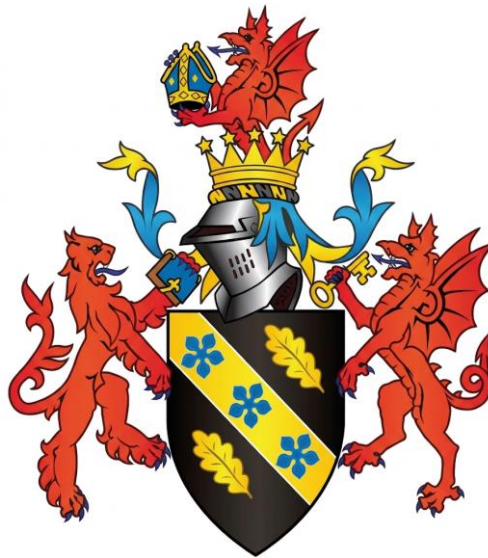




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Student name: Anandu Unnikrishnan Nair

Student ID number: 2411630

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Abstract

This study investigates the combustion behaviour and performance of a 1.6 L Ford Sigma spark-ignition engine using both one-dimensional and three-dimensional simulation tools. Ricardo WAVE was used to develop a baseline single-cylinder model, which was validated using available experimental engine data for pressure and temperature trends.

Following this, ANSYS Forte was used to carry out detailed combustion simulations using a 1/8th sector model of the engine cylinder. Three fuels were analysed: n-heptane (as a gasoline surrogate), methane, and hydrogen. The results obtained from the simulations show clear differences in combustion characteristics between the fuels.

Hydrogen demonstrated very fast combustion with high pressure and temperature peaks, indicating strong potential for high efficiency, but also showing increased NO_x formation. Methane produced slower and cooler combustion, which is beneficial for emissions but may reduce performance.

The study highlights the importance of combining 1D and 3D simulation approaches to better understand combustion behaviour. The results suggest that hydrogen and methane can be considered as potential low-carbon fuels for future internal combustion engines beyond 2035. Additionally, the study evaluates emission characteristics, particularly NO_x formation, highlighting key trade-offs between efficiency and environmental impact for alternative fuels.

1. Introduction

The global shift towards low-emission mobility is accelerating due to increasing concerns over climate change, energy security, and advancements in sustainable technologies. In the United Kingdom and European Union, legislation aims to phase out the sale of new petrol and diesel vehicles by 2035. However, internal combustion engines (ICEs) are expected to remain relevant in specific sectors such as lightweight vehicles and high-utilisation applications, where limitations in cost, infrastructure, and packaging restrict full electrification.

Synthetic and alternative fuels offer a practical solution to lower carbon emissions in this situation while preserving current engine technologies. Renewable energy sources can be used to produce fuels like hydrogen and methane, allowing for almost zero lifecycle emissions. In comparison to traditional petrol, methane delivers cleaner burning with lower particulate emissions, while hydrogen, in particular, offers the benefit of carbon-free combustion and fast flame speed.

This study uses the Ford Sigma 1.6L engine as a reference to examine how hydrogen and methane burn in a spark-ignition engine. A mixed simulation technique is used, with ANSYS Forte for three-dimensional combustion modelling and Ricardo WAVE for one-dimensional performance analysis. This allows for both thorough in-cylinder investigation and system-level validation.

The primary goal is to assess how these fuels affect temperature, pressure development, and combustion phasing in relation to a baseline of petrol. Additionally, the study looks at how well they can retain engine performance while complying with future emissions laws. By using this method, the project hopes to aid in the creation of low-carbon fuel options for internal combustion engines of the future.

Internal combustion engines continue to be crucial in applications where battery utilisation is limited by energy density, cost, and infrastructure constraints, even with the quick advancement of electrification. As a result, increasing ICE efficiency with alternate fuels is a workable temporary fix. But every fuel has different combustion issues, especially when it comes to temperature management, combustion stability, and emissions control.

2. Literature Review

In order to comprehend and forecast the performance and emissions of spark-ignition (SI) engines, combustion modelling is essential. Two-zone combustion models are one of the most popular methods because they strike a balance between physical accuracy and computational economy. Compared to more straightforward single-zone models, these models, which split the combustion chamber into burnt and unburned areas, enable more precise predictions of in-cylinder pressure, temperature, and emissions.

The Wiebe function, which shows the percentage of fuel burnt as a function of crank angle, is frequently used to characterise the rate of heat escape in such machines. This technique makes it possible to estimate combustion phasing factors like CA50, which are essential for engine efficiency and performance. Two-zone models are appropriate for one-dimensional simulation tools like Ricardo WAVE because prior research has demonstrated that they offer accurate forecasts of pressure development and combustion time under various operating situations.

Methane and hydrogen have drawn a lot of interest as possible low-carbon fuel substitutes for traditional petrol. Wide flammability limits, rapid flame propagation, and a high lower heating value are only a few benefits of hydrogen. These characteristics help to increase thermal efficiency and allow stable combustion in lean conditions. However, high in-cylinder temperatures linked to hydrogen combustion encourage the production of nitrogen oxides (NO_x). In order to reduce emissions, control methods like water injection or exhaust gas recirculation (EGR) are frequently needed.

However, because of its high hydrogen-to-carbon ratio, methane offers a cleaner burning process. Compared to petrol, it emits less particulate matter and less carbon dioxide. Furthermore, methane can operate at greater compression ratios, which can increase engine efficiency, due to its considerable resistance to knock. Its comparatively low flame speed, however, may cause incomplete oxidation and delayed burning, which could result in emissions of unburned methane. This emphasises how crucial it is to optimise after-treatment systems and ignition timing when utilising methane as a fuel.

Local mixture conditions and combustion temperature are the main factors influencing the

production of emissions in SI engines. Temperature-dependent processes, especially at high flame temperatures, produce nitrogen oxides. Incomplete combustion usually results in the production of carbon monoxide and unburned hydrocarbons, particularly in areas rich in fuel or close to the walls of combustion chambers. Fuel characteristics have a big impact on these processes; methane lowers NO_x but may increase emissions from unburned fuel, whereas hydrogen lowers carbon-based emissions while raising NO_x.

Simulation programs like ANSYS Forte and Ricardo WAVE are frequently used to investigate these combustion events. Using simplified combustion models, Ricardo WAVE offers a one-dimensional modelling approach that forecasts overall engine performance metrics, such as pressure, temperature, and mean effective pressure. On the other hand, ANSYS Forte simulates intricate in-cylinder processes like turbulence, flame propagation, and chemical reactions using three-dimensional computational fluid dynamics (CFD). Using a sector model lowers computational costs while providing an accurate description of combustion activity.

Comparative studies of hydrogen, methane, and gasoline highlight clear differences in combustion characteristics. Hydrogen typically exhibits rapid combustion and high efficiency due to its fast flame speed, but this can result in increased NO_x formation. Methane provides more stable and cleaner combustion with lower peak temperatures, although at the expense of slower burn rates. Gasoline remains a useful baseline for comparison due to its well-established combustion behaviour and widespread use in current engines.

All things considered, research suggests that methane and hydrogen have great promise as substitute fuels for upcoming SI engines. While methane offers a useful transition fuel with reduced emissions and compatibility with current infrastructure, hydrogen offers high efficiency and zero carbon emissions at the time of use. However, each fuel has unique emissions and combustion control issues that need to be resolved using sophisticated modelling and engine optimisation techniques.

Previous studies have also highlighted the importance of combining modelling approaches to improve prediction accuracy. While one-dimensional models provide rapid performance estimation, they lack the ability to capture localised combustion phenomena such as flame propagation and turbulence interactions. In contrast, three-dimensional CFD simulations offer detailed insight into in-cylinder processes but require higher computational resources. Therefore, integrating both approaches provides a balanced methodology for analysing alternative fuel performance.

3. Methodology

3.1 Engine Configuration

Due to the availability of experimental data and its significance as a contemporary SI engine, the Ford 1.6L four-cylinder spark-ignition engine was chosen. Production cars like the Ford Fiesta and Focus frequently employ this engine. The manufacturer's standards were used to specify important geometric parameters, such as bore, stroke, and compression ratio.

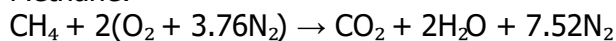
Metric	Value
Bore	79 mm
Stroke	81.4 mm
Compression ratio	11.1
Max power @ RPM	123 hp @ 6,000 rpm
Max torque @ RPM	118 lb-ft @ 4,100 rpm
Cylinder head	Aluminum
Cylinder block	Aluminum
Camshaft drive	Belt

Figure 1- Ford 1.6 L Sigma Engine specifications

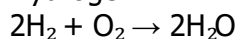
3.2 Stoichiometric and Combustion Chemistry

Correct air-fuel ratio specification is necessary for accurate combustion modelling. The necessary air-fuel mixes for each fuel were determined using stoichiometric combustion equations. It was supposed that the air was composed of 79% nitrogen and 21% oxygen.

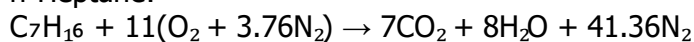
Methane:



Hydrogen:



n-Heptane:



The corresponding air-fuel ratios were approximately 15.1:1 for n-heptane, 17.2:1 for methane, and 34.3:1 for hydrogen

3.3 Ricardo WAVE Model

To simulate engine performance, a one-dimensional single-cylinder model was created in Ricardo WAVE.

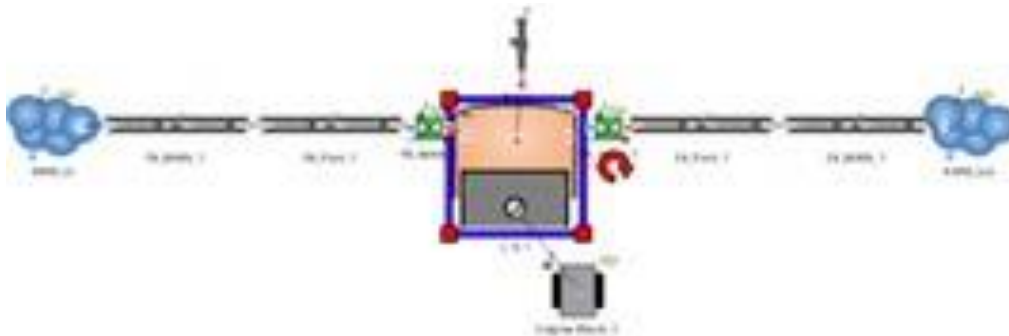


Figure 2- Single Cylinder Wave Model Replicating a Ford Sigma 1.cL Cylinder

The model included intake and exhaust systems, valves, and a central combustion chamber.

Ports	
Valve intakes count	1
1: IN_Valve_1	
Valve exhausts count	1
1: EX_Valve_1	

Geometry	
Bore	79 mm
Crank Stroke	81.4 mm
Clearance height	2 mm
Head area multiplier	1.6
Piston area multiplier	1
☐ Sinusoidal Movement	
Connecting rod length	137 mm
Pin offset	0 mm
Compression ratio	11.16

Boundary Conditions	
Piston	525 K
Liner	500 K
Head	550 K
Intake valve	400 K
Exhaust valve	450 K

Initial Conditions	
Swirl ratio	0
Fluid state	Intake_Initial

Figure 3 -Geometric inputs for the single cylinder replicating a Ford Sigma Engine

Simulations were carried out under steady-state, wide-open throttle conditions across engine speeds from 1000 to 6000 rpm. A Wiebe function was used to define heat release in a two-zone combustion model. The model was verified using data from experimental engines.

3.4 ANSYS Forte Model

ANSYS Forte was used to simulate combustion in three dimensions. To lower computational costs without sacrificing combustion accuracy, a 1/8th sector model was employed.

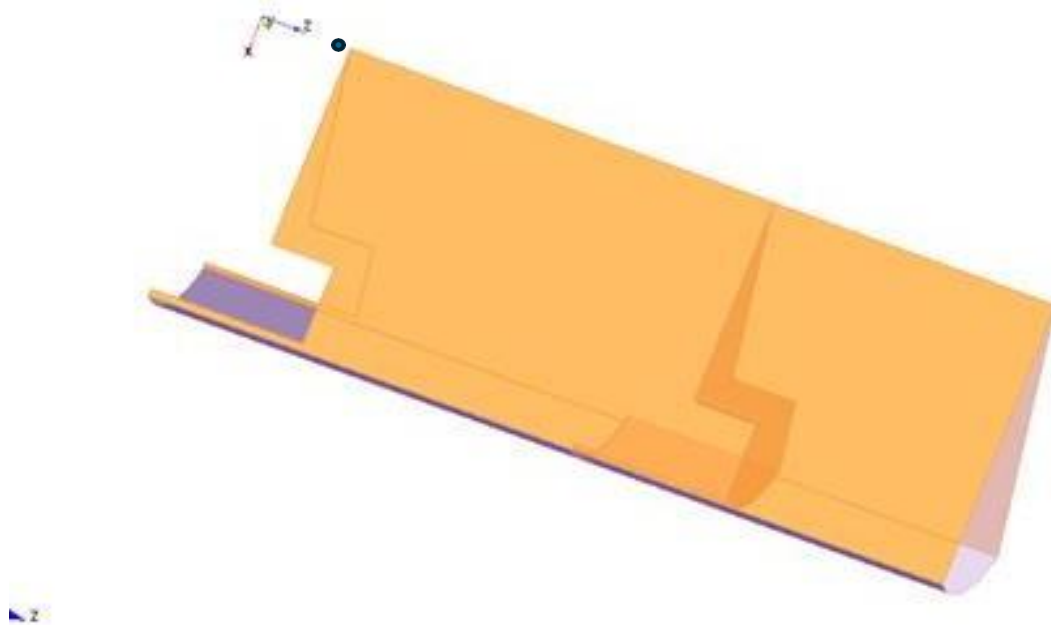


Figure 4- Ansys Forte Sector Mesh Model Replicating the Ford Sigma 1.cL engine, with a black dot to indicate the propagation centre of the air-fuel mixture and spark igniter

To capture flame propagation, the mesh was fine-tuned close to the spark plug and valve areas. The SAGE solver was used to simulate combustion, while the RNG k- ϵ model was used to model turbulence.

3.5 Fuel Definition

Three fuels were analysed: hydrogen, methane, and n-heptane. Fuel compositions were defined under stoichiometric conditions.

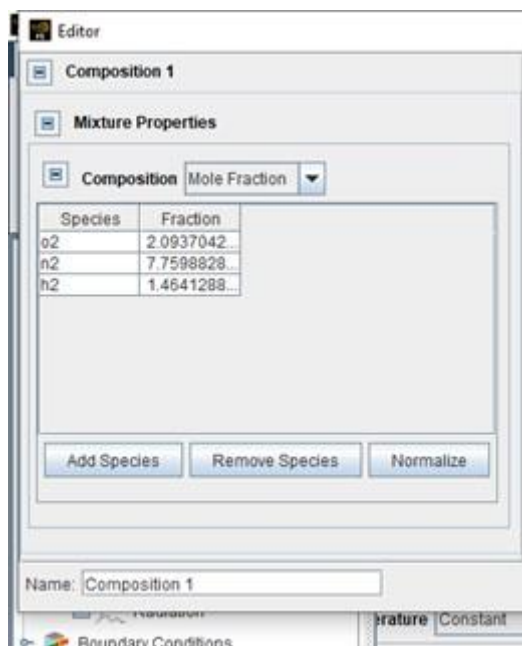


Figure 5 - Hydrogen Fuel Composition in Ansys Forte

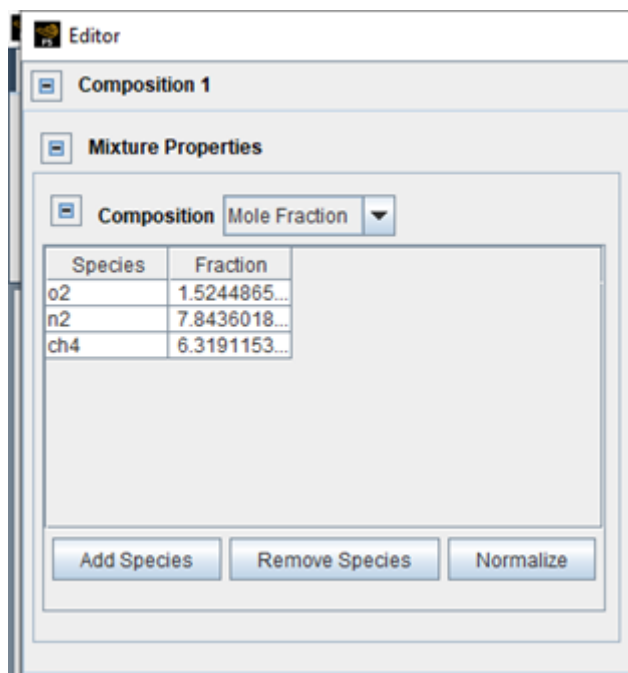


Figure 6- Methane Fuel Composition in Ansys Forte

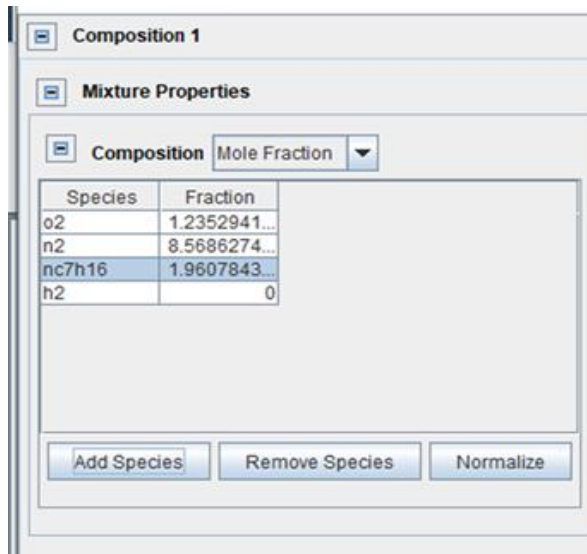


Figure 7 - N-Heptane Fuel Composition in Ansys Forte

These compositions ensured consistent comparison across all simulations.

3.6 Simulation Setup

To guarantee consistent outcomes, simulations were run over several engine cycles. Heat release rate, temperature, and in-cylinder pressure were important outputs. To assess combustion behaviour, the outcomes were contrasted with experimental data and WAVE forecasts.

The combined use of 1D and 3D simulation tools allows both global engine performance and detailed combustion characteristics to be analysed. This integrated approach improves the reliability of the results and reduces the need for extensive experimental testing.

4. Results and Discussion

4.1 WAVE Model Validation

By comparing simulation findings with existing experimental data, the baseline engine performance was validated using the Ricardo WAVE model. Over the engine speed range of 1000–6000 rpm, the projected trends for in-cylinder pressure and temperature demonstrated strong agreement.

Engine Speed (rpm)	P _{MAX_AV} (bar)	T _{MAX_AV} (K)
1000	47.6	2463.8
2000	62.3	2545.5
3000	67.4	2489.5
4000	69.1	2400.9
5000	66.0	2296.0
6000	62.8	2278.0

Table 1 Ford Sigma Engine Data for Maximum Pressure and Maximum Temperature

According to the findings, peak pressure rises with engine speed up to about 4000 rpm before significantly declining at higher speeds because of decreased combustion efficiency. Maximum temperature has a similar pattern, peaking at mid-range engine rpm. These patterns align with the typical behaviour of spark-ignition engines.

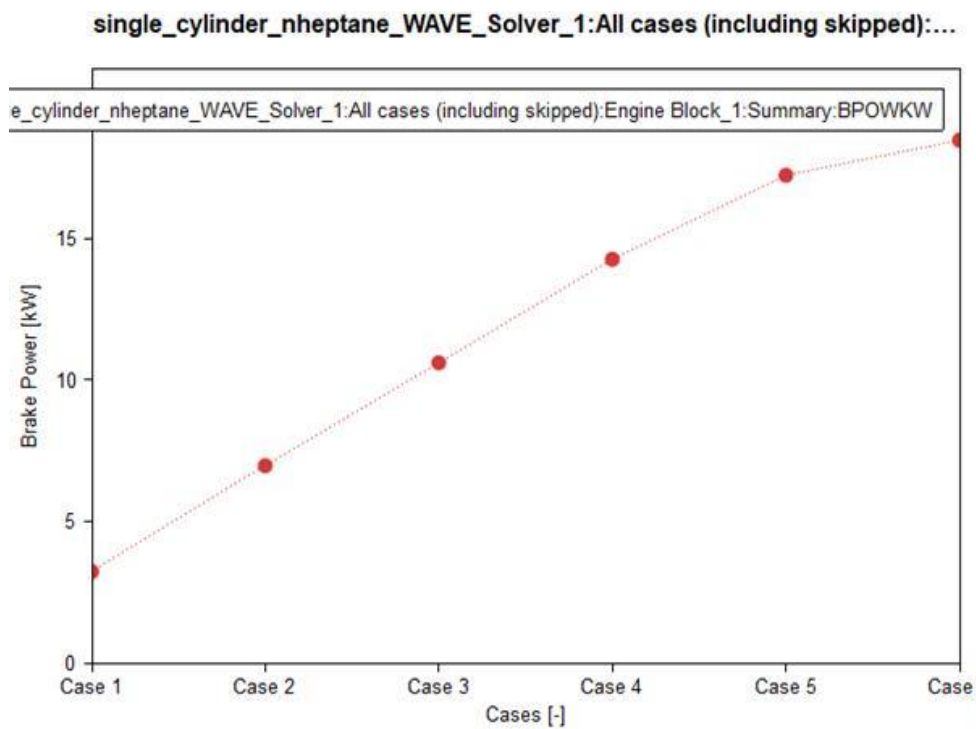


Figure 8 - Ford Wave Model brake power output

The WAVE model predicts that as engine speed increases, so does braking power. The model generates about 74 kW at 6000 rpm when scaled to a four-cylinder version. This results in an error of about 20–25%, which is acceptable for a one-dimensional simulation model since it is marginally less than the manufacturer value of 91.7 kW.

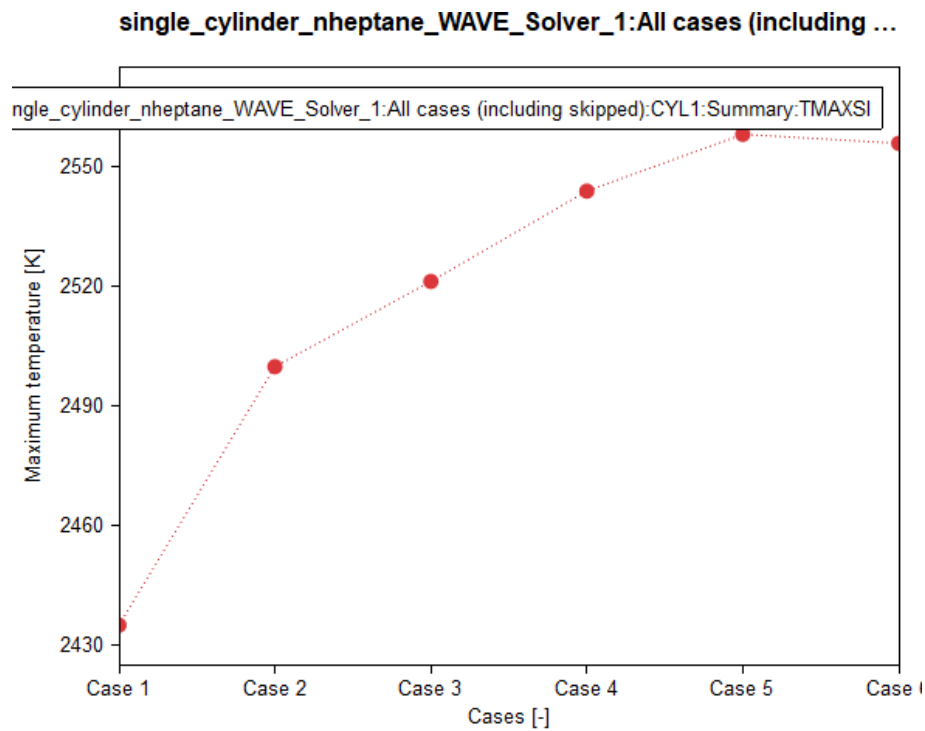


Figure 9 - Ford Single Cylinder Wave Model Graph showing Maximum Temperature (K)

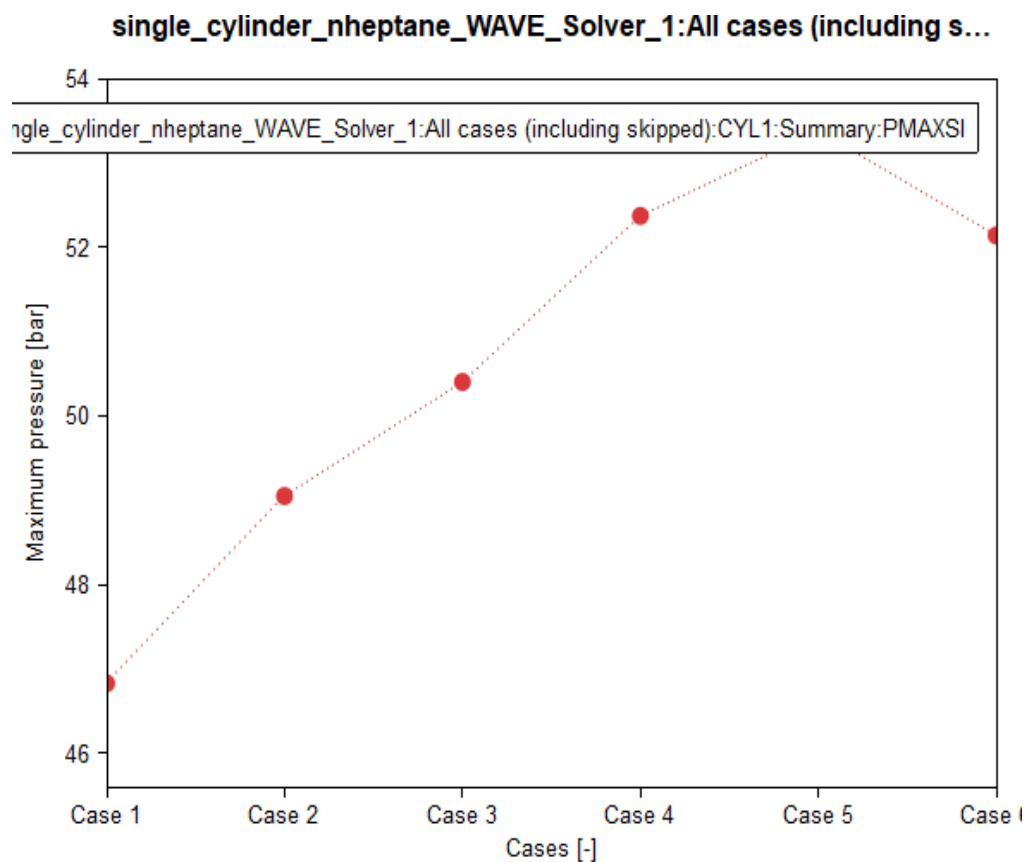


Figure 10 - Ford Single Cylinder Wave Model Graph showing Maximum Pressure (Bar)

All things considered, the WAVE model shows consistent agreement with experimental data, indicating that it is appropriate for baseline validation prior to using sophisticated combustion simulations.

4.2 ANSYS Forte Results (Fuel Comparison)

ANSYS Forte simulations were conducted to analyse combustion behaviour for methane, hydrogen, and n-heptane under identical engine conditions

- Methane (CH₄)

Methane combustion produced a maximum in-cylinder pressure of approximately 55.7 bar and a peak temperature of around 1850 K.

1	Cylinder mass	1
2	Engine Bore [cm]	13.97
3	Displacement Volume [cm ³]	2335.972
4	Comb. Chamber Volume at TDC [cm ³]	229.1103
5	Geometric Compression Ratio	11.19584
6	Gross Indicated Work [kJ]	2.61566
7	Gross Indicated Power [kW]	26.1566
8	IMEP [MPa]	1.11973
9	Total Fuel Mass [g]	1.89E-01
10	Fuel Mass per Engine Cycle [g]	1.89E-01
11	Fuel Lower Heating Value per g [kJ]	44
12	Gross ISFC per kW-h [g]	260.2514
13	ISFC (IVC->EVO) per kW-h [g]	289.6573
14	Combustion Efficiency	1
15	Thermal Efficiency	0.31438
16	Max Pressure [MPa]	5.5745
17	Max Temperature [K]	1850.428
18	Max Pressure Rise Rate per CA deg [MPa]	0.13372
19	Total Chemical Heat Release [J]	9046.033
20	Total Wall Heat Transfer Loss [J]	535.4082

Figure 11- Ansys Forte Simulation Results Using CH₄ as fuel

Although methane produces lower peak temperatures, its slower combustion may reduce overall engine efficiency if combustion duration extends beyond optimal crank angle timing.

- Hydrogen (H₂)

Hydrogen combustion resulted in significantly higher peak pressure and temperature, reaching approximately 155 bar and 3017 K.

2	Engine Bore [cm]	13.97
3	Displacement Volume [cm ³]	2335.972
4	Comb. Chamber Volume at TDC [cm ³]	229.1103
5	Geometric Compression Ratio	11.19584
6	Gross Indicated Work [kJ]	3.77291
7	Gross Indicated Power [kW]	37.72913
8	IMEP [MPa]	1.61514
9	Total Fuel Mass [g]	1.20E-01
10	Fuel Mass per Engine Cycle [g]	1.20E-01
11	Fuel Lower Heating Value per g [kJ]	120
12	Gross ISFC per kW-h [g]	114.0661
13	ISFC (IVC->EVO) per kW-h [g]	120.7749
14	Combustion Efficiency	0.87221
15	Thermal Efficiency	0.26301
16	Max Pressure [MPa]	15.49861
17	Max Temperature [K]	3017.129
18	Max Pressure Rise Rate per CA deg [MPa]	10.55827
19	Total Chemical Heat Release [J]	12512.17

Figure 12- Ansys Forte Simulation Results Using H₂ as fuel

The rapid combustion behaviour is caused by hydrogen's high diffusivity and flame speed. While this improves thermal efficiency, it also increases NO_x formation due to higher combustion temperatures and rapid heat release.

The significantly higher pressure rise rate observed in hydrogen combustion may also increase the risk of engine knock and mechanical stress, requiring careful control of ignition timing and mixture formation.

n-Heptane

n-Heptane combustion produced moderate pressure and temperature values, representing typical gasoline-like behaviour.

	A	B	C
1	Cylinder Index	1	
2	Engine Bore [cm]	13.97	
3	Displacement Volume [cm ³]	2335.972	
4	Comb. Chamber Volume at TDC [cm ³]	229.1101	
5	Geometric Compression Ratio	11.19585	
6	Gross Indicated Work [kJ]	2.18573	
7	Gross Indicated Power [kW]	21.85728	
8	IMEP [MPa]	0.93568	
9	Total Fuel Mass [g]	1.45E-01	
10	Fuel Mass per Engine Cycle [g]	1.45E-01	
11	Fuel Lower Heating Value per g [kJ]	44	
12	Gross ISFC per kW-h [g]	239.0654	
13	ISFC (IVC->EVO) per kW-h [g]	257.8972	
14	Combustion Efficiency	1	
15	Thermal Efficiency	0.34224	
16	Max Pressure [MPa]	12.83059	
17	Max Temperature [K]	1975.794	
18	Max Pressure Rise Rate per CA deg [MPa]	5.29409	
19	Total Chemical Heat Release [J]	6541.095	
20	Total Wall Heat Transfer Loss [J]	2232.024	
21	Cooling Loss Ratio	3.41E-01	
22	Total Apparent Heat Release (Chemical - Heat Loss) [J]	4309.071	
23	Total Apparent Heat Release (From PV With Variable C)	4282.633	
24	Total Apparent Heat Release (From PV With Constant C)	3422.092	
25	Degree of Constant Volume (Apparent)	1.06171	
26	Degree of Constant Volume (Burning)	0.96288	
27	Degree of Constant Volume (Cooling)	0.94239	
28	CA ATDC @ 2% Heat Release [deg]	9.2341	
29	CA ATDC @ 10% Heat Release [deg]	12.4672	
30	CA ATDC @ 30% Heat Release [deg]	14.3928	
31	CA ATDC @ 50% Heat Release [deg]	18.6257	
32	CA ATDC @ 90% Heat Release [deg]	29.8253	
33	10%-90% Heat Release Duration [deg]	20.5912	
34	Soot @ EVO [g]	4.42E-09	
35	Soot @ EVO per kg-fuel [g]	3.05E-05	

Figure 13 - Ansys Forte Simulation Results Using N-heptane as fuel

The results lie between methane and hydrogen, making n-heptane a suitable baseline for comparison

Model Comparison

Modelling techniques are the main cause of discrepancies between WAVE and Forte outcomes. While ANSYS Forte records intricate flow, turbulence, and chemical kinetics, the WAVE model assumes homogeneous conditions inside the cylinder. Forte therefore forecasts more localised combustion effects and stronger pressure peaks.

These results demonstrate that fuel selection involves a trade-off between efficiency and emissions. Hydrogen offers maximum efficiency potential, while methane provides improved emission characteristics and combustion stability.

4.3 Combustion Phasing Analysis

Combustion phasing analysis was used to evaluate burn duration and timing across different fuels.

Fuel Type	CA@2%	CA@10%	CA@50%	CA@G0%	Duration (10-G0%)
CH ₄	23° ATDC	32° ATDC	49° ATDC	64° ATDC	32°
H ₂	12° ATDC	12° ATDC	12° ATDC	12° ATDC	0°
C ₇ H ₁₆	9° ATDC	12° ATDC	18° ATDC	30° ATDC	20°

Table 2 Crank Angle Timing of Specific Heat Release Percentages (Combustion stages)

The findings indicate slower flame propagation because methane combustion takes place later and for a longer period of time. Hydrogen, on the other hand, burns almost instantly, releasing energy quickly and increasing pressure.

With combustion properties more akin to those of regular petrol, n-heptane behaves in an intermediate manner. These variations demonstrate how fuel characteristics affect combustion time, efficiency, and emissions production.

The variation in combustion phasing highlights the importance of ignition control strategies, particularly when adapting existing engines for alternative fuels.

5. Discussion

The findings unequivocally show that fuel characteristics have a major impact on engine performance, emissions production, and combustion behaviour. Because of its quick combustion and fast flame speed, which raises peak pressure and temperature, hydrogen has the highest efficiency potential. However, because of the increased NO_x generation caused by these high temperatures, controlling emissions is extremely difficult.

On the other hand, methane offers a more steady and regulated combustion process.

Although the observed lower peak temperatures reduced NO_x emissions, the slower flame speed results in longer combustion times, which, if not adjusted, may have a negative effect on engine efficiency.

From an engineering perspective, these findings highlight that alternative fuels cannot be directly applied to conventional engines without modification. Hydrogen requires advanced combustion control strategies such as exhaust gas recirculation (EGR), water injection, or lean-burn operation to manage temperature and emissions. Methane engines require optimisation of ignition timing and turbulence to improve combustion speed.

Overall, the study confirms that fuel-specific optimisation is essential for achieving a balance between efficiency, emissions, and combustion stability in future internal combustion engines.

6. Conclusion

This study investigated the combustion behaviour and performance of hydrogen, methane, and n-heptane in a spark-ignition engine using a combined simulation approach with Ricardo WAVE and ANSYS Forte. The use of both one-dimensional and three-dimensional modelling enabled comprehensive analysis, where WAVE provided reliable baseline validation of engine performance, and Forte delivered detailed insight into in-cylinder combustion processes.

The findings showed distinct variations in the fuels' combustion properties. With a far greater peak pressure and temperature and incredibly quick combustion, hydrogen showed great promise for increased thermal efficiency. However, because of the high combustion temperatures and quick heat release, this was followed by an increase in NO_x generation. Methane, on the other hand, burnt more slowly and coolly, which lowered NO_x emissions but may have compromised combustion efficiency because of the longer burn time. As a petrol surrogate, n-heptane produced intermediate results and was a trustworthy benchmark for verifying both simulation methods.

The results show that using alternative fuels requires striking a compromise between emissions and performance. Although hydrogen has the greatest potential for efficiency, controlling emissions and combustion intensity requires sophisticated management techniques. Methane, on the other hand, is a useful transitional fuel since it offers a more stable and clean combustion process.

All things considered, this study demonstrates that combining 1D and 3D simulation tools is a useful technique for assessing alternative fuels in internal combustion engines. To enable the practical application of low-carbon fuels in next-generation engine systems, future development should concentrate on fuel-specific engine optimisation, including ignition control, mixture preparation, and emission reduction techniques.

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8. Appendix

Forte Engine Data:

	A	B	C
6	Gross Indicated Work [kJ]	2.18573	
7	Gross Indicated Power [kW]	21.8573	
8	IMEP [MPa]	0.93568	
9	Total Fuel Mass [g]	1.45E-01	
10	Fuel Mass per Engine Cycle [g]	1.45E-01	
11	Fuel Lower Heating Value per g [kJ]	44	
12	Gross ISFC per kW-h [g]	239.065	
13	ISFC (IVC->EVO) per kW-h [g]	257.897	
14	Combustion Efficiency	1	
15	Thermal Efficiency	0.34224	
16	Max Pressure [MPa]	12.8306	
17	Max Temperature [K]	1975.79	
18	Max Pressure Rise Rate per CA deg [MPa]	5.29409	
19	Total Chemical Heat Release [J]	6541.09	
20	Total Wall Heat Transfer Loss [J]	2232.02	
21	Cooling Loss Ratio	3.41E-01	
22	Total Apparent Heat Release (Chemical - Heat Loss) [J]	4309.07	
23	Total Apparent Heat Release (From PV With Variable Gamma) [J]	4282.63	
24	Total Apparent Heat Release (From PV With Constant Gamma) [J]	3422.09	
25	Degree of Constant Volume (Apparent)	1.06171	
26	Degree of Constant Volume (Burning)	0.96288	
27	Degree of Constant Volume (Cooling)	0.94239	
28	CAATDC @ 2% Heat Release [deg]	9.2341	
29	CAATDC @ 10% Heat Release [deg]	12.4672	
30	CAATDC @ 30% Heat Release [deg]	14.3928	
31	CAATDC @ 50% Heat Release [deg]	18.6257	
32	CAATDC @ 90% Heat Release [deg]	29.8253	
33	10%-90% Heat Release Duration [deg]	20.5912	
34	Soot @ EVO [g]	4.42E-09	
35	Soot @ EVO per kg-fuel [g]	3.05E-05	
36	Soot @ EVO per kW-h [g]	7.28E-06	
37	Soot @ EVO in ppm	1.95E-06	
38	EINOx @ EVO [g]	4.15E-04	
39	EINOx @ EVO per kg-fuel [g]	2.86E+00	
40	EINOx @ EVO per kW-h [g]	6.84E-01	
41	EINOx @ EVO in ppm	4.73E+01	
42	CO @ EVO [g]	1.14E-08	
43	CO @ EVO per kg-fuel [g]	7.84E-05	
44	CO @ EVO per kW-h [g]	1.87E-05	
45	CO @ EVO in ppm	2.13E-03	

Figure 14 NHeptane Ansys Data

2	Engine Bore [cm]	13.97
3	Displacement Volume [cm3]	2335.97
4	Comb. Chamber Volume at TDC [cm3]	229.11
5	Geometric Compression Ratio	11.1958
6	Gross Indicated Work [kJ]	2.61566
7	Gross Indicated Power [kW]	26.1566
8	IMEP [MPa]	1.11973
9	Total Fuel Mass [g]	1.89E-01
10	Fuel Mass per Engine Cycle [g]	1.89E-01
11	Fuel Lower Heating Value per g [kJ]	44
12	Gross ISFC per kW-h [g]	260.251
13	ISFC (IVC->EVO) per kW-h [g]	289.657
14	Combustion Efficiency	1
15	Thermal Efficiency	0.31438
16	Max Pressure [MPa]	5.5745
17	Max Temperature [K]	1850.43
18	Max Pressure Rise Rate per CA deg [MPa]	0.13372
19	Total Chemical Heat Release [J]	9046.03
20	Total Wall Heat Transfer Loss [J]	535.408
21	Cooling Loss Ratio	5.92E-02
22	Total Apparent Heat Release (Chemical - Heat	8510.62
23	Total Apparent Heat Release (From PV With Va	9381.41
24	Total Apparent Heat Release (From PV With Cc	7427.61
25	Degree of Constant Volume (Apparent)	0.62047
26	Degree of Constant Volume (Burning)	0.61869
27	Degree of Constant Volume (Cooling)	0.57999
28	CAATDC @ 2% Heat Release [deg]	23.0204
29	CAATDC @ 10% Heat Release [deg]	32.02
30	CAATDC @ 30% Heat Release [deg]	42.028
31	CAATDC @ 50% Heat Release [deg]	49.012
32	CAATDC @ 90% Heat Release [deg]	64.024
33	10%-90% Heat Release Duration [deg]	32.004
34	Soot @ EVO [g]	1.11E-11
35	Soot @ EVO per kg-fuel [g]	5.90E-08
36	Soot @ EVO per kW-h [g]	1.53E-08
37	Soot @ EVO in ppm	5.03E-09
38	EINOx @ EVO [g]	3.75E-04
39	EINOx @ EVO per kg-fuel [g]	1.99E+00
40	EINOx @ EVO per kW-h [g]	5.17E-01
41	EINOx @ EVO in ppm	4.38E+01
42	CO @ EVO [g]	4.09E-05
43	CO @ EVO per kg-fuel [g]	2.16E-01
44	CO @ EVO per kW-h [g]	5.63E-02
45	CO @ EVO in ppm	7.82E+00
46	UHC @ EVO [g]	8.07E-03

Figure 15- Methane Ansys Data

S24			
	A	B	C
4	Comb. Chamber Volume at TDC [cm3]	229.11	
5	Geometric Compression Ratio	11.1958	
6	Gross Indicated Work [kJ]	3.77291	
7	Gross Indicated Power [kW]	37.7291	
8	IMEP [MPa]	1.61514	
9	Total Fuel Mass [g]	1.20E-01	
10	Fuel Mass per Engine Cycle [g]	1.20E-01	
11	Fuel Lower Heating Value per g [kJ]	120	
12	Gross ISFC per kW-h [g]	114.066	
13	ISFC (IVC->EVO) per kW-h [g]	120.775	
14	Combustion Efficiency	0.87221	
15	Thermal Efficiency	0.26301	
16	Max Pressure [MPa]	15.4986	
17	Max Temperature [K]	3017.13	
18	Max Pressure Rise Rate per CA deg [MPa]	10.5583	
19	Total Chemical Heat Release [J]	12512.2	
20	Total Wall Heat Transfer Loss [J]	4195.88	
21	Cooling Loss Ratio	3.35E-01	
22	Total Apparent Heat Release (Chemical - Heat Loss) [J]	8316.29	
23	Total Apparent Heat Release (From PV With Variable Gam	14854.9	
24	Total Apparent Heat Release (From PV With Constant Gam	6647.61	
25	Degree of Constant Volume (Apparent)	1.03151	
26	Degree of Constant Volume (Burning)	0.95653	
27	Degree of Constant Volume (Cooling)	0.86622	
28	CA ATDC @ 2% Heat Release [deg]	12.0044	
29	CA ATDC @ 10% Heat Release [deg]	12.0044	
30	CA ATDC @ 30% Heat Release [deg]	12.0044	
31	CA ATDC @ 50% Heat Release [deg]	12.0044	
32	CA ATDC @ 90% Heat Release [deg]	12.0044	
33	10%-90% Heat Release Duration [deg]	0	
34	Soot @ EVO [g]	7.11E-24	
35	Soot @ EVO per kg-fuel [g]	5.95E-20	
36	Soot @ EVO per kW-h [g]	6.79E-21	
37	Soot @ EVO in ppm	3.73E-21	
38	EINOx @ EVO [g]	1.74E-02	
39	EINOx @ EVO per kg-fuel [g]	1.45E+02	
40	EINOx @ EVO per kW-h [g]	1.66E+01	
41	EINOx @ EVO in ppm	2.36E+03	
42	CO @ EVO [g]	5.19E-11	
43	CO @ EVO per kg-fuel [g]	4.34E-07	
44	CO @ EVO per kW-h [g]	4.95E-08	
45	CO @ EVO in ppm	1.16E-05	
46	UHC @ EVO [g]	7.50E-18	
47	UHC @ EVO per kg-fuel [g]	6.28E-14	
48	UHC @ EVO per kW-h [g]	7.16E-15	

Figure 16 - Hydrogen Ansys Data

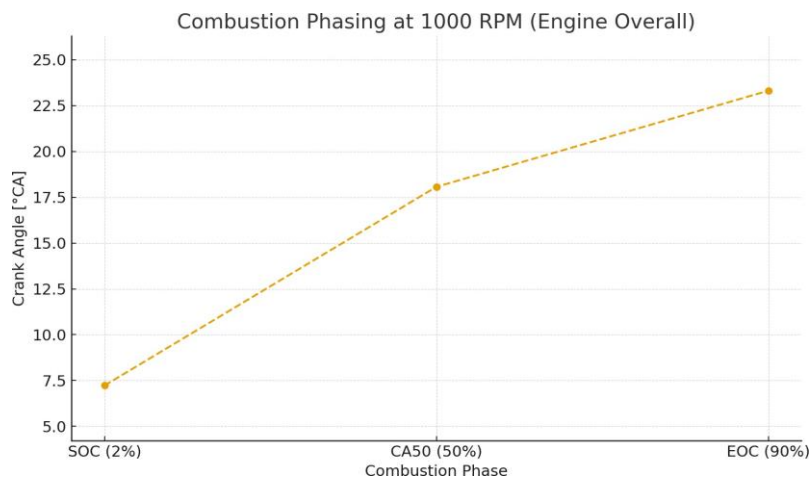


Figure-17 Ford Real Engine Data Combustion Phasing @1000 RPM

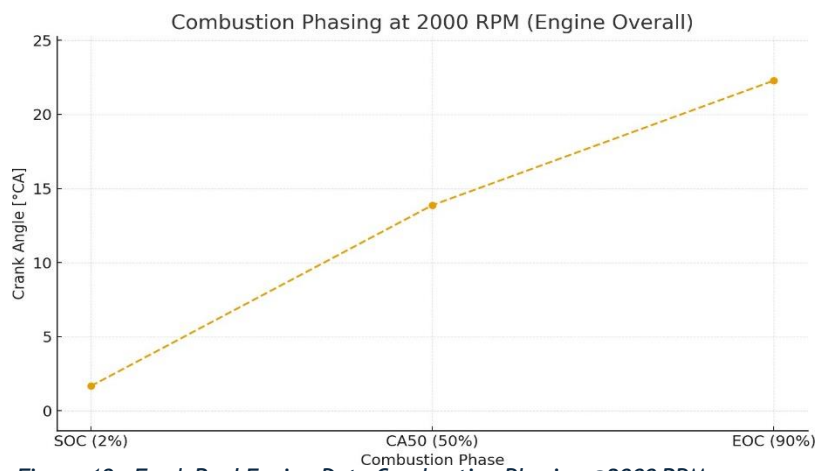


Figure 18- Ford Real Engine Data Combustion Phasing @2000 RPM

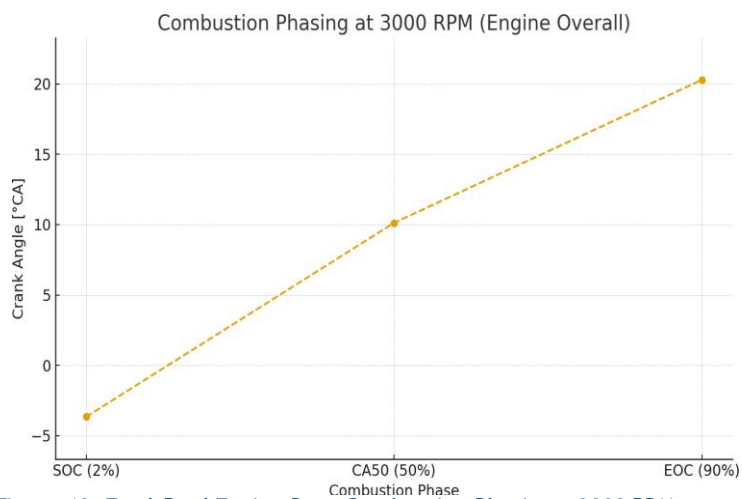


Figure 19- Ford Real Engine Data Combustion Phasing @3000 RPM

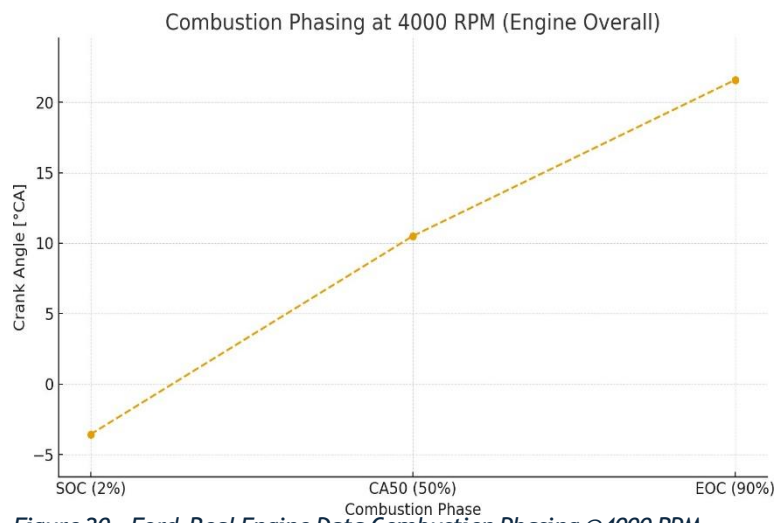


Figure 20- Ford Real Engine Data Combustion Phasing @4000 RPM

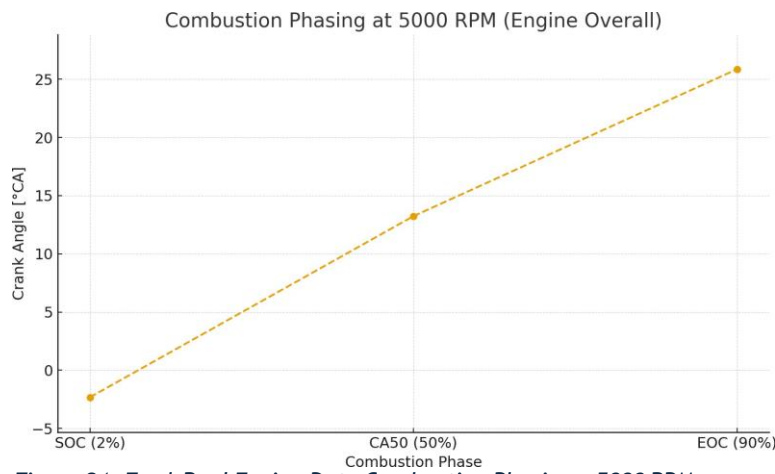


Figure 21- Ford Real Engine Data Combustion Phasing @5000 RPM

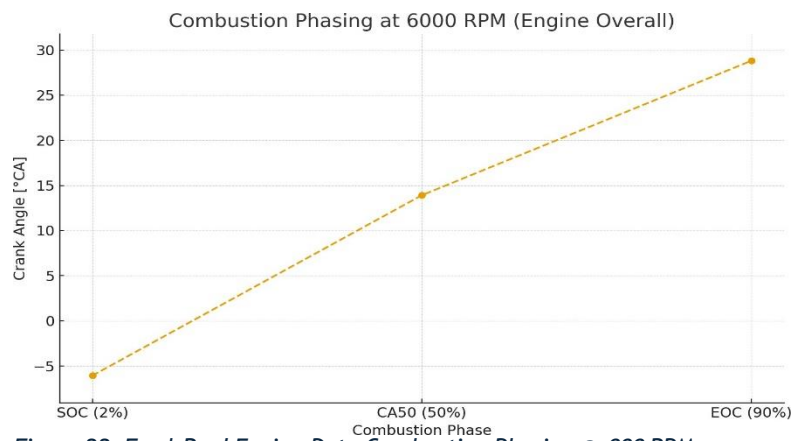


Figure 22- Ford Real Engine Data Combustion Phasing @6000 RPM

