providing a stronger basis for assessing the accuracy of the WAVE and Forte predictions.**

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Appendix A

mesh refinement areas

Spark

Outer Spark Refinement

A spherical refinement region with a 5 mm radius is suggested to be centred on the spark location to provide increased mesh resolution within the region surrounding ignition. Liu et al. (2018) used ANSYS Forte for three-dimensional spark-ignition engine modelling and reported refining the spark-plug location using a 10 mm ball with mesh size of one quarter of that of the global refinement. This was interpreted as a 10 mm diameter region, giving the 5 mm radius adopted in the present study. A local cell size of one-quarter of the global mesh size is suggested based on Aktas (2022), who similarly used ANSYS Forte and specified local refinement around the spark-plug location at 1/4 of the global mesh size. The combination of these studies therefore supports an outer spark refinement of 5 mm radius with a mesh size of 1/4 of the global mesh size.

Inner Spark Refinement

A second spherical refinement region with a 0.6 mm radius is suggested to be centred on the spark location to provide additional resolution immediately surrounding the ignition point. Wang et al. (2019) used a 0.6 mm radius spherical embedding with a cell size of 0.125 mm to resolve the spark region. Relative to the 1 mm base mesh used in the study, this represents a refinement ratio of 1/8. This refinement ratio is also supported by Guo et al. (2021), who applied a 0.25 mm mesh around the spark-plug gap from a 2 mm base mesh, again corresponding to a refinement ratio of 1/8. Janas (2017) additionally employed local spark-plug cell sizes of 0.125–0.25 mm, further supporting the requirement for increased mesh resolution immediately surrounding the spark. The reviewed literature therefore supports an inner spark refinement of 0.6 mm radius with a mesh size of 1/8 of the global mesh size.

Summary

Based on the reviewed literature, two concentric spherical refinement regions are suggested at the spark location. The outer region has a radius of 5 mm and a mesh size of 1/4 of the global mesh size, while the inner region has a radius of 0.6 mm and a mesh size of 1/8 of the global mesh size. The outer region provides increased resolution surrounding the ignition location, while the smaller inner region provides the finer resolution required immediately around the spark. This approach concentrates mesh refinement where it is required while avoiding the computational cost of applying the same resolution throughout the combustion chamber.

Walls

Local mesh refinement is suggested to be applied to the cylinder head, cylinder liner and piston crown, using two cell layers at one-quarter of the global mesh size. This selection is supported by Aktas (2022) and Aktas and Karyeyen (2022), who both applied refinement to “all wall definitions” in ANSYS Forte using a 1/4 global size fraction and two cell layers. Independent support is provided by Ferretti (2018), who applied Scale 2 boundary refinement with two cell layers to the cylinder head, liner and piston. De Candia (2019) similarly applied Scale 2 boundary refinement to the head, liner and piston, although the number of cell layers was not specified. The relationship between the CONVERGE embedding scale and base cell size is defined by Equation X, as described by Testa (2018). From this relationship, a Scale 2 refinement corresponds to a cell dimension of one-quarter of the base mesh size. Therefore, the Scale 2 wall-refinement strategies employed by Ferretti (2018)

and De Candia (2019) provide independent support for the 1/4 global mesh refinement suggested for the proposed approach. Collectively, these studies support applying two cell layers at 1/4 of the global mesh size to the cylinder head, cylinder liner and piston surfaces.

Equation 7: CONVERGE Mesh Embedding Scale Relationship (Testa (2018))

$$dx_{embed} = \frac{dx_{base}}{2^{embed_scale}}$$