

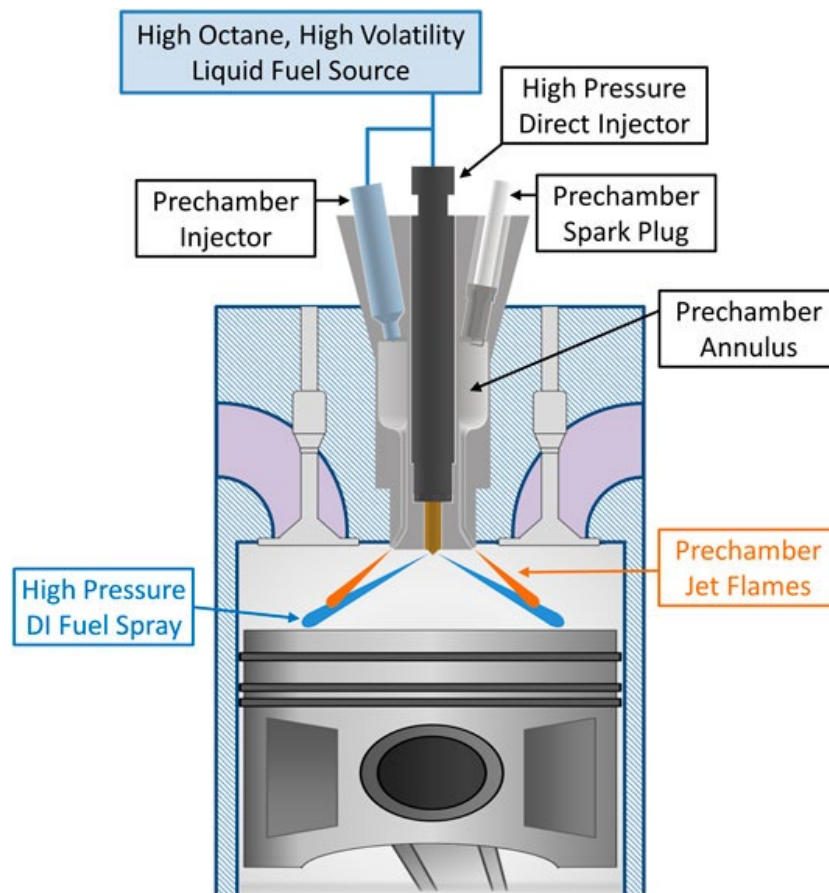
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Ford GDI Prechamber Retrofit Combustion Analysis

This Dissertation 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.

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Date: 07.09.2025

Abstract

The demand for higher efficiency and reduced emissions in spark-ignition engines has renewed interest in advanced ignition technologies, with turbulent jet ignition (TJI) emerging as a promising pathway. This study investigates the feasibility and performance of a pre-chamber retrofit applied to the Ford 1.0 L Fox EcoBoost engine, modelled in Ricardo WAVE. A two-zone combustion model was developed and validated against experimental pressure data at 1000, 3000, and 6000 rpm. Validation showed strong overall agreement in compression and expansion phases, with peak pressures predicted within a few bar of experimental values, though the model tended to overpredict burn sharpness and underrepresent the pre-peak shoulder, highlighting the need for more advanced multi-zone approaches.

Integration of a pre-chamber ignition system produced notable performance effects. Simulations predicted an approximate 15 bar rise in peak pressure compared with baseline spark ignition, reflecting faster and more robust combustion. At higher engine speeds (>2500 rpm), air–fuel ratios, burn durations, and IMEP trends aligned with theoretical expectations, supporting the concept’s validity. However, at low speeds, fuelling anomalies and unrealistic volumetric efficiencies ($>200\%$) revealed shortcomings in the simplified jet and fuelling sub-models, limiting quantitative accuracy.

The findings confirm the potential of TJI to extend lean-burn operation, reduce pumping and heat transfer losses, and improve thermal efficiency, with literature indicating peak indicated efficiencies above 50% under ultra-lean conditions. Importantly, TJI also provides a pathway for zero-carbon fuels such as ammonia–hydrogen blends, addressing ignition barriers while reducing NO_x emissions. Future improvements in model fidelity, particularly in turbulence–jet interaction and fuelling strategies, will be essential to fully quantify efficiency and emissions benefits.

Overall, this work demonstrates that Ricardo WAVE can serve as a robust tool for evaluating pre-chamber ignition concepts, and that TJI has the potential to significantly enhance the efficiency and sustainability of gasoline engines as part of the transition toward net-zero internal combustion technology.

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List of symbols / Nomenclature

[/] are the units for the corresponding symbol unless explicitly stated otherwise

A – Arrhenius pre-exponential factor [s^{-1}]
a – Wiebe function parameter (burn rate shaping) [–]
 C_p – Specific heat at constant pressure [$J\ kg^{-1}\ K^{-1}$]
 C_v – Specific heat at constant volume [$J\ kg^{-1}\ K^{-1}$]
 E_a – Activation energy [$J\ mol^{-1}$]
k – Rate constant (Arrhenius) [s^{-1}]
m – Wiebe function parameter (shape factor) [–]
 m_f – Mass of fuel [kg]
 m_a – Mass of air [kg]
p – Cylinder pressure [bar or Pa]
Q – Heat release [J]
R – Universal gas constant (= $8.314\ J\ mol^{-1}\ K^{-1}$)
T – Temperature [K]
V – Cylinder volume [m^3]
 γ – Ratio of specific heats (C_p/C_v) [–]
 ϕ – Equivalence ratio [–]
 λ – Relative air–fuel ratio (inverse of ϕ) [–]
 θ – Crank angle [$^{\circ}CA$]
 η_{th} – Thermal efficiency [–]
 η_b – Brake thermal efficiency [–]
 η_i – Indicated thermal efficiency [–]

Subscripts

a – Air
b – Brake (e.g. brake power, brake torque)
c – Compression
e – Expansion / exhaust
f – Fuel
i – Indicated
in – Intake
out – Outlet
Pc – Prechamber
1 – Main Combustion Cylinder
4 – Prechamber

Acronyms

ABDC – After Bottom Dead Centre
ATDC – After Top Dead Centre
AFR – Air–Fuel Ratio
BBDC – Before Bottom Dead Centre
BDC – Bottom Dead Centre
BMEP – Brake Mean Effective Pressure
BTDC – Before Top Dead Centre
BTE – Brake Thermal Efficiency
BSFC – Brake Specific Fuel Consumption

CA – Crank Angle
CA10, CA50, CA90 – Crank angles corresponding to 10%, 50%, 90% mass fraction burned
CFD – Computational Fluid Dynamics
CO – Carbon Monoxide
CO₂ – Carbon Dioxide
COV – Coefficient of Variation
EGR – Exhaust Gas Recirculation
FMEP – Friction Mean Effective Pressure
HC – Hydrocarbons
HRR – Heat Release Rate
ICE – Internal Combustion Engine
IMEP – Indicated Mean Effective Pressure
ITE – Indicated Thermal Efficiency
MAPE – Mean Absolute Percentage Error
MBT – Minimum advance for Best Torque
NO_x – Nitrogen Oxides
NVH – Noise Vibration Harshness
PM – Particulate Matter
RON – Research Octane Number
SI – Spark Ignition
SCR – Selective Catalytic Reduction
TDC – Top Dead Centre
TJI – Turbulent Jet Ignition
TWC – Three-Way Catalyst
WAVE – *Ricardo WAVE* (engine simulation software)

1. Introduction

Climate change mitigation has driven an urgent search for higher efficiency, lower emission alternatives to conventional spark-ignition (SI) combustion engines. While electrification is rapidly expanding, improving the performance of gasoline engines remains highly relevant, particularly in downsized, turbocharged architectures widely used in passenger vehicles. Among advanced ignition technologies, prechamber turbulent jet ignition (TJI) has emerged as a promising retrofit solution for boosting thermal efficiency, reducing fuel consumption, and cutting emissions in gasoline engines without fundamentally altering the main fuel infrastructure (Mahmoud et al., 2017; Musculus et al., 2019).

Conventional SI combustion is limited by flame propagation speed, knock resistance, and the lean burn stability threshold. These constraints restrict efficiency improvements, particularly at high loads and under ultra lean conditions where stable ignition is difficult to achieve (Zhou et al., 2021). In contrast, turbulent jet ignition uses a small prechamber, replacing the spark plug, that is fuelled with a richer mixture and ignited separately. Combustion in the prechamber generates jets of hot, chemically active gases which penetrate into the main chamber, rapidly initiating distributed ignition throughout the lean main charge (Gold et al., 2016; Morucci et al., 2018). This process provides a stronger, spatially distributed ignition source than a conventional spark, significantly extending lean burn capability, increasing efficiency, and reducing NO_x formation through cooler combustion temperatures.

Experimental studies with gasoline fuelled engines have shown that TJI can enable stable combustion at equivalence ratios $\lambda > 2$ with indicated thermal efficiencies of 42–45%, compared with ~36–38% typical for conventional spark ignition (Mahmoud et al., 2017). Prechamber systems have also demonstrated 20–25% reductions in specific fuel consumption under lean operation (Zhou et al., 2021). Active prechamber configurations, in particular, can suppress knock and allow higher compression ratios, enabling reported indicated thermal efficiencies approaching 47% in ultra lean gasoline operation (Ebrahimi et al., 2022). These benefits highlight prechamber ignition's potential as a cost-effective retrofit pathway for existing turbocharged gasoline engines such as the Ford Fox 1.0 L EcoBoost.

The present research employs Ricardo WAVE, a one dimensional (1D) engine simulation tool, to investigate the performance of the Ford 1.0 L EcoBoost three cylinder turbocharged engine under conventional spark ignition operation and with a prechamber retrofit. Ricardo WAVE models the engine as a network of ducts and components, solving unsteady gas dynamics to predict mass flow, pressure wave behaviour, combustion, and emissions. By inputting the detailed engine geometry (bore $\times$ stroke 71.9 mm $\times$ 82.0 mm, compression ratio 10.0:1,

turbocharging configuration), the software computes outputs such as cylinder pressure traces, airflow, torque, efficiency, and NO_x emissions. Crucially, the model will first be validated against available experimental data for the baseline Fox engine to ensure predictive accuracy. Previous work has shown that calibrated WAVE models can reproduce torque and pressure curves within ~1–3% of measured values.

Once validated, the model will be modified to incorporate a prechamber ignition system, represented in 1D as an auxiliary combustion zone linked to the main cylinder. By varying the fuelling strategy in the prechamber and the global air–fuel equivalence ratio, the study will compare combustion characteristics and efficiency between the stock SI engine and the prechamber retrofitted variant. Key metrics of interest include in cylinder pressure development, heat release rate, combustion duration, specific fuel consumption, thermal efficiency, and emissions trends. Particular attention will be given to the extension of lean burn capability and the trade-off between fuel consumption improvements and NO_x formation.

The findings are expected to clarify the relative benefits of a prechamber retrofit compared to conventional SI combustion in a downsized turbocharged gasoline engine. In particular, the simulation will provide insight into how TJI could improve efficiency and emissions performance without departing from existing gasoline infrastructure.

Overall, this Master’s project integrates advanced engine modelling with ignition system innovation. The remainder of this dissertation is structured as follows:

Chapter 2 reviews the literature on gasoline lean burn limitations, prechamber ignition principles, and prior TJI studies.

Chapter 3 details the Ricardo WAVE baseline model development and validation against the Ford 1.0 L EcoBoost engine.

Chapter 4 presents the prechamber retrofit model, simulating performance across a range of operating conditions, any issues with initial testing.

Chapter 5 discusses the comparative results.

Chapter 6 concludes the work, summarising contributions and outlining recommendations for future experimental and modelling research into prechamber gasoline combustion.

Through this study, we aim to demonstrate the feasibility and potential efficiency benefits of prechamber ignition as a retrofit strategy for turbocharged gasoline engines, supporting the transition to more sustainable and efficient internal combustion technologies.

2. Literature Review

Turbulent Jet Ignition is an advanced prechamber combustion system that replaces or augments the traditional spark plug in a gasoline engine. In a TJI-equipped engine, a small prechamber is integrated into the cylinder head and connects to the main combustion chamber via a set of small orifices. During operation, a portion of the air-fuel mixture is ignited in this prechamber, producing jets of hot, reacting gases that shoot through the orifices into the main chamber and ignite the main charge at multiple points. This creates a fast and distributed combustion of the fuel-air mixture, rather than a single flame kernel growing outward from one spark. Figure 1 schematically illustrates this concept in a cross-section of an engine cylinder (with an active prechamber system as described below).

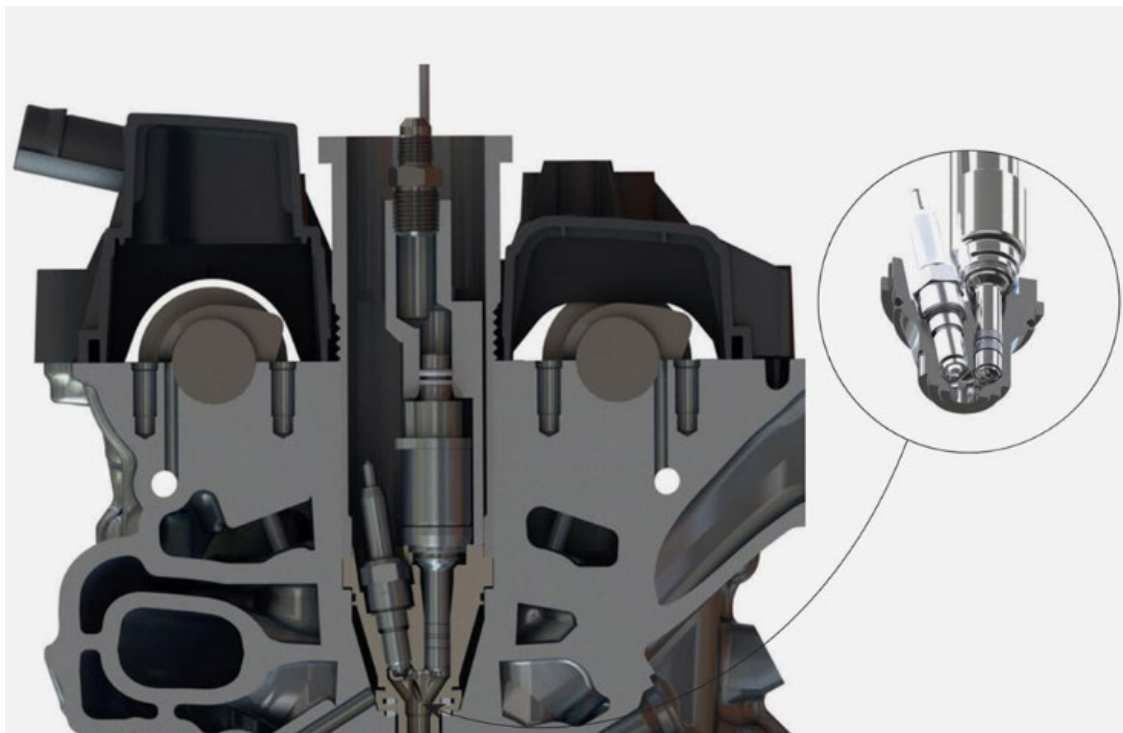


Figure 1 Cross-sectional diagram of an engine with an active pre-chamber (TJI) system

A small prechamber above the combustion chamber houses a spark plug (and in active systems, a direct injector). When the prechamber mixture ignites, jets of flame shoot down through orifices to ignite the main chamber uniformly, yielding rapid and stable combustion (Crosse, 2020).

In practice, there are two main types of TJI systems: active and passive prechambers. In an active TJI system, the prechamber has its own fuel supply (an injector that delivers a small amount of fuel, typically a rich mixture, to the prechamber) while the main chamber is supplied with most of the fuel as a lean mixture (either via port injection or direct injection in the cylinder). A spark in the prechamber ignites the rich charge there, and the resulting high energy

jets of partially combusted fuel shoot out through the nozzle holes to ignite the lean charge in the main chamber. In contrast, a passive TJI system has no dedicated injector in the prechamber. Instead, during the compression stroke a small portion of the homogeneous charge in the cylinder is pushed (squeezed) into the prechamber through the orifices. The spark plug (located inside the prechamber) ignites this mixture, and the flame jets then propagate out to ignite the remainder of the charge. Passive systems are simpler since they don't require a separate fuel circuit for the prechamber, but they offer less control over the prechamber mixture richness – it essentially depends on how the main charge stratifies and flows into the prechamber. Active systems allow the prechamber mixture to be optimized (often richer than stoichiometric for strong ignition, even if the main charge is very lean), at the cost of additional complexity (extra injector and calibration).

Regardless of type, the fundamental mechanism is the same: the prechamber ignition creates turbulent flame jets that penetrate the main chamber, igniting the main charge in multiple locations simultaneously. This results in a much faster and more complete burn compared to conventional spark ignition. In a traditional engine, a single spark kernel expands relatively slowly, and the flame front must travel across the chamber, which can be a lengthy process especially with lean mixtures or in large bore engines. With TJI, however, ignition occurs in a distributed fashion, essentially many flame kernels are introduced via the jets, so the whole mixture burns almost at once in a rapid, turbulence enhanced combustion. Experimental visualization has shown that TJI produces a highly turbulent ignition plume that lights the main charge uniformly and quickly.

2.1 Active vs. Passive Prechamber

In an active TJI system, only a small fraction of the total fuel (~1-5%) is used in the prechamber, where it is kept rich for ignition, while the remaining ~95% of fuel is delivered to the main chamber as a lean mixture. Mahle's prototype active Jet Ignition system, for instance, used about 5% of the fuel in a separate prechamber injector (creating a rich mixture around the spark) and the rest via port injection into the intake for a globally lean combustion in the cylinder. The spark initiates combustion of the rich prechamber contents, and the ensuing flame jets reliably ignite the ultra-lean charge in the main chamber (which a single spark could not easily ignite). By contrast, a *passive* system has no fuel injector in the prechamber: it relies on the main chamber mixture itself. During compression, some of that mixture is forced up into the prechamber; typically this mixture may be slightly richer than the global average (because fuel tends to stratify or the last portion of gas compressed into the small prechamber may have less air). The spark then ignites this small amount of mixture. The passive approach still yields

the jet ignition effect but gives the engine designer less direct control over the prechamber equivalence ratio. Both active and passive TJI ultimately create the same outcome of multiple ignition points and high turbulence in the combustion, enabling extremely fast flame propagation.

It is worth noting that prechamber ignition concepts have historical precedent. Prechamber or dual-chamber combustion is not entirely new. Honda's Compound Vortex Controlled Combustion (CVCC) engine in the 1970s used a small precombustion chamber to ignite a stratified charge in the main chamber. However, those systems were relatively primitive and often were used primarily for emissions control with carbureted or early fuel injection technology. Modern Turbulent Jet Ignition systems, made viable by precise fuel injection and computationally optimized design, are far more effective and easier to implement than those earlier prechamber concepts (Crosse, 2020). In motorsports, prechamber ignition made a resurgence around the 2010s: Formula 1 engines adopted TJI (often called Mahle Jet Ignition or similar) to allow very lean mixtures and high compression ratios under fuel-flow-limited regulations. Mahle's passive jet ignition was introduced in at least one major F1 team's engine by the mid-2010s, contributing to efficiency gains in the turbo-hybrid era. By 2017, Honda's F1 power unit (RA617H) had switched to a prechamber ignition system to speed up combustion, and it proved so effective that engineers could push compression ratios up to the regulatory limit of 18:1 without knock. (For context, typical production gasoline engines have compression ratios around 10:1–12:1, and even performance cars seldom exceed 14:1 on pump fuel.) The success of TJI in racing has spurred interest in using it for road car engines to improve fuel efficiency and emissions. Mahle and other companies have since been developing prechamber ignition systems for series-production engines, including both *passive* versions that could be drop-in replacements for spark plugs and *active* versions for ultimate efficiency gains.

2.2 Combustion Advantages of TJI

The turbulent jet ignition system provides several combustion and thermodynamic advantages over conventional spark ignition. These advantages directly translate to higher engine efficiency, better performance, and lower emissions.

2.2.1 Ultra-Fast Burn Rates

By creating multiple ignition sites and intense turbulence, TJI dramatically shortens the combustion duration. The flame does not have to travel across the chamber from a single point; instead, combustion starts almost simultaneously in many regions once the jets penetrate. This rapid burn means the bulk of the fuel release happens closer to the optimal timing (around top dead centre), which increases the work extracted and reduces heat loss to the walls. Experimental studies have quantified this benefit: for example, one study found that switching from a traditional spark plug to a prechamber ignition in a kerosene-fuelled engine reduced the combustion duration by about 20.8% (i.e. the burn was ~20% faster). The authors also observed that the faster burn suppressed knocking significantly, because there was less time for end-gas autoignition to occur. The same study reported that at various engine speeds, the indicated mean effective pressure (IMEP), essentially the cylinder pressure work output, correlated with torque – could be increased by 10% to 27% with TJI compared to conventional ignition. This implies more power/torque is obtained, likely due to being able to run optimal spark timing and higher compression without knock. Figure 1 also qualitatively illustrates how the flame jets create a faster, more uniform combustion. In general, by shortening burn duration, TJI ensures the combustion completes while the piston is near TDC, maximizing pressure during the expansion stroke rather than wasting it late in the cycle. Faster combustion also inherently mitigates knocking because the unburned end gas spends less time at high temperature. Any tendency for knock is outrun by the flame consuming the mixture quickly.

2.2.2 Lean Burn Capability (Ultra-Lean Combustion)

TJI enables stable ignition of much leaner fuel-air mixtures than a normal spark plug can handle. A lean mixture (excess air, $\phi < 1$ or $\lambda > 1$) is beneficial because it *reduces combustion temperature* and can eliminate throttling losses at part load (since you can run with more air and less fuel instead of choking the intake). However, lean mixtures are hard to ignite with a single spark – they have low flame propagation speeds and are prone to misfire or slow burning. The turbulent jets from a prechamber, by contrast, are energetic and carry active radical species, allowing even very lean mixtures in the main chamber to ignite reliably and burn quickly. With TJI, engines have been demonstrated at air-fuel ratios far beyond the conventional limits. For instance, research engines with prechambers have run stably at lambda (λ) 2.0–2.5 (more than twice the air needed for a stoichiometric mixture). One recent study reported maintaining reliable combustion at an equivalence ratio as low as ~0.37 ($\lambda \approx 2.7$), an extremely lean mixture, using a passive prechamber ignition, and achieved high efficiency in that regime (Niizato, 2024, as cited in Miao *et al.*, 2025). In contrast, a typical modern spark-ignition engine might

be limited to about $\lambda \approx 1.5$ before combustion becomes unstable. TJI extends the lean limit significantly, which is crucial for efficiency. Lean burn operation increases the ratio of specific heats (γ) due to excess air and reduces heat losses, and it allows the engine to run with a wider throttle opening (less pumping loss) for a given load. These effects together raise the engine's thermal efficiency.

2.2.3 Higher Thermal Efficiency

Because of the fast burn and lean operation, TJI-equipped engines can achieve substantially higher indicated thermal efficiency (ITE) and brake thermal efficiency (BTE) than conventional spark-ignition engines. Lean mixtures reduce peak temperatures (cutting heat transfer losses) and allow operation without a throttle in many conditions (eliminating pumping work loss), while the fast combustion ensures the cycle is closer to the ideal Otto cycle. Many research programs have reported record efficiency numbers with prechamber ignition. For example, a single cylinder test engine using a passive prechamber achieved a peak ITE of about 47.2% at $\lambda \approx 2.7$ (Niizato, 2024). Using an active prechamber with a slightly richer ultra-lean mixture ($\lambda \sim 2.4$), another experiment reached about 52.5% ITE (Hu *et al.*, 2023). These are remarkable figures – by comparison, a conventional gasoline engine's ITE might be $\sim 40\%$ at best (which corresponds to $\sim 35\%$ BTE when accounting for friction and parasitic losses). In fact, by combining TJI with other optimizations (high compression ratio, Miller cycle cam timing, etc.), researchers have demonstrated over 50% brake thermal efficiency on a gasoline engine. For instance, the Japanese SIP automotive project integrated a lean prechamber combustion system and achieved $\sim 48\%$ BTE, and with additional measures (like waste heat recovery) pushed to 51.5% BTE on a multi-cylinder engine. Similarly, a team at Tongji University reported a single-cylinder research engine reaching 47% BTE (51.3% ITE) at 2800 rpm using active TJI with $\lambda = 2.4$, compression ratio 16.8, and boosting. Such efficiency levels rival or exceed diesel engines, which have traditionally been around 40–45% BTE in light duty applications. This improvement directly translates into lower fuel consumption and lower CO₂ emissions per power output. In summary, TJI can improve efficiency by 10–15 percentage points in absolute terms over the typical spark-ignition engine – a huge leap in engine development.

2.2.4 Improved Knock Resistance

TJI's combustion process inherently resists engine knock (detonation). There are two reasons for this: (1) the end gas (the portion of mixture that hasn't yet burned when combustion is initiated) is much leaner and hence cooler, making it less prone to auto ignition; and (2) the

rapid combustion consumes the charge before knock has time to develop. Lean mixtures have higher ignition delay and are less reactive, so the phenomenon of knock is greatly suppressed. This improved knock tolerance means the engine can safely use higher compression ratios and more advanced ignition timing than it normally could on a given fuel. Mahle engineers noted that with Jet Ignition, “*higher compression ratios than normal can be used before detonation becomes a threat*”, precisely because the fast, distributed combustion avoids creating the hot spots and long burn duration that lead to knock. In practice, engines with TJI have been run at very high compression, for example, some F1 engines operate at the FIA-regulation limit of ~18:1 geometric compression, enabled by prechamber ignition and high-octane fuel. Even in more road-relevant research, compression ratios of 14:1 to 17:1 have been successfully used alongside TJI and lean burn, without knock issues, to reach those >50% efficiency figures. Essentially, TJI provides some of the benefits that high-octane fuels or knock additives would otherwise provide, by making the combustion process itself more knock proof. (Of course, using a high-octane fuel in addition to TJI can unlock even further compression or efficiency, but the point is TJI reduces the *dependency* on octane for knock avoidance.)

2.2.5 Drastically Lower NO_x Emissions

A striking benefit of TJI reported in the literature is the potential for ultra-low NO_x (oxides of nitrogen) emissions when operating in the lean regime. NO_x formation in engines is highly temperature dependent, it skyrockets at high flame temperatures (as in stoichiometric or rich combustion) but drops dramatically when combustion is cooler (lean burn), more information about this can be found in the appendix. Since TJI facilitates stable combustion at very lean mixtures, it can keep combustion temperatures low enough that NO_x production is minimal. Moreover, the distributed, rapid burn avoids the prolonged high-temperature pockets that a slower combustion might create. Mahle Powertrain demonstrated that at $\lambda \gtrsim 2$ (very lean), engine-out NO_x could be reduced by around 99% compared to a conventional stoichiometric operation. In other words, an engine running ultra-lean with TJI can have near-zero NO_x emissions straight out of the cylinder. This is important because NO_x is a regulated pollutant (linked to smog and respiratory issues), and achieving such low NO_x with combustion alone could potentially obviate the need for expensive aftertreatment systems like SCR (Selective Catalytic Reduction) in some use cases. It should be noted, however, that if the engine is not run extremely lean, say only moderately lean ($\lambda \sim 1.5$) to maximize efficiency, some NO_x will still form and would need cleanup. But TJI provides the option to operate in a virtually NO_x free regime (albeit with some efficiency trade-off at the extreme lean limit). Besides NO_x, other emissions are relatively well-behaved under TJI: hydrocarbon (HC) emissions can remain

low because the burn is more complete (Mahle noted HC levels on par with a normal port-injected engine), and particulate matter (soot) is generally very low in lean combustion. In fact, lean, premixed combustion inherently produces far less soot than rich combustion. Tests showed particulate emissions from a passive TJI system to be between those of port and direct injection systems (acceptable). Thus, TJI engines can potentially meet strict emissions standards with simpler or reduced aftertreatment, especially when running in lean mode. (One must keep in mind that if an engine toggles between lean and stoichiometric modes, it may still need a three-way catalyst for the latter; but heavy use of lean burn can significantly reduce cumulative NO_x output.)

2.4.6 Reduced Cycle-to-Cycle Variations

Lean or dilute combustion in a conventional engine often suffers from high cyclic variability – meaning one cycle might burn slower or partial-burn (misfire) compared to the next, causing rough running and efficiency loss. TJI greatly stabilises combustion due to its robust ignition and multiple flame initiation points. The result is much more consistent combustion from cycle to cycle. A dramatic example of this was seen in a small 2-stroke engine test: adding a pre-chamber ignition system resolved combustion stability issues of that engine and led to a reduction of up to 85% in the cycle-to-cycle variation (measured as COV of IMEP) when the pre-chamber orifices were optimally sized. This huge improvement means the engine runs significantly smoother and each cycle produces nearly the same power, whereas before there were large fluctuations. In general, researchers have reported that TJI reduces the Coefficient of Variation (COV) of IMEP to very low levels (often <2–3% even in lean operation, whereas a spark engine might be 5% or higher when running lean). This improved stability not only is good for NVH (noise, vibration, harshness) and drivability, but it also means the engine can be reliably operated at lean mixtures or high EGR dilution without misfiring – conditions that would be borderline unstable with a single spark ignition. Low cyclic variability also helps reduce the occurrence of extreme cycles that could produce high emissions spikes.

The net effect of these advantages is that a TJI-equipped engine can achieve diesel-like efficiency and low emissions while still using the Otto-cycle (premixed) combustion of gasoline. It combines the best of both worlds: the high efficiency and lean-burn cleanliness traditionally associated with diesel, and the smoother, higher-speed operation of a gasoline engine. This makes TJI especially attractive as the internal combustion engine industry seeks ways to meet tightening CO₂ and pollutant standards.

2.5 Role in Future Net-Zero Carbon Strategies

TJI technology is not only about improving today's gasoline engines, it also has important synergies with emerging net-zero carbon fuels and strategies. There are two key contributions of TJI in this context.

2.5.1 Maximising Efficiency to Reduce Fuel Consumption

Even as the world transitions to renewable or carbon-neutral fuels, the efficiency of engines remains crucial. Many alternative fuels (like synthetic e-fuels) are likely to be expensive or energy-intensive to produce, so using them efficiently is paramount. TJI's ability to push gasoline engines to 45–50% efficiency means that for a given work output, the engine consumes significantly less fuel than a conventional engine would. For example, consider a modern downsized 1.0 L turbo engine that might have ~35% BTE at best; if TJI allows it to reach ~45% BTE, that's roughly a 22% reduction in fuel consumption for the same power. In terms of CO₂ emissions, that's 22% less carbon emitted (if running on fossil fuel) – a substantial drop. Even if running on a carbon-neutral fuel, higher efficiency reduces the quantity of fuel needed, easing the burden on fuel production infrastructure. In hard-to-electrify sectors (like heavy-duty transport or hybrids requiring engines), these high-efficiency TJI engines could play a role in lowering overall emissions. In short, TJI is a bridging technology that can make combustion engines as frugal as possible, buying time and emissions headroom as electrification and renewables ramp up.

2.5.2 Enabling Difficult Carbon-Neutral Fuels

Perhaps even more significant is TJI's capability to facilitate the use of zero-carbon fuels such as hydrogen (H₂) and ammonia (NH₃) in combustion engines. These fuels do not emit CO₂ when burned, but they each have challenging combustion properties. Hydrogen has a very low ignition energy and burns extremely fast (which can cause pre-ignition or knock-like issues), whereas ammonia has a very high ignition energy requirement and burns very slowly (making it prone to misfire and producing lots of unburned fuel). Turbulent Jet Ignition can help tame both. In hydrogen's case, a pre-chamber can allow ultra-lean operation or controlled stratification to avoid overly rapid combustion; in ammonia's case, a pre-chamber (often with a small amount of hydrogen enrichment) can supply the necessary ignition energy and flame speed boost to combust ammonia cleanly. Recent research has shown that using an active hydrogen-fuelled pre-chamber to ignite ammonia-air mixtures significantly improves the combustion: the hydrogen in the pre-chamber produces strong jets that ignite the ammonia in the main chamber much more effectively than a spark plug could. This approach extends the lean limit of ammonia combustion, shortens burning duration, and improves stability. It also

reduces the total amount of hydrogen needed, instead of blending 10–20% hydrogen into the entire charge (as often needed in conventional spark-ignited ammonia engines), a TJI engine can use a small hydrogen pilot just in the pre-chamber to ignite a main charge of nearly 100% ammonia. That finding means we could have ammonia engines that only use minimal hydrogen, or even no hydrocarbon pilot fuel at all, by leveraging TJI. Moreover, since ammonia is very knock-resistant (octane number >120), one can run high compression ratios with it; TJI provides the means to ignite ammonia even at those high compression settings, unlocking efficient combustion (Reggeti & Northrop, 2024). Hydrogen as a fuel, on the other hand, ignites readily but tends to form NO_x due to high flame temperatures. Running hydrogen ultra-lean with TJI could mitigate NO_x formation by lowering temperatures while still maintaining stable combustion (some researchers have proposed lean or dilute hydrogen TJI to get near-zero NO_x emissions, similar to lean methane engines). In summary, TJI makes these carbon-free fuels more viable by solving their ignition and combustion-speed challenges, thereby aligning with net-zero goals. *(Detailed discussions of hydrogen and ammonia combustion are beyond our scope here, but it's important to acknowledge that TJI is a key enabling technology in those research areas.)*

2.6 Case Study: TJI Retrofit on a Modern 1.0 L Engine (Ford Fox)

To illustrate the practical impact of turbulent jet ignition, it's useful to compare a representative modern engine in stock form versus a TJI-equipped variant. Consider the Ford Fox 1.0 L EcoBoost engine, turbocharged 3-cylinder direct-injection gasoline engine, well-known for its high specific output (~125 PS) and used in many Ford compact cars. In stock form, this engine has a compression ratio around 10:1, runs stoichiometric combustion with a spark plug, and achieves a peak brake thermal efficiency of roughly 35–37% at best (around 230 g/kWh brake specific fuel consumption). Now imagine retrofitting this engine with a pre-chamber ignition system (replacing the spark plug with a pre-chamber module). What changes in performance and efficiency might we expect? Drawing from literature and experiments on similar conversions, we can outline several likely outcomes:

2.6.1 Power and High-Load Operation

If maximising power output is the goal, a TJI engine can be pushed harder without knock. The fast combustion and cooler charge (from either lean mixture or EGR dilution) means that at high load, one could advance ignition timing closer to MBT (minimum advance for best torque) without hitting knock limits or increase boost and compression to make more torque. In racing applications, TJI has enabled extremely high BMEP (brake mean effective pressure), for instance, F1 engines with pre-chambers run BMEP well above 20 bar, something a road

gasoline engine would struggle with on pump fuel. In our 1.0 L example, a TJI retrofit could potentially allow more aggressive turbo boost or a higher compression ratio (if fuel octane allows), thereby increasing peak power. One experimental study (on a similar scale engine) observed that IMEP at full load increased by 10–27% across various speeds when using pre-chamber ignition versus conventional spark. This suggests the engine could make proportionally more torque. Even if the goal isn't to increase absolute power, TJI could achieve the same power with a lower octane fuel or with added efficiency. Essentially, the knock suppression from TJI provides more headroom. Ford's engine calibrated on 95 RON gasoline could, with TJI, potentially run as if it had 98+ RON – meaning it wouldn't lose performance on regular fuel, or conversely, it could take advantage of premium fuel by running more boost. In summary, at high loads a TJI engine can either produce more output or operate with greater knock safety margins, depending on the calibration approach.

2.6.2 Part-Throttle Efficiency (Fuel Economy in Cruise)

One of the biggest gains with TJI appears in the low-to-mid load region (which is most of normal driving). Conventional engines must throttle the intake air to reduce power, which causes pumping losses (the piston has to pull against a partial vacuum). However, a TJI engine can run lean burn at part loads – meaning you don't throttle as much, you just dilute the mixture with excess air to cut power. With TJI reliably igniting lean mixtures, the throttle can stay more open. For example, tests have shown that when using TJI, the throttle opening at a given light load could be 42–66% wider than with spark ignition, dramatically reducing pumping work. As a result, fuel consumption drops. The same study noted the engine could maintain stable operation at loads with much less throttling, improving indicated efficiency proportionally. We would expect the TJI-enabled 1.0 L engine to have a much-improved efficiency map in the part-load region. Instead of, say, 25–30% efficiency at a 2–3 bar IMEP cruising condition, it might achieve well into the 40%+ range by running ultra-lean ($\lambda \sim 2$) with no throttle. Indeed, pre-chamber research engines have demonstrated >40% indicated efficiency at modest loads (even 5–6 bar IMEP) thanks to lean combustion strategies. In practical terms, this means significantly better fuel economy during highway cruising or city driving. The engine could operate in a high-efficiency lean mode whenever high power is not needed, then possibly switch to stoichiometric (with aftertreatment) for high load if necessary. This kind of bimodal operation (lean at cruise, stoichiometric at power) could yield 20%+ real-world fuel economy improvement, which is huge for a single technology change.

2.6.3 Octane Requirement and Knock

As discussed earlier, TJI reduces the octane sensitivity of the engine. The stock Ford engine is designed for ~95 RON gasoline. With TJI, it could likely run at the optimal spark timing on 95 RON even under conditions where normally it might retard timing to avoid knock. In other words, the effective octane requirement drops. This could allow the manufacturer to tune for regular fuel without performance loss, or to use lower-octane fuels in markets where premium is not available, all while maintaining high compression efficiency benefits. On the other hand, if 98 RON fuel is used, the TJI engine can exploit it further, potentially by raising the compression ratio a couple of points (since knock is not limiting). For example, one could envision increasing compression from 10:1 to, say, 13:1 in the TJI version (if structural limits allow) to gain additional efficiency. Normally, going to 13:1 would cause knock on 95 RON, but with TJI the knock is mitigated, especially if the engine is usually run lean. In summary, TJI broadens the fuel octane tolerance: it makes the engine less dependent on high octane, yet it can still benefit from it if available. (In F1, they used specialty high-octane race fuel in combination with pre-chamber ignition to push efficiency toward 50%; on the street, we might use standard fuel yet still get efficiency that previously might have required premium.)

2.6.4 Emissions and Aftertreatment

The stock engine likely uses a three-way catalyst (TWC) and runs stoichiometric at almost all times to keep emissions in check (NO_x, HC, CO). A lean-burn TJI engine poses a different emissions control strategy. NO_x emissions would be extremely low during ultra-lean operation (as noted, up to 99% lower), which is excellent. However, a TWC cannot function under lean conditions (it needs stoichiometric exhaust). So, if the engine spends a lot of time lean, it might need a lean NO_x trap or SCR catalyst to capture the small amount of NO_x that is produced. Alternatively, if it only goes lean in regions where NO_x is almost negligible ($\lambda > 2$), it might meet standards without additional aftertreatment. In practice, a compromise could be to run moderately lean ($\lambda \sim 1.6\text{--}1.8$) for efficiency, where there will be some NO_x; that NO_x would then require reduction via an SCR system (perhaps using a little ammonia or urea). The encouraging part is that because engine-out NO_x is so much lower with lean TJI, the aftertreatment system can be smaller or operate with higher conversion efficiency (or even passive SCR using ammonia slip if ammonia fuel is used in future). For hydrocarbons (HC) and CO, lean combustion typically results in a bit more HC (from incomplete burn of a sparse mixture), but the TJI's strong ignition helps keep HC emissions comparable to normal port fuel injection levels. The rapid burn and high turbulence ensure most fuel is consumed. The Ford engine with TJI would likely produce exhaust HC and CO that a simple oxidation catalyst could

handle, if not the TWC. Particulates (soot) should be very low because we're not operating in rich conditions at all – the mixture is always lean homogeneous, which is inherently low-sooting. Mahle's data indicated particulate levels with their jet ignition were between port-injection (very low soot) and direct-injection (higher soot) levels, suggesting it's not a big concern, especially if most fuel is injected in the port in a retrofit scenario.

To summarise some quantitative differences one might expect between the stock and TJI versions of a gasoline engine, the following are key metrics and their anticipated improvement (based on various literature sources and experiments):

2.6.4.1 Combustion Duration 20–30% shorter burn duration (crank angle degrees) with TJI versus a single spark, due to the rapid multipoint ignition.

2.6.4.2 Cycle-by-Cycle Variability Major reduction in COV of IMEP. In one case, variability dropped by 50–85% when using an optimised pre-chamber, indicating much more stable combustion.

2.6.4.3 Lean Limit Extended from about $\lambda \sim 1.4$ (spark ignition limit) to $\lambda \sim 2.5$ or even higher with TJI. Active pre-chambers with enrichment can reliably ignite mixtures beyond $\lambda = 2.0$.

2.6.4.4 Indicated Thermal Efficiency Increased from roughly 40% (typical spark ignition peak ITE) to ~45–50% under TJI lean operation. For example, 47%+ ITE at $\lambda 2.7$ (passive) and >52% ITE at $\lambda 2.4$ (active) have been reported. This corresponds to brake thermal efficiency climbing from ~35% to ~45% (accounting for losses) – an enormous improvement in fuel economy.

2.6.4.5 Knock Onset / Octane Number With TJI, the engine can run MBT timing at higher loads on a given fuel. Effectively, it's as if the engine's octane requirement is lowered by several points. One can also utilise a higher compression ratio without knocking. Mahle noted compression "*higher than normal*" could be used thanks to knock suppression in TJI mode. In practical terms, this might mean a jump from 10:1 to ~13–14:1 compression on regular fuel or using 98 RON to go even further (beyond what a conventional engine could do even with that fuel).

2.6.4.6 NO_x Emissions In ultra-lean operation ($\lambda > 2$), NO_x can be 99% lower than in stoichiometric combustion. Even at moderately lean ($\lambda \sim 1.5$ –1.8), NO_x is far below what it would be without TJI because the combustion temperatures are lower. This can help meet stringent standards when combined with appropriate aftertreatment.

2.6.4.7 Pumping Losses Drastically reduced at part load. With lean burn, the throttle can be nearly open, cutting pumping work. The earlier-cited study showed throttle positions 40–65% wider at the same output when using TJI, translating to big efficiency gains. This benefit is reflected in the engine's fuel consumption map, especially in low-load regions (where conventional engines suffer the most efficiency penalty from throttling).

2.6.4.8 Heat Transfer Losses Lean combustion yields lower peak combustion temperatures, which reduces heat losses to cylinder walls. However, there is a subtle trade-off: the pre-chamber itself has surface area that can extract heat. If a pre-chamber is too large or not well-scavenged, it might run hot and increase overall heat loss slightly. Proper design (including cooling or purge air) mitigates this. For instance, the Jetfire® active pre-chamber system includes a purge air blast that also cools the pre-chamber, preventing it from overheating. Net-net, engines running lean with TJI tend to have lower average combustion temperatures, benefiting the thermodynamic efficiency (γ remains higher due to excess air and less energy lost to the walls).

Overall, a retrofitted 1.0 L TJI engine would likely demonstrate significantly better fuel efficiency (especially in real-world driving conditions where part-load operation dominates) and a cleaner exhaust profile. Its peak power might also improve or at least become less sensitive to fuel quality. The main downsides would be the added system components (if active, an extra injector, fuel supply line, control for the pre-chamber; if passive, just a specialised spark plug assembly), and the need to recalibrate the engine's control strategies (especially for transients, as discussed next). Ford (and other OEMs) have indeed been researching such technology for future engines, often in conjunction with hybrid powertrains where an extremely efficient engine can charge a battery or provide cruise power while running at lean conditions, then be supplemented by electric assist when more power is needed.

2.7 Design Challenges and Considerations

While Turbulent Jet Ignition offers impressive benefits, implementing it in an engine involves addressing several engineering challenges. A balanced review must consider these factors and the solutions being pursued:

2.7.1 Pre-Chamber Design and Size Trade-offs

The geometry of the pre-chamber – its volume, shape, and the size/number of nozzle orifices, is critical. A larger pre-chamber can hold a greater ignitable mixture, potentially producing stronger jets that enable ignition of very lean main-chamber mixtures. However, if the pre-chamber is too large relative to the cylinder, it can increase heat losses (a bigger surface area

exposed to burning gases) and also trap too much residual gas from the previous cycle. Residuals in the pre-chamber dilute the fresh charge there and can make ignition less reliable (especially in passive systems where you can't purge it easily). Researchers found that incomplete scavenging of a passive pre-chamber limited its ability to handle EGR dilution – the leftover exhaust in the pre-chamber caused misfires when trying high EGR rates, because it degraded the mixture quality in the pre-chamber (Zhu *et al.*, 2022). The Jetfire® system (developed by Michigan State University/US Army) addresses this by adding a small scavenge air purge: essentially a timed blast of air into the pre-chamber each cycle, through a poppet valve, just before ignition. This purges residual gas and ensures a fresh (and cool) charge in the pre-chamber, even if the main chamber is heavily diluted. It also helps cool the pre-chamber nozzles; in testing, after several hundred hours of running with this system, they reported no degradation or oxidation of the pre-chamber nozzle holes, indicating the purge kept temperatures in check. Without such measures, other researchers have noted that pre-chamber orifices can overheat or even suffer torching damage over long use. Apart from volume, the nozzle arrangement matters: the number of holes and their angle influence how the jets penetrate and distribute. A study by Müller *et al.* (2018) found that adding more orifices and introducing a swirl effect in the pre-chamber can increase turbulence at the spark plug by ~50%, which helps ignite ultra-lean mixtures. They optimised a 6-hole design that produced much higher turbulent kinetic energy (TKE) at ignition than a conventional 4-hole. However, too much turbulence in the pre-chamber can actually make it harder for the spark to initiate (the spark could be blown out or must work harder to start a flame). So designers must strike a balance: enough and well-directed orifices to get good jet distribution and turbulence, but not so large or so many that the pre-chamber mixture is diluted or the spark struggles. The location of the orifices (e.g., one row or two rows, pointing to centre or edges) also affects how the flame jets travel – e.g., one optical study showed that only the lower-row jets in a two-row design were reaching near the cylinder walls effectively. These details underscore that pre-chamber design is an active area of R&D, often aided by CFD simulations and testing. Modern approaches like 3D printing allow intricate pre-chamber geometries to be prototyped, which is accelerating development.

2.7.2 Active vs. Passive Complexity

Choosing between a passive or active pre-chamber architecture has implications. Passive systems are attractive for their simplicity – you essentially replace the spark plug with a pre-chamber spark plug (which can even fit in the same hole in many cases). There's no additional fuel system needed. This makes passive TJI a potential retrofit or an easy addition to existing

engine designs. The downside is, as mentioned, less control over the exact mixture in the pre-chamber. At very lean overall mixtures, a passive pre-chamber might not get a rich-enough pocket to reliably ignite, leading to misfire if pushed too far. Active systems, with a dedicated injector, allow one to always have a near-stoichiometric or slightly rich mixture in the pre-chamber regardless of how lean the main chamber is. This extends the lean limit and gives more consistent performance. Mahle's original research actually focused on an active system (with 5% of fuel in pre-chamber) to maximize the benefits, but they later also developed a passive version for Formula 1 (due to the regulation of only one injector per cylinder). The complexity of an active system includes packaging an injector and spark plug together (which can be challenging in multi-cylinder heads with limited space) and controlling the timing and quantity of that pre-chamber injection precisely each cycle. It effectively becomes a second fuel injection system to calibrate. For OEM adoption, cost and durability of that extra injector (especially if it has to handle frequent small fuel doses or gaseous fuels) need consideration. Some automakers might opt for a passive system if it delivers, say, ~90% of the benefit with only ~10% of the complexity. However, for achieving >50% efficiency goals, many studies lean on active pre-chambers. For example, one experiment found that with a passive pre-chamber they were limited in EGR and lean burn, whereas the *active* pre-chamber could reach the optimal efficiency at $\lambda \sim 1.7\text{--}1.8$ with a lot of dilution. The passive configuration in that case suffered from residual gas and heat loss issues that capped its performance. Thus, in applications like stationary engines, race engines, or high-end vehicles, the active system may be justified. In simpler consumer engines, a passive system might still provide a big jump in efficiency (running moderately lean, e.g. λ 1.5, but not extreme lean) and be easier to implement. There's also a middle ground known as "dual-mode" TJI (DM-TJI) where the engine can operate as passive under certain conditions and active under others or uses a small air blast as in Jetfire which adds complexity but maybe not as much as a full second fuel system.

2.7.3 Transient Operation and Control

Most of the impressive results for TJI engines are in steady-state, controlled conditions. Transient behavior, rapid changes in load or speed, introduces challenges. For instance, if the engine goes from low load (lean) to high load (need more power) quickly, the control system may need to transition from lean operation to stoichiometric (if the turbo and aftertreatment strategy demand it) or to enrich the pre-chamber differently. There could be a brief hesitation or overshoot in torque if not carefully managed, since the fuelling and ignition characteristics differ between modes. If using an active pre-chamber, the injector response time and the mixing in the pre-chamber could impose some delay or complexity in transients. Additionally, at *very*

light loads or idle, a pre-chamber might be overkill, it might be easier to just use conventional spark (because the engine might not need to be lean at idle and a large pre-chamber could misfire if there's not enough fuel). Some designs might deactivate or not fire the pre-chamber at idle to avoid issues like incomplete combustion when there's little fuel. These mode switches have to be seamless. Another factor is cold start: pre-chambers (especially small ones) could be harder to fire when the engine is cold, and the fuel doesn't vaporize well. A spark plug by itself at least is trying to ignite a homogeneous charge; a cold pre-chamber might quench the flame if it's big and the walls are cold. Strategies like heating the pre-chamber (with electric heaters or with spark discharge techniques) or using a richer mixture at start might be needed. All these considerations mean the engine ECU (engine control unit) for a TJI engine will be quite sophisticated, managing perhaps two fuelling systems and variable ignition modes. This is certainly solvable, it parallels what has been done for other advanced combustion modes (like switching between HCCI and SI in some research engines, or managing cylinder deactivation, etc.). But it's a development effort that manufacturers must invest in. The payoff is huge efficiency gain; the cost is a more complex control calibration.

2.7.4 Emissions Control Compromises

While ultra-lean combustion yields negligible NO_x, not all operating points can be ultra-lean if we want maximum efficiency or power. There is a trade-off, the highest efficiency is often found at somewhat lean conditions (say $\lambda \sim 1.6\text{--}1.8$ with some EGR) rather than the leanest possible, because extremely lean operation can increase combustion losses or incomplete combustion. At those optimal points, some NO_x will be formed. For example, Reggeti & Northrop (2024) found that with ammonia fuel the best efficiency (around $\phi = 0.7\text{--}0.8$) coincided with the highest NO_x emissions, whereas very lean had low NO_x but worse efficiency. In a gasoline context, running at $\lambda \sim 1.7$ might yield great efficiency but produce, say, a moderate level of NO_x that still needs cleaning. Thus, a Selective Catalytic Reduction (SCR) system might be paired with a lean-burn TJI engine, using a small injector of urea (or even using ammonia if the fuel is ammonia) to reduce NO_x in the exhaust. This is similar to what diesel engines already do. Alternatively, some lean NO_x trap could briefly store NO_x and then purge it when the engine cycles to stoichiometric for a moment (like some stratified-charge gasoline engines in the past). TJI's benefit is it minimizes NO_x output to begin with – 90%+ less – making the burden on aftertreatment much easier. Another emission to watch is unburned fuel (HC or in the case of ammonia, NH₃ slip). If the pre-chamber jets don't reach some pockets or the mixture is extremely lean, a fraction of fuel could go unburned. TJI generally promotes complete burn, but at limits like $\lambda > 2.5$, combustion might be incomplete

in far corners of the chamber, raising HC a bit. Good chamber design and mixture preparation help; also, a light-up of an oxidation catalyst can clean up HCs. Nitrous oxide (N_2O) is a minor byproduct that can come from lean combustion of ammonia or even gasoline with certain EGR conditions; it's something researchers will monitor since N_2O is a potent greenhouse gas. Fortunately, N_2O tends to be low when combustion is fast and hot enough to complete (Reggati observed minimal N_2O at optimal ϕ in ammonia runs). Lastly, because TJI engines can run on the ragged edge of lean, any misfire will result in a spike of HC/CO (from that cycle's unburned fuel). However, since TJI is more stable, misfires are rarer than in a spark-lean engine. The control system can also detect a misfire via pressure sensors (some advanced engines have them) and adjust quickly.

2.7.5 Knock and Pressure Oscillations

Although TJI avoids traditional end-gas knock, one must be cautious about pressure oscillations that can occur due to the rapid energy release. If the jets are too strong or collide in the chamber, they can create a sort of ringing (sometimes called “jet-induced knock” or pressure wave excitation). Studies have noted that very large orifices can produce a slower jet velocity which might reduce this risk, whereas intermediate sizes might cause more intense pressure waves. The NDIA Jetfire paper reported seeing oscillations up to about 4 bar amplitude in the cylinder pressure after ignition when running with hydrogen fuel (which burns extremely quickly with TJI). However, these oscillations occurred *before* the main pressure peak and were not destructive – they were not classified as knock in the sense of end-gas autoignition, but rather a result of “*the rapid burning of the bulk charge associated with the turbulent jets.*”. In essence, the combustion was so fast it rang like a pseudo-detonation wave, but since the mixture was lean, it didn't develop into damaging knock. Still, if an engine experiences strong pressure oscillations, it could impose mechanical stresses or noise issues. Designing the pre-chamber to moderate the jet penetration (e.g., multiple smaller jets instead of one big torch) and using some combustion chamber shaping (maybe to absorb or damp waves) can alleviate this. In practice, engines like those in F1 have managed to harness rapid combustion without problem – they likely carefully tuned the ignition timing and pre-chamber geometry to avoid tangential pressure modes that cause knock-like ringing. It's worth stating that TJI in general makes knock less likely, so this issue is more of a high-performance corner case than a common problem.

2.7.6 Durability and Spark Plug Life

The environment inside a pre-chamber is harsh. The spark plug (or whatever ignition device is used, e.g. some use small ignition coils or corona discharge) is subject to higher pressures and a fiercer flame than in a normal cylinder. Electrode wear, spark erosion, and deposit formation on the plug are concerns. Traditional spark plugs may need modification – for instance, using precious metal electrodes or designs that protect the electrode from direct impingement of the flames. Some TJI systems integrate the spark into the pre-chamber body in clever ways or even explore alternatives like plasma ignition to light the pre-chamber. Maintenance intervals for the pre-chamber igniter might be shorter if using conventional plugs. Similarly, the small orifices could be prone to clogging by carbon deposits (from fuel or oil) over time. However, since each orifice sees a high-velocity flame jet every cycle, they tend to be self-cleaning to some extent (like a tiny blowtorch effect). Using good fuel (low sulphur, with detergents) and low oil consumption engine designs will mitigate deposit risks. The Jetfire tests running heavy fuel did not report coking of orifices, and they consciously kept the nozzle cool to avoid coking. Another factor is material: the pre-chamber tip experiences very high thermal stress. If it's not cooled well, it could crack or deform. Materials like stainless steel or Inconel alloys are typically used, and sometimes with thermal barrier coatings. Some designs, as mentioned, use active cooling (air purge or water jackets around the pre-chamber). All these add cost and complexity but are manageable in a development program. Finally, integration and packaging: in multi-cylinder engines, if each cylinder has a pre-chamber with an injector and spark plug, the cylinder head needs enough space to accommodate them plus fuel rails, etc. This can be challenging in small engines with 4 valves per cylinder where the spark plug hole is centrally located and already crowded. Solutions include smaller injectors, angle-mounted pre-chambers, or going to 3-valve head designs (some researchers have looked at sacrificing a valve for a pre-chamber in certain cases). Car companies will weigh these practical aspects when deciding how to implement TJI.

3. Methodology

This chapter sets out the approach used to evaluate a pre-chamber–assisted, dual-fuel retrofit for the 1.0 L Ford Fox GTDI engine in which hydrogen is supplied to an active pre-chamber and ammonia is used as the principal fuel in the main cylinder. The methodology combines a calibrated one-dimensional (1-D) engine model in Ricardo WAVE with targeted experimental measurements on a single-cylinder research engine configured to the Fox geometry. The objectives are to quantify, at full load across the speed range, brake thermal efficiency, nitrogen oxides (NO_x) formation, and combustion stability, and to compare these outcomes against the stock gasoline engine. The approach follows established practice in engine systems research, in which a validated 1-D model is used to couple gas exchange, turbocharging and in-cylinder heat release with tractable computational cost, while experimental data anchor combustion and emissions sub-models (Heywood, 2018; Ferguson and Kirkpatrick, 2015; Ricardo Software, 2019).

3.1 1.0 GTDI Baseline Engine

The baseline system, seen in figure 2, is the production Ford 1.0 L inline three-cylinder turbocharged direct-injection (GTDI) engine. The schematic used in this study shows the conventional EcoBoost layout: a single-entry compressor draws fresh air through an air filter, compresses it, and discharges to an air-to-air charge cooler before the plenum and intake manifold distribute the cooled charge to each cylinder. The throttle is retained upstream of the manifold because, although torque control in modern GTDI engines is primarily via boost and spark, the throttle remains necessary for transient control and failsafe modes. Each cylinder features intake ports and exhaust ports within a crossflow cylinder head. Exhaust runners merge in a compact manifold, feeding the turbocharger turbine which returns shaft power to the compressor; a wastegate regulates turbine power. Sensors and a control unit (represented in the schematic by a PID block; govern boost, air–fuel ratio and ignition timing. This architecture is representative of the Ford Fox engine, which has a bore of approximately 71.9 mm, stroke of 82.0 mm, compression ratio around 10.0:1 and a rated power in the region of 92 kW at 6000 rpm in stock gasoline form (SAE, 2012; Ford Motor Company, 2012).

1.0L GTDI Turbo

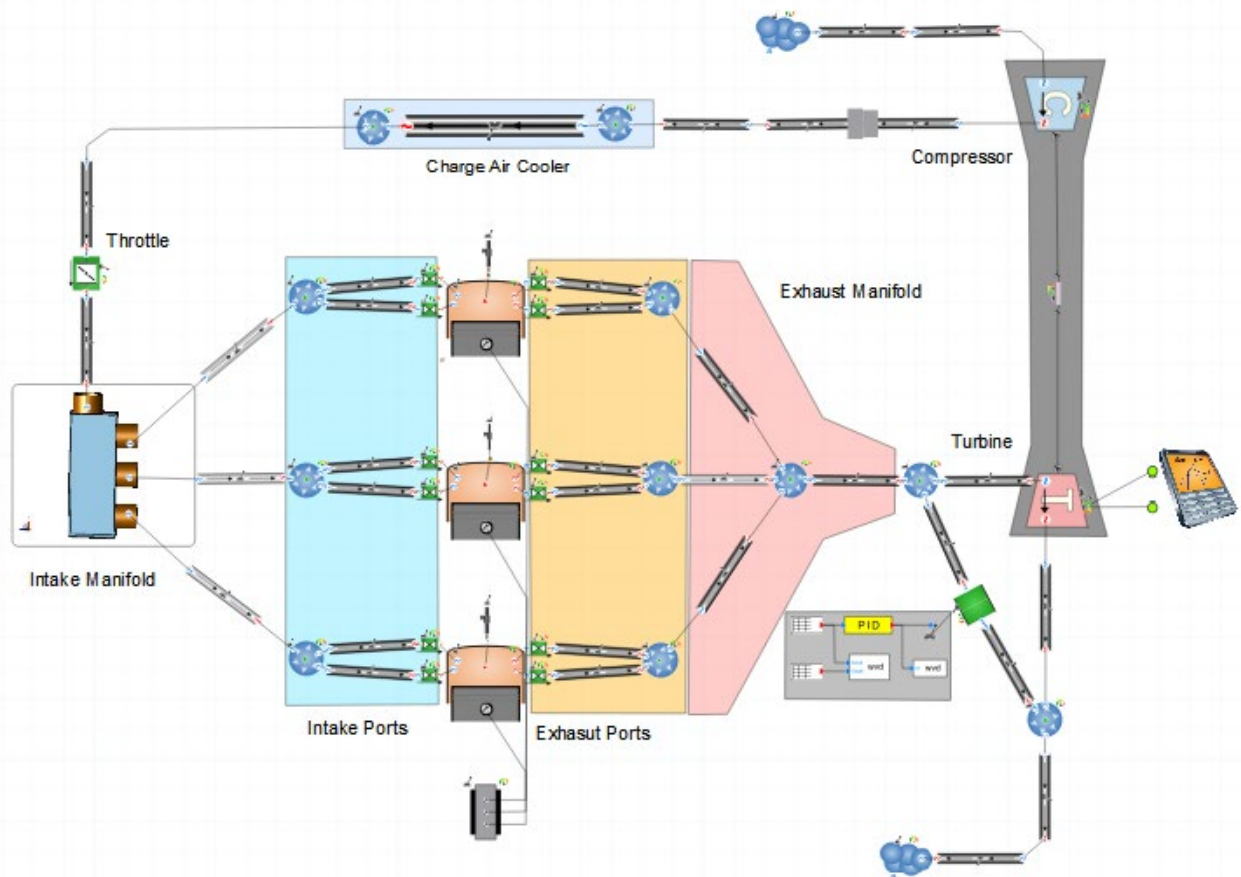


Figure 2 Ford Fox Ricardo Wave Model (Ricardo Wave, N.d.)

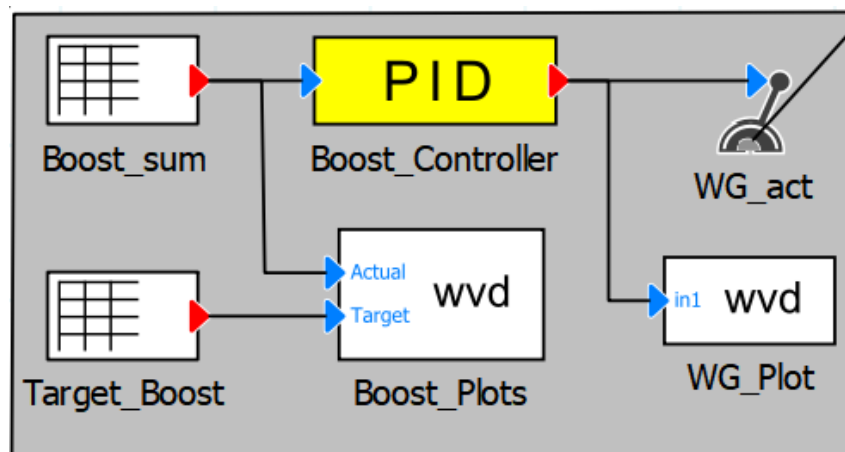


Figure 3 PID Booster Controller

Interpreted in methodological terms, the schematic defines the boundaries of the system to be modelled and assessed: the air path from compressor inlet to exhaust tailpipe, the turbocharger matching problem, and the in-cylinder processes that convert gasoline's chemical energy into indicated work. The retrofit examined in this study retains the stock gas-exchange architecture and turbocharger hardware, with the only modification being the replacement of the spark plug

by a pre-chamber assembly and the corresponding adjustment of the fuel system to deliver gasoline to both the main chamber and the pre-chamber as required.

3.2 Pre-chamber retrofit and fuelling strategy

The pre-chamber adopted here is an active design packaged in the original spark-plug boss of the cylinder head. It comprises a small cavity, approximately 1–2% of the cylinder clearance volume, that communicates with the main chamber through several calibrated orifices. A conventional igniter resides within the pre-chamber. At the commanded spark timing, the locally prepared pre-chamber mixture is ignited first; the ensuing high-temperature, radical-laden products are expelled through the orifices as turbulent jets, generating multiple ignition sites and intense local turbulence in the main chamber. This turbulent jet ignition (TJI) mechanism is well established to shorten burn duration, improve phasing toward top dead centre, and extend lean and dilution limits relative to conventional single-point spark ignition (Toulson et al., 2010; Heywood, 2018).

3.2.1 Mechanical integration and calibration envelope

To maximise retrofit feasibility, the pre-chamber module is designed as a direct thread-in replacement for the stock spark plug, maintaining comparable protrusion into the combustion chamber and clearances to valves and squish regions. The added volume is explicitly represented in the simulation model by a slight adjustment of the clearance volume so that compression ratio effects are captured. The orifice counts and diameters are selected to balance jet momentum (for reliable main-chamber ignition and penetration) against heat loss and quench risk within the pre-chamber; typical practice and prior literature guided the initial specification, with subsequent optimisation performed via sensitivity sweeps on total discharge area and hole orientation. The design preserves serviceability (plug reach, wrench access) and avoids interference with the injector spray or tumble structures.

Because the system is active, a small, dedicated fuel supply to the pre-chamber is provided. In the present configuration, gasoline is used for both the pre-chamber and the main chamber. A low-flow, high-precision injector meters a minor fraction of total fuel directly into the pre-chamber (nominally ~3–6% of cycle fuel at baseline), while the remaining ~94–97% is delivered to the main chamber via the production system (port or direct injection, depending on the specific engine variant). This arrangement enables a near-stoichiometric (or mildly rich, if required for robustness) pre-chamber equivalence ratio even when the global mixture in the main chamber is lean, thereby preserving reliable ignition over a wide operating envelope. Fuel rail pressure, injector targeting, and end-of-injection timing are calibrated to minimise fuel impingement on pre-chamber walls and to enhance mixture homogeneity prior to spark.

3.2.2 Operating strategy

At full load, two strategies are considered. The first maintains stoichiometric operation in the main chamber to exploit the three-way catalyst while using the pre-chamber to accelerate the burn and suppress end-gas knock, permitting operation closer to MBT timing and, where permitted, modest increases in effective compression or boost. The second pursues moderately lean combustion (e.g., $\lambda \approx 1.5\text{--}1.7$) at mid loads, leveraging TJI's robust ignition to reduce throttling losses and peak temperatures. In both cases, the pre-chamber mixture is targeted near stoichiometric immediately prior to the pre-chamber spark to generate strong jets, while the pre-chamber fuel share is kept deliberately small to minimise parasitic heat loss and maintain overall fuel economy.

A series of calibration sweeps is defined to quantify trade-offs among stability, burn rate, knock margin, and emissions. These include varying: (i) pre-chamber fuel fraction (approximately 2–8% of total cycle fuel); (ii) pre-chamber equivalence ratio (from slightly lean to moderately rich about stoichiometry); (iii) orifice total discharge area and jet targeting; and (iv) main-chamber global λ across the lean stability boundary identified by the model. Combustion metrics of interest are CA10–CA90 burn duration, phasing (CA50), coefficient of variation of IMEP, and apparent heat-release rates, with constraints placed on peak pressure rise rate to avoid excessive pressure oscillations. Where the aftertreatment requires, the strategy permits stoichiometric reversion at high load or during catalyst light-off, with TJI retained to maintain rapid, stable combustion.

3.2.3 Knock, octane sensitivity, and efficiency

For gasoline operation, TJI's distributed ignition and shorter burn duration reduce end-gas residence time at high temperature and lower the propensity for auto-ignition, thereby expanding knock-free spark advance latitude (Toulson et al., 2010; Heywood, 2018). This provides two levers: either maintain the stock compression ratio and recover efficiency by advancing toward MBT under conditions that were formerly knock-limited or raise effective compression ratio (subject to mechanical limits and fuel octane) to harvest additional thermodynamic efficiency. Sensitivity to fuel octane (95 vs 98 RON) is expected to narrow relative to conventional spark ignition because TJI mitigates knock structurally; nevertheless, premium fuel can still be exploited to extend the MBT plateau or support more aggressive boosting were warranted by performance targets.

3.2.4 Safety, materials, and implementation

The active pre-chamber adds a low-flow gasoline circuit local to the plug boss. Standard gasoline-compatible materials, stainless-steel or coated lines, automotive-grade seals, and heat-shielding—are specified. The pre-chamber body and nozzle tip use high-temperature alloys with attention to thermal mass and cooling via conduction into the head; orifice sizing is selected to balance jet momentum with acceptable thermal loading and to resist coking. The retrofit preserves existing service operations and diagnostic access. In the laboratory environment, leak-down tests and intrusive pressure transducers are used during development; production intent would rely on existing rail pressure, ionisation current, and cylinder pressure modelling for diagnostics.

3.2.5 Modelling representation

In the Ricardo WAVE model, the pre-chamber is represented as a small control volume coupled to the cylinder via calibrated restrictors that reproduce the effective discharge characteristics of the orifices. A two-stage heat-release formulation captures the pre-chamber ignition event followed by the jet-driven main-chamber combustion, with parameters identified from baseline pressure data and literature-informed priors on TJI burn acceleration. The added pre-chamber volume is included in the compression ratio calculation, and the pre-chamber fuelling schedule is synchronised to the intake and compression strokes to approximate realistic mixture preparation. Validation against experimental pressure traces ensures that predicted changes in CA50, burn duration, and pressure rise rates remain within acceptable error bounds before extending the analysis across the speed-load map.

Overall, the active gasoline pre-chamber retrofit provides a pragmatic pathway to reap the core benefits of TJI, fast, stable, low-temperature combustion; improved knock tolerance; widened lean operating range; and reduced pumping losses, without introducing dual-fuel complexity. The approach is consistent with the established understanding of pre-chamber jet ignition physics and its demonstrated efficacy in modern gasoline engines (Toulson et al., 2010; Heywood, 2018).

3.3 Ricardo wave model

A detailed 1-D engine model was built in Ricardo WAVE to simulate the full three-cylinder engine with unsteady gas dynamics. Each cylinder was represented with its own cylinder element and crank-angle resolved combustion model. The intake and exhaust systems were constructed using pipe elements and plenums dimensioned to the stock engine runners and manifolds. The turbocharger was represented by a turbine and compressor, estimates for shaft speed were used. An air-to-air charge cooler model was included with a fixed effectiveness and a quadratic pressure-loss characteristic. Mechanical losses were represented by WAVE's default friction correlation with coefficients adjusted to match the published brake-specific fuel consumption of the baseline engine. The model was first calibrated for the stock gasoline configuration to reproduce the reported full-load torque, boost and power curves; a single-Wiebe heat-release model was tuned to match measured phasing and burn duration within approximately five per cent across the speed range. This step ensures that the gas-exchange and turbo-matching sub-models are credible before introducing the pre-chamber and dual-fuel combustion (Ricardo Software, 2019; SAE, 2012).

3.4 Model calibration and validation

Model–experiment validation the stock gasoline model was validated against Ford CalVal (Calibration Validation report) engine dynamometer data reports. Comparison between ford data and simulation data are compared to validate the software and model within Ricardo wave.

3.4.1 Model Validation 6000 RPM Case

At 6000 rpm, the Ricardo WAVE simulation of the Ford Fox 1.0T, configured with a two-zone combustion model, shows close agreement with the experimental cylinder-pressure trace from Ford's calibration validation dataset. Both profiles capture the expected pressure rise during compression, a sharp peak around top dead centre, and the subsequent decay during expansion. Quantitatively, digitised traces from the two datasets yield a root-mean-square error (RMSE) of 1.44 bar, corresponding to a normalised RMSE of 1.83% relative to the experimental peak pressure, and a mean absolute percentage error (MAPE) of 2.66% across the full 720° cycle (<10-15% is often considered acceptable for validation in engineering). The simulated peak pressure is 80.3 bar compared with the measured 76.5 bar, representing a modest over-prediction of +4.7%. The main residual errors are localised around top dead centre (a slightly sharper modelled heat-release peak) and during the early expansion stroke (a faster pressure decay in the model), while the late-cycle region shows a small positive bias in measured pressure relative to the simulation. These discrepancies are consistent with heat-transfer and boundary-condition sensitivities: in this study, the piston, liner, cylinder-head, intake-valve,

and exhaust-valve boundary temperatures were estimated, and the coolant-side surface area in the sub-model was a rough approximation. Such assumptions tend to make peak heat release appear slightly more aggressive and can under- or over-estimate post-combustion heat losses. Overall, the magnitude and phasing of the pressure trace are well reproduced, and the small errors (RMSE ≈ 1.4 bar; $\leq 5\%$ peak deviation) validate Ricardo WAVE with the two-zone combustion approach for system-level performance prediction of this engine at high speed. For studies that are heat-transfer-limited, or knock-constrained, improved specification of thermal boundary conditions and coolant-side geometry is recommended to further reduce the near-TDC and expansion-stroke residuals.

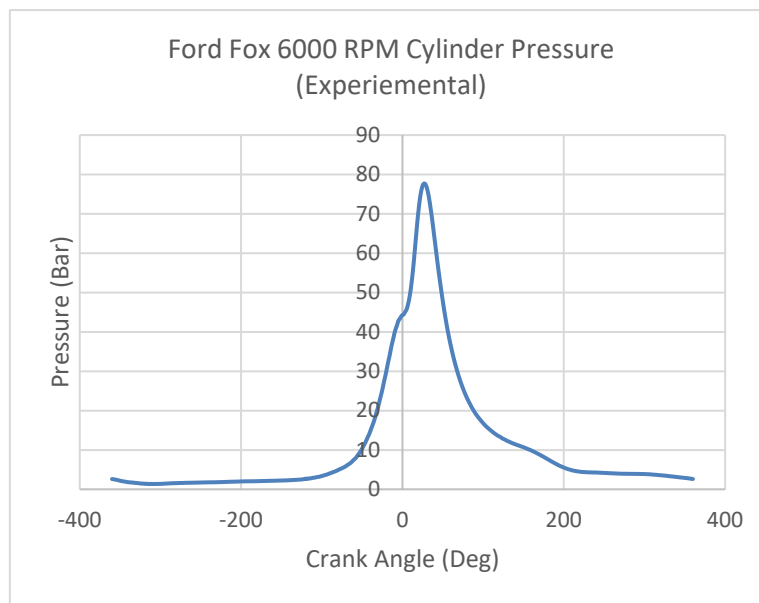


Figure 4 Cylinder Pressure @ 6000 RPM (Experimental)

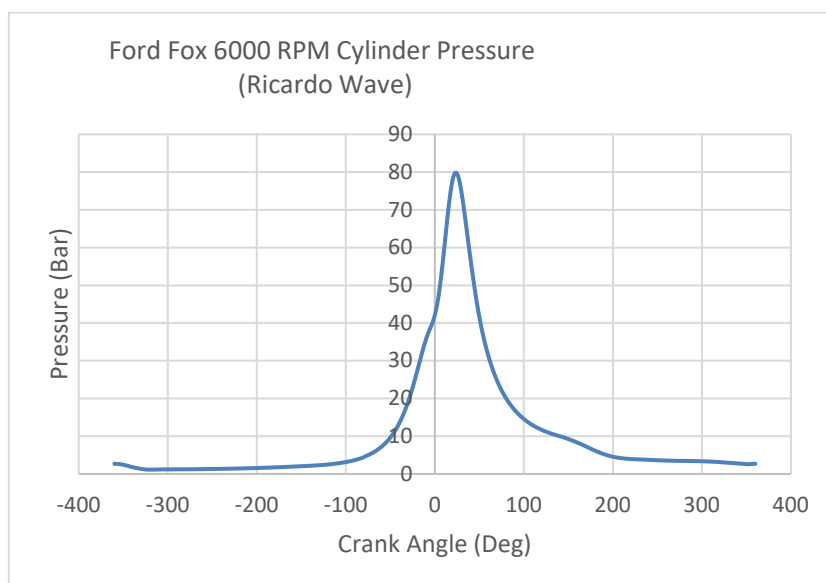


Figure 5 Cylinder Pressure @ 6000 RPM (Ricardo Wave)

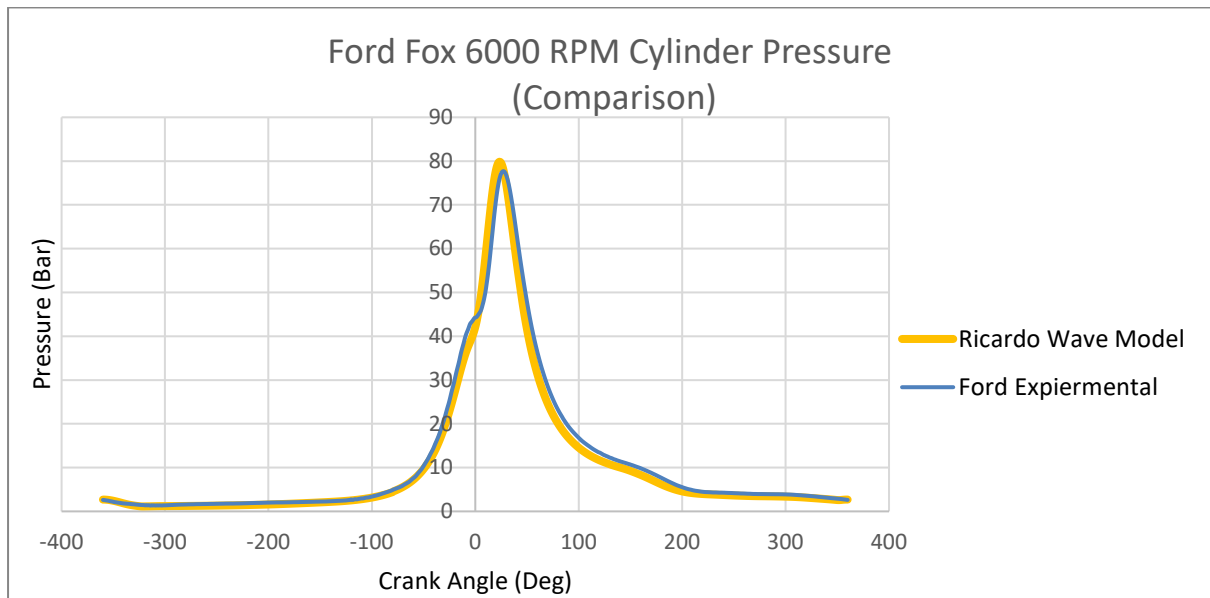


Figure 6 Cylinder Pressure @ 6000 RPM (Comparison)

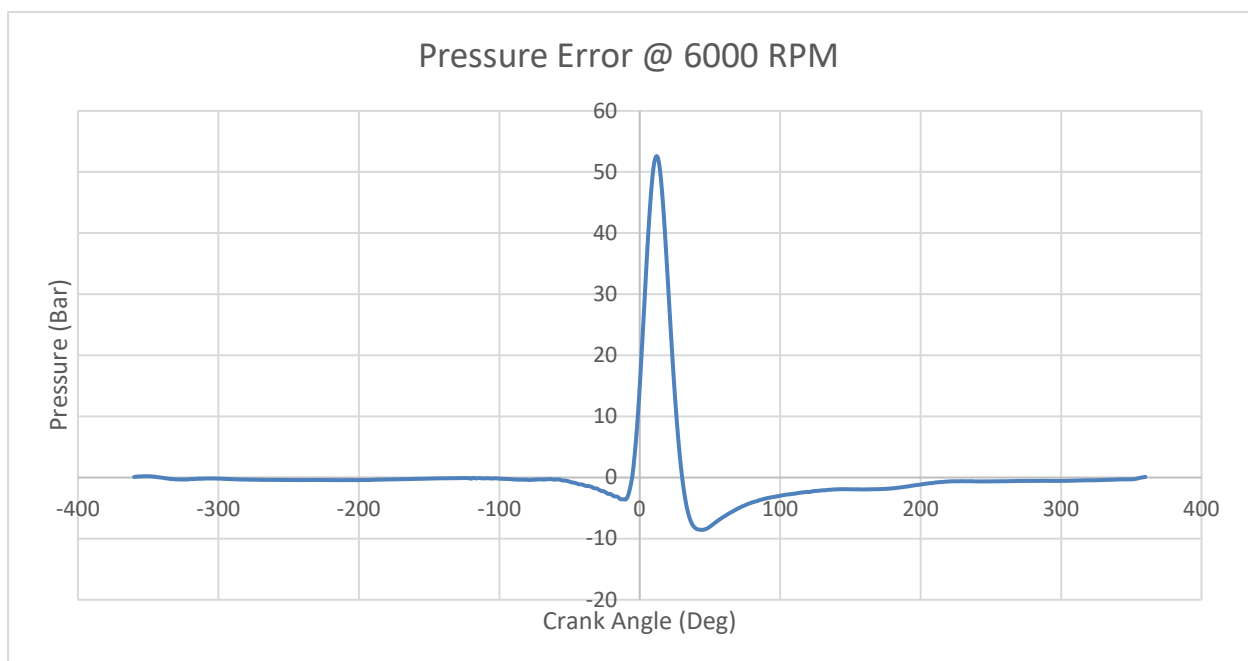


Figure 7 Pressure Error @ 6000 RPM

3.4.2 Model Validation 3000 RPM Case

A comparison between the experimental cylinder pressure trace and the Ricardo Wave simulation for the Ford Fox engine at 3000 RPM (Figures 8–9) demonstrates that the model captures the overall pressure evolution with reasonable accuracy. Both datasets exhibit the expected compression rise, peak combustion pressure around top dead centre, and subsequent expansion decay, with the simulation reproducing the peak pressure magnitude (~82 bar) and general phasing of events. However, notable discrepancies arise near top dead centre: the experimental data shows a distinct two-stage pressure rise with a pre-peak shoulder (~45–50 bar), whereas the Ricardo Wave model predicts a sharper, single pressure peak. This results in a systematic error pattern, with underprediction prior to peak pressure, overprediction at the peak, and underprediction during early expansion (Figure 10). These deviations suggest that the combustion model, which applies a simplified single-zone Wiebe function, does not fully capture the multi-stage heat release evident in the experimental data. Nevertheless, beyond ~200° crank angle, the error diminishes, indicating that the compression and expansion processes are well represented. Overall, the validation confirms that the Ricardo Wave model provides a good first-order approximation of cylinder pressure behaviour but highlights the need for refinements in combustion phasing and heat release modelling, potentially through multi-zone or double-Wiebe approaches, to improve predictive fidelity.

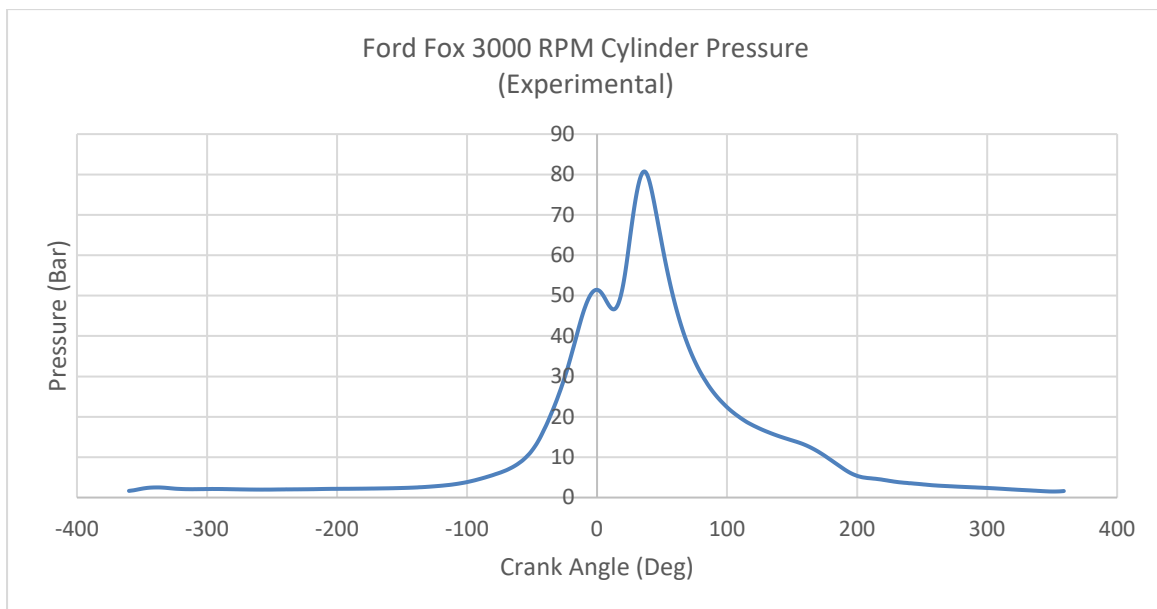


Figure 8 Cylinder Pressure @ 3000 RPM (Experimental)

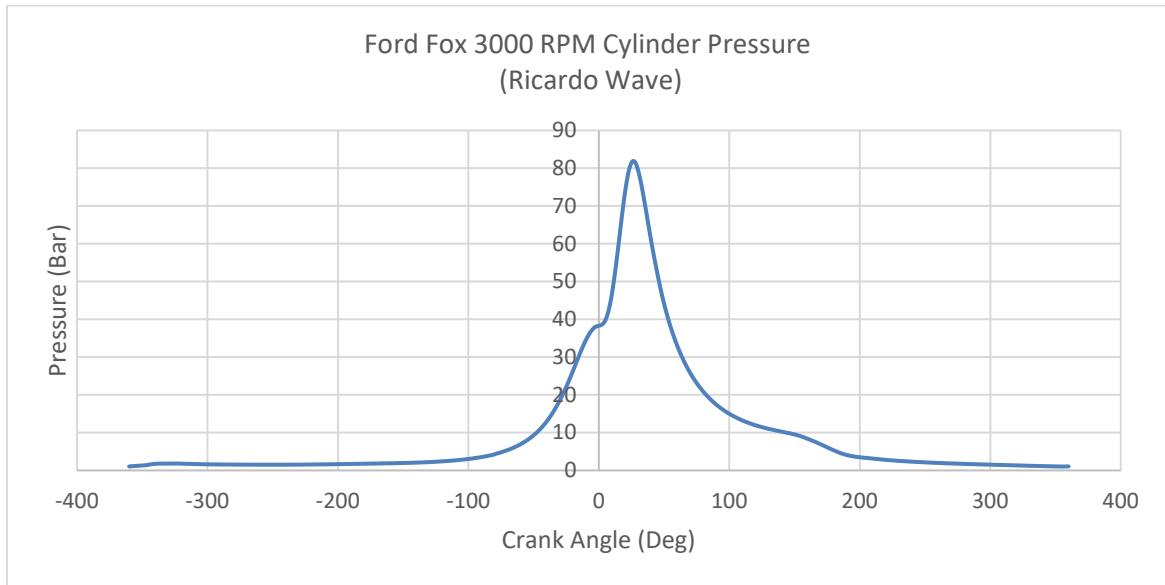


Figure 9 Cylinder Pressure @ 3000 RPM (Ricardo Wave)

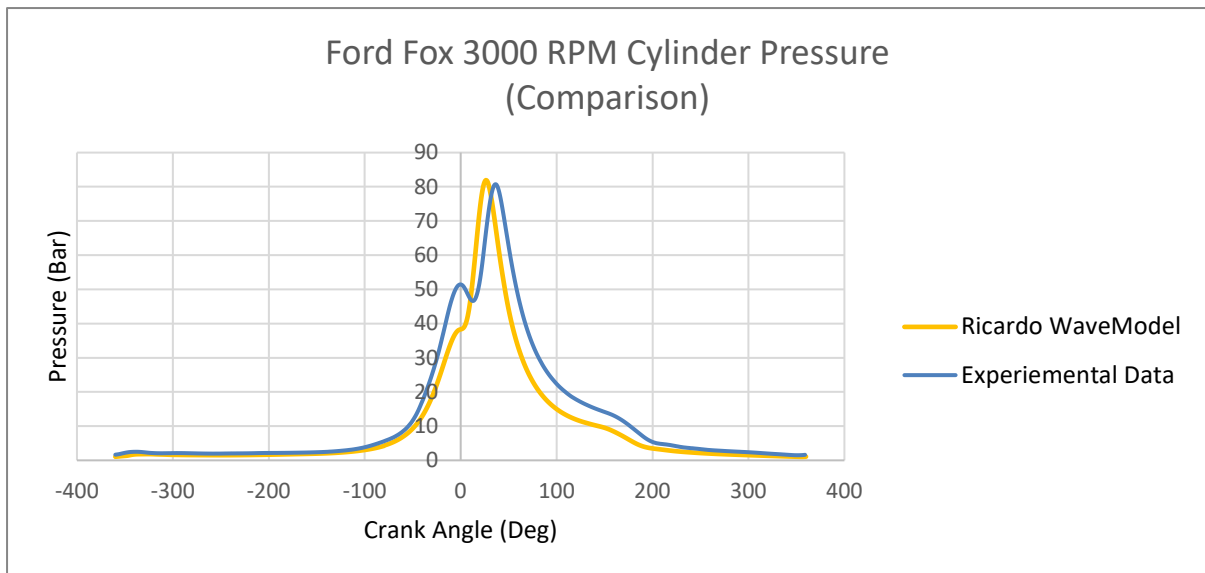


Figure 10 Cylinder Pressure @ 3000 RPM (Comparison)

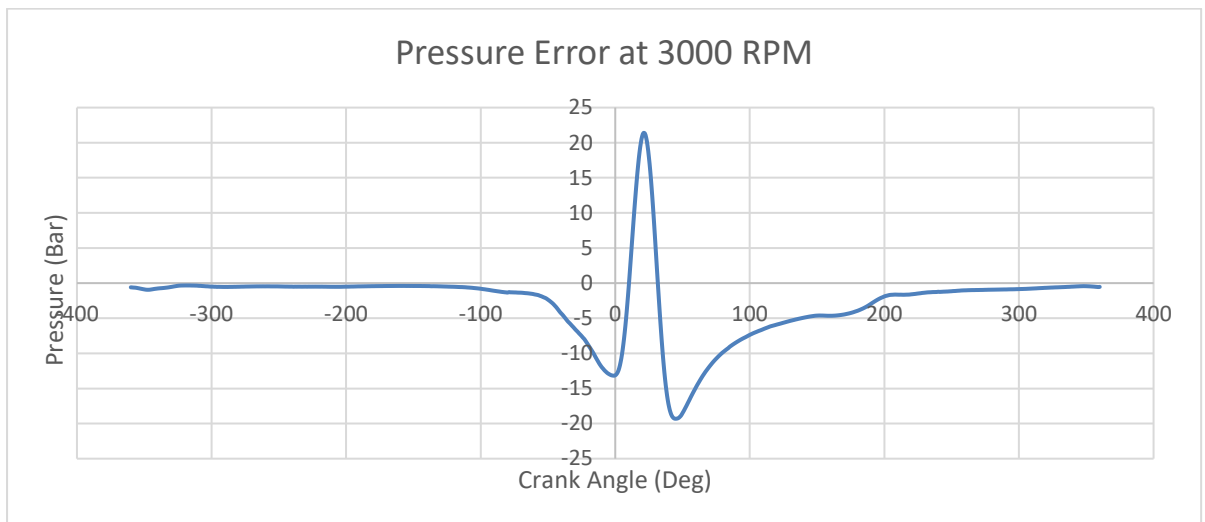


Figure 11 Pressure Error @ 3000 RPM
MSc Motorsport Engineering

3.4.3 Model Validation 1000 RPM Case

At 1000 rpm, comparison of the cylinder-pressure trace from the Ford Fox 1.0T experimental dataset with the Ricardo WAVE simulation highlights the challenges of accurately reproducing idle operation using a one-dimensional two-zone combustion model. The experimental data show a peak in-cylinder pressure of approximately 55 bar, whereas the WAVE model predicts a higher value of 62.7 bar, representing an over-prediction of 13.9%. Across the full cycle, RMSE between the traces is 6.8 bar, corresponding to a normalised RMSE of 12.3% of the experimental peak pressure. The mean absolute percentage error (MAPE) is 30.7%, indicating relatively large cycle-wide deviations compared with those seen at higher engine speeds. The primary discrepancies are concentrated near top dead centre, where the modelled combustion phasing and heat-release rate appear more aggressive than measured, and during the expansion stroke, where the simulated pressure decays more rapidly. These differences can be attributed to uncertainties in boundary condition specification, piston, liner, head and valve temperatures, and coolant-side area were estimated, and to inherent limitations of the cycle-averaged two-zone approach in capturing cycle-to-cycle variability and residual gas effects at very low speed. Despite these shortcomings, the overall pressure trace shape is reproduced, and the model remains a useful predictive tool for system-level analysis. However, the results underline the need for careful calibration and improved boundary-condition definition if Ricardo WAVE is to be extended reliably into idle and low-speed operating regimes.

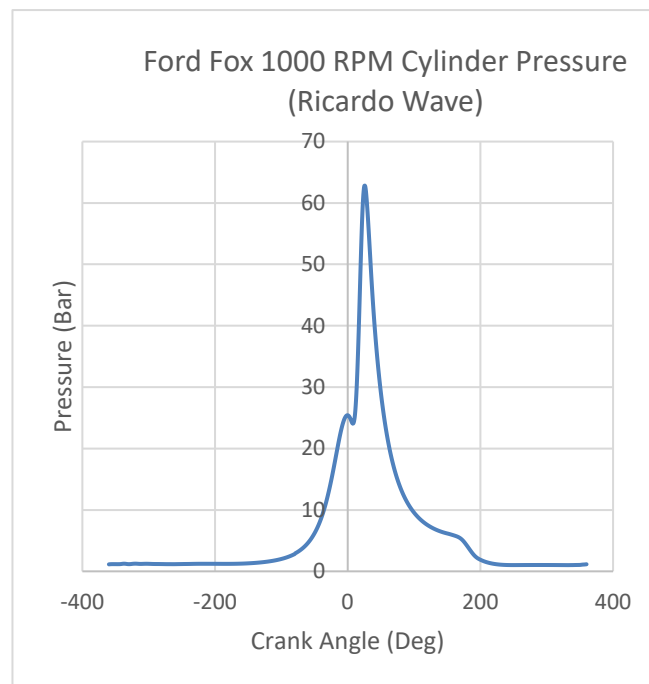


Figure 12 Cylinder Pressure @ 1000 RPM (Ricardo Wave)

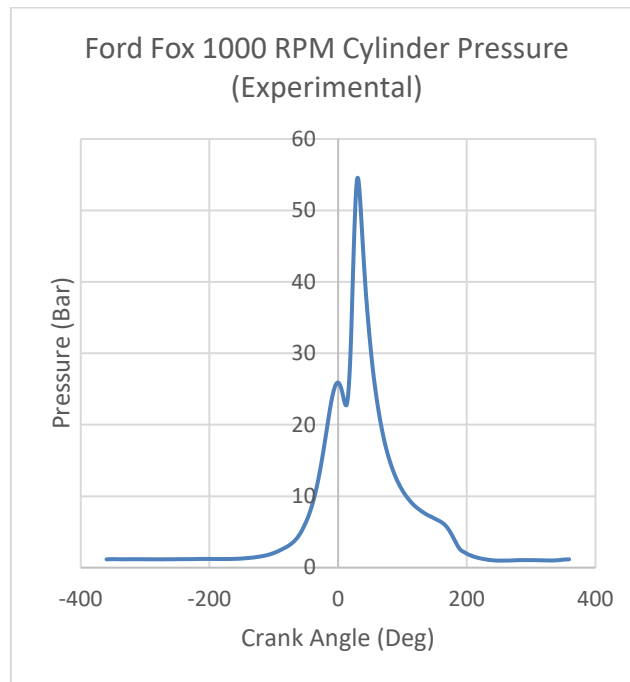


Figure 13 Cylinder Pressure @ 1000 RPM (Experimental)

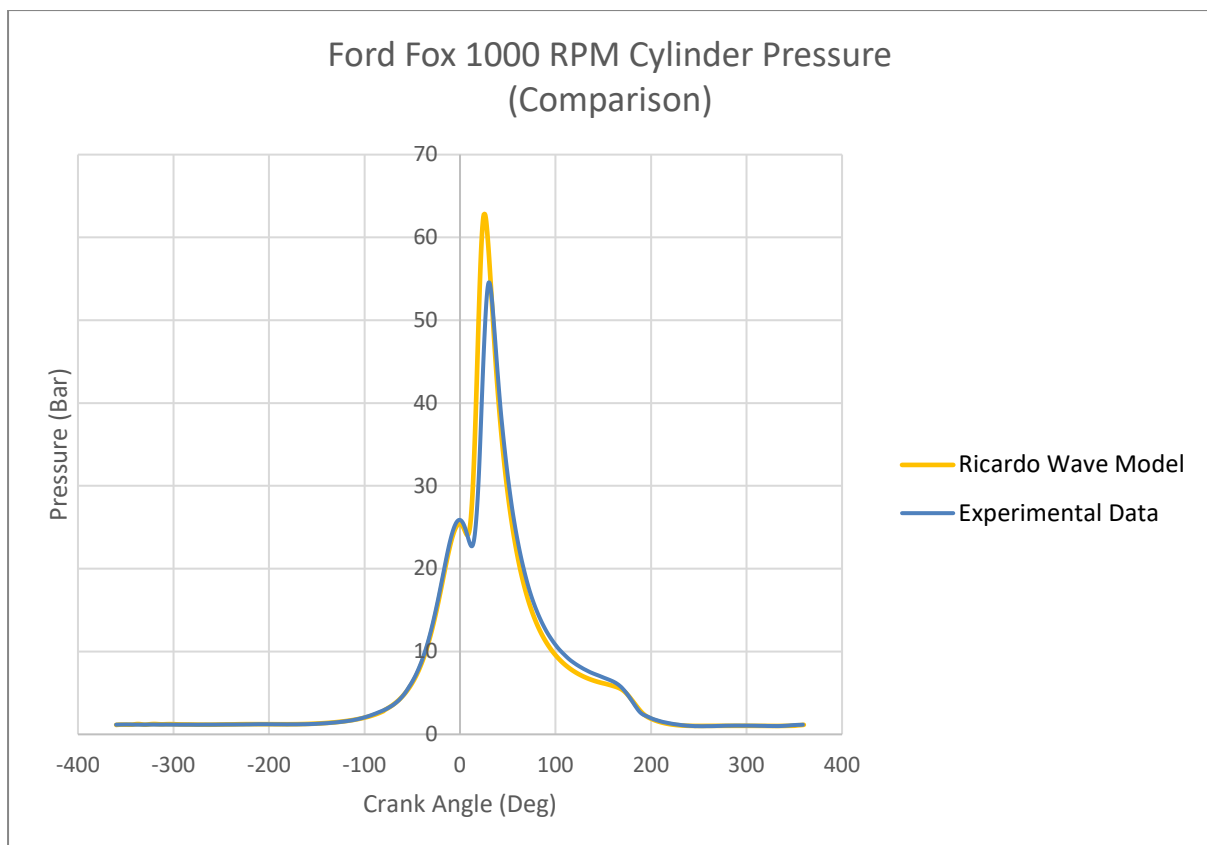


Figure 14 Cylinder Pressure @ 1000 RPM (Comparison)

The Ricardo WAVE model overpredicts in-cylinder pressure, with the largest error occurring just after TDC, as seen in figure 15. This suggests the modelled combustion is initiated slightly earlier and proceeds more aggressively than in reality. Likely causes include estimated boundary temperatures and coolant-side geometry, which reduce predicted heat transfer losses,

as well as limitations of the two-zone combustion model in representing residual gas effects and low-speed combustion variability.

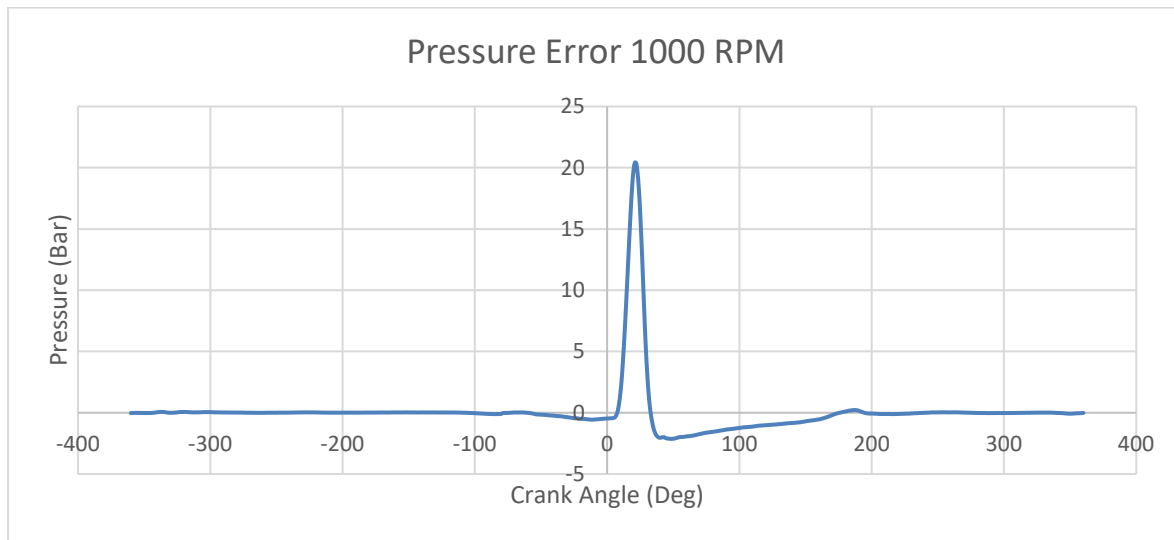


Figure 15 Pressure Error @ 1000 RPM

3.4.4 Duration of fuel mass burn (10-90%) vs Engine speed

In the validation of the combustion model, the duration between 10–90% fuel mass burned predicted by the Ricardo WAVE simulation was compared against experimental data from the Ford 1.0L Fox engine operating at full load with 95 RON fuel. The simulation results show an almost constant burn duration of approximately 20°CA across the entire speed range, with minimal variation as engine speed increases. In contrast, the experimental data exhibit a clear upward trend, with burn duration increasing from around 15°CA at 1000 rpm to approximately 26°CA at 6000 rpm. This behaviour is physically expected, as higher engine speeds reduce the available time per crank angle and increase turbulence and flame stretch, leading to longer combustion durations (Heywood, 2018). The discrepancy arises partly because the combustion in the WAVE model is represented using a two-zone approach, in which the charge is divided into burned and unburned zones, and the heat release is governed by a Wiebe function. While computationally efficient, this formulation simplifies turbulence–flame interactions and often assumes weakly varying or fixed burn durations, which limits its ability to capture the strong sensitivity of combustion duration to engine speed observed experimentally (Rakopoulos and Kyritsis, 2001; Ricardo, 2021). Consequently, calibration of the combustion model constants against measured data, or the adoption of a more advanced combustion model such as multi-zone or CFD-based combustion, is required to improve predictive accuracy (Turns, 2012; Zhao, 2014).

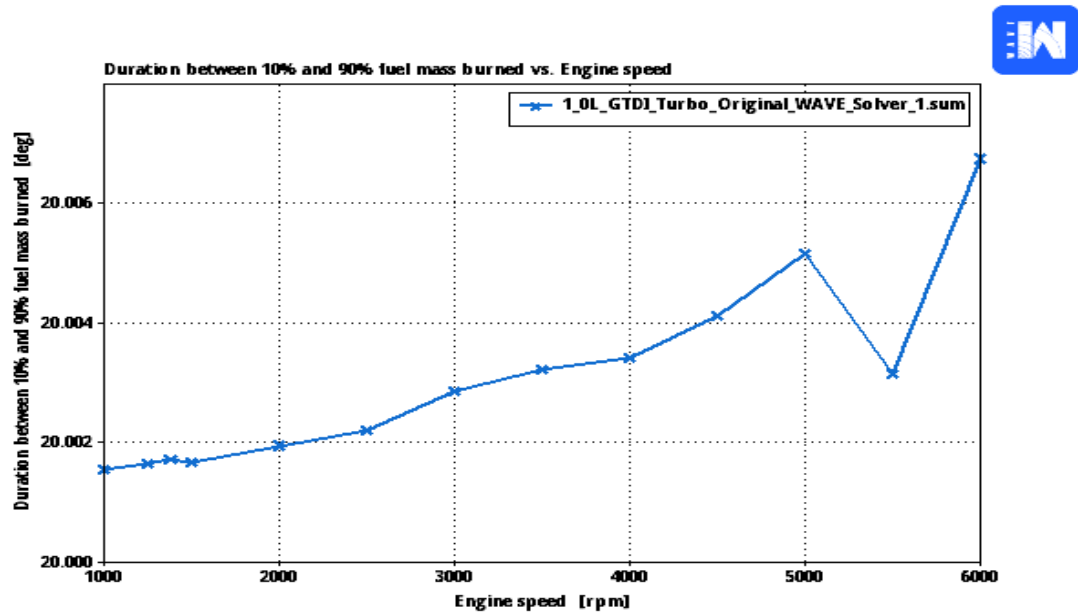


Figure 16 CA 10-90% (Ricardo Wave)

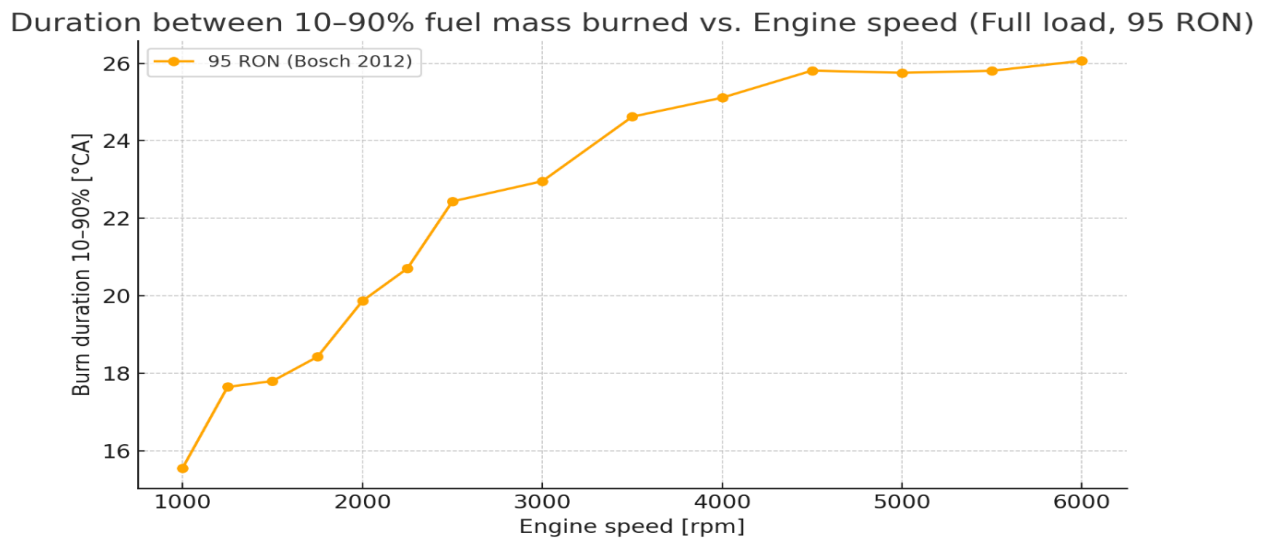


Figure 17 CA 10-90% (Experimental)

3.4.5 Brake Power and Torque

The experimental results for the Ford 1.0T Fox 125 PS engine are shown in Figures 19 and 21, presenting both the power and torque curves as a function of engine speed. The measured power output rises steadily with increasing RPM, reaching a peak of approximately 125.2 PS at around 6000 RPM before slightly tapering off beyond this point. This is consistent with the rated power of the engine and confirms that the experimental data aligns closely with the manufacturer's specifications.

The torque curve, on the other hand, highlights the effect of Ford's over boost system. Under normal operating conditions, the engine delivers around 170 Nm (125 ft·lbf) of torque. However, with over boost engaged, the experimental data shows a plateau of 200–202 Nm between 2000 and 3000 RPM, which is higher than the expected baseline. This additional torque provides improved mid-range performance, particularly during transient acceleration demands, before gradually declining at higher engine speeds. Overall, the data indicates that the Ford 1.0T Fox 125 PS engine achieves its design targets, with the over boost function providing short-duration enhancements to torque delivery beyond its nominal specification. This behaviour is advantageous for drivability and flexibility, especially in real-world driving conditions where mid-range torque is critical.

A comparison with the Ricardo WAVE simulation model (Figures 18 and 20) further illustrates this behaviour. The WAVE model, configured without an over boost function in the turbo and wastegate PID controller, reproduces the rated power and nominal torque values of the engine but does not capture the higher mid-range torque observed experimentally. In the model, power increases smoothly to ~128–130 hp (~127–132 PS) at 6000 rpm, slightly above the experimental result, representing an over-prediction of ~3–6% at the top end. The torque discrepancy is more pronounced in the mid-range: while the model produces a plateau of ~165–168 Nm between 2000–3000 rpm, the experimental results show ~200–202 Nm due to the activation of over boost. This difference highlights the ~15–20% additional torque contribution from over boost, which the current model configuration cannot replicate. At higher engine speeds (>4000 rpm), both datasets display the expected torque decline, though the model predicts a smoother roll-off, reaching ~135–140 Nm at 6000 rpm.

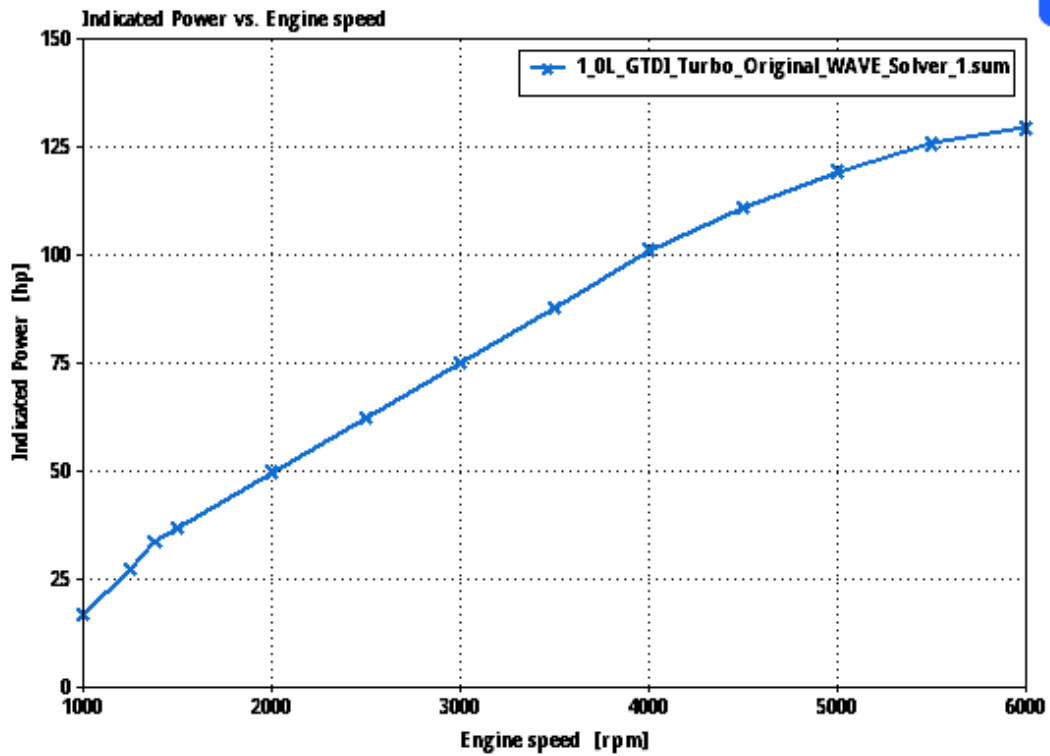


Figure 18 Power (Ricardo Wave)

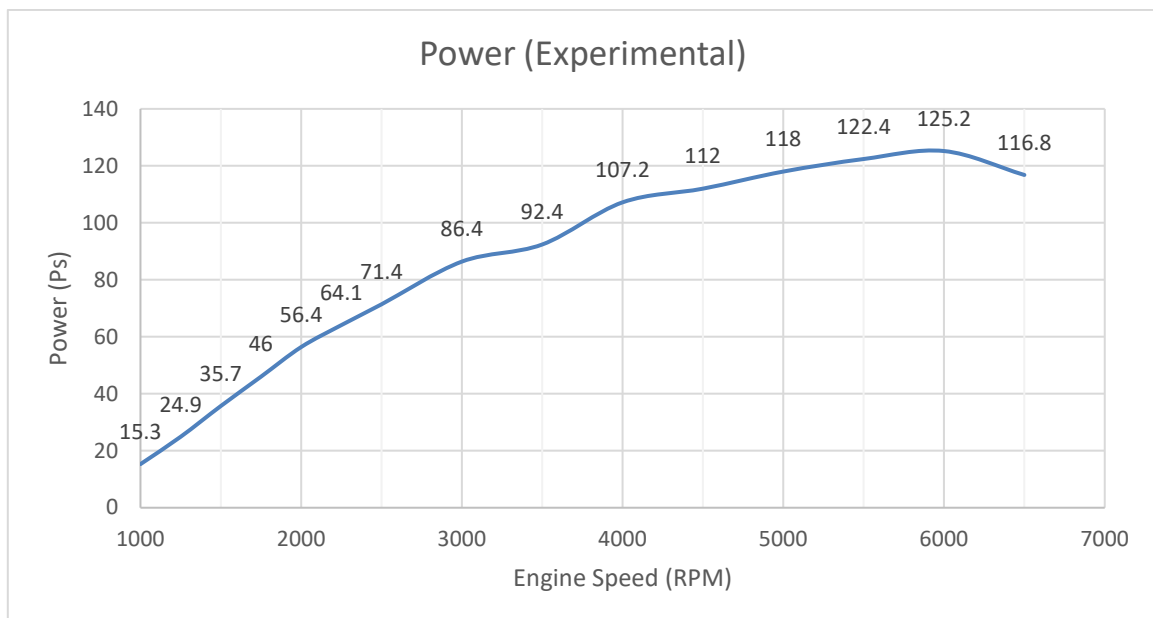


Figure 19 Power (Experimental)

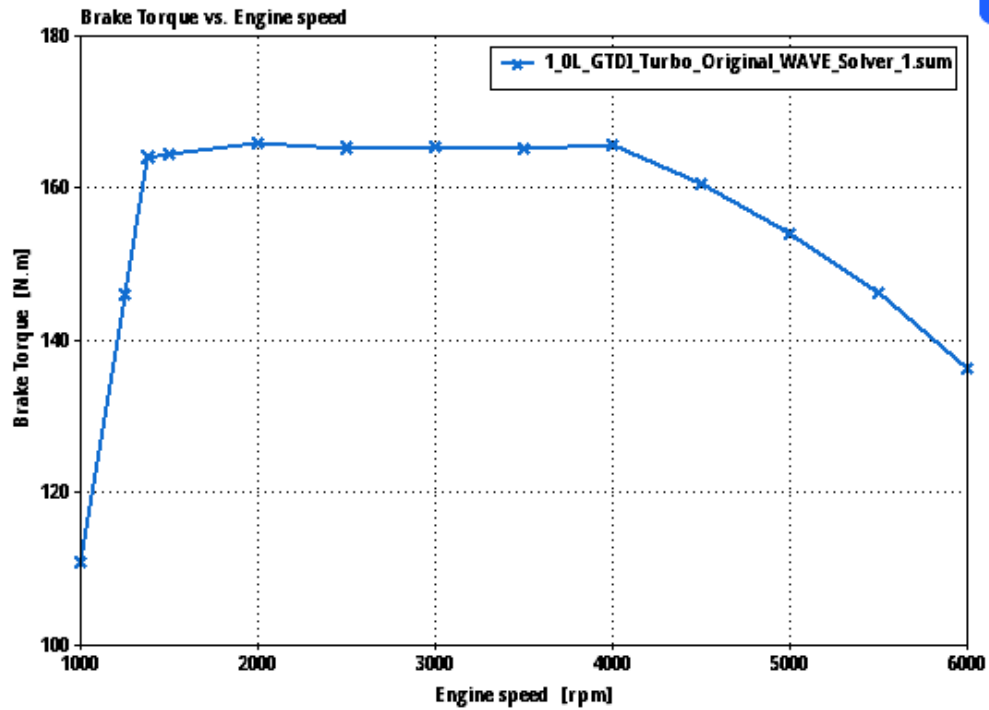


Figure 20 Torque (Ricardo Wave)

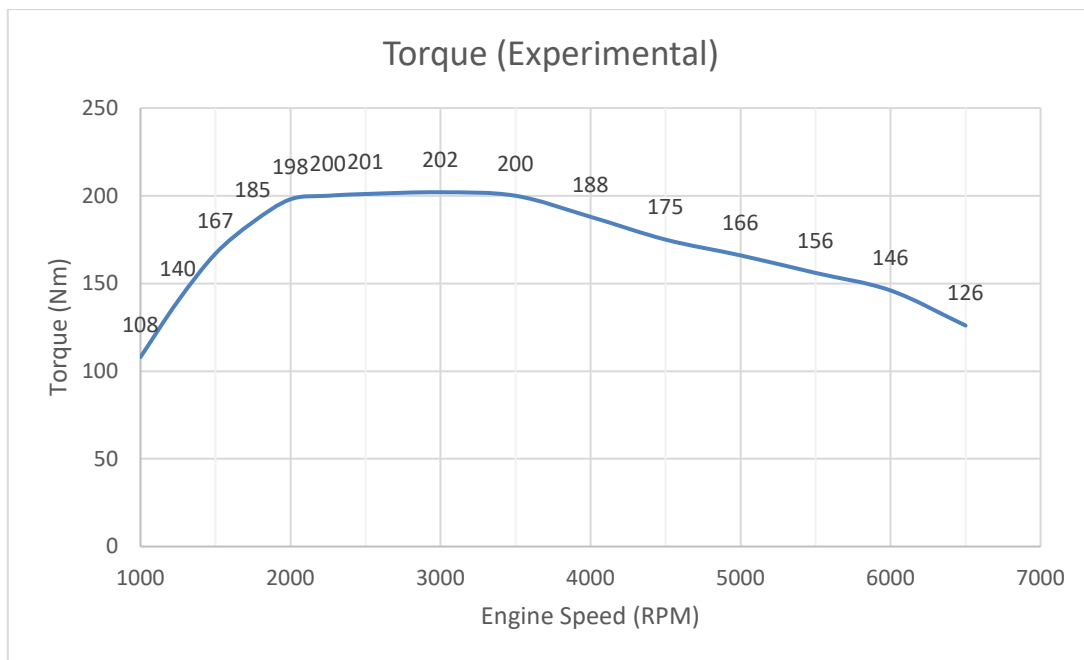


Figure 21 Torque (Experimental)

3.4.5 Thermal Efficiency

The brake and indicated thermal efficiency trends extracted from the Ricardo WAVE model for the Ford 1.0 L GTDI Fox engine are presented in Figures 22 and 23. Brake thermal efficiency (BTE) peaks at approximately 33–34 % around 2,000–4,000 rpm, while indicated thermal efficiency (ITE) reaches 36–37 % in the same mid-range. These values fall within the typical performance envelope for current production spark-ignition gasoline engines, where BTE commonly ranges from 30–36 % and theoretical upper limits approach 45 % with optimized strategies (Zhang et al., 2022). Recent research and development efforts suggest that with advanced technologies, such as lean combustion, Miller/Atkinson cycles, thermal barrier coatings, friction reduction, intake boosting, and engine downsizing, BTE values exceeding 50 % may be achievable (Wang et al., 2021).

While no published thermal efficiency figures for the Ford 1.0 T Fox engine are directly available, the simulated BTE of ~33–34 % suggests reasonable alignment with expected production-level performance. The fact that ITE exceeds BTE reflects inherent mechanical and auxiliary losses present in the engine, as per standard thermodynamic understanding (Stone, 1999). Given that over boost logic was not incorporated in the model, mid-range torque surges present in the real engine, potentially enhancing transient efficiency, are not represented. Thus, the modelled efficiencies provide a robust baseline with room for improvement. Future model enhancements incorporating over boost strategies, cooled EGR, or other advanced combustion control could shift BTE closer to the 40 % range, trending toward the innovative benchmarks, such as prechamber ignition, as seen in experimental literature (Zhang et al., 2022; Wang et al., 2021).

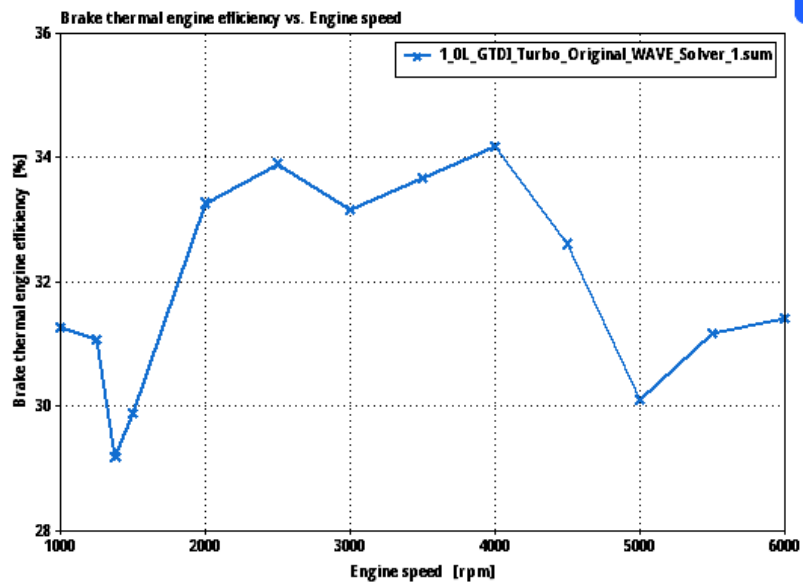


Figure 22 Ford Fox Brake Thermal Efficiency

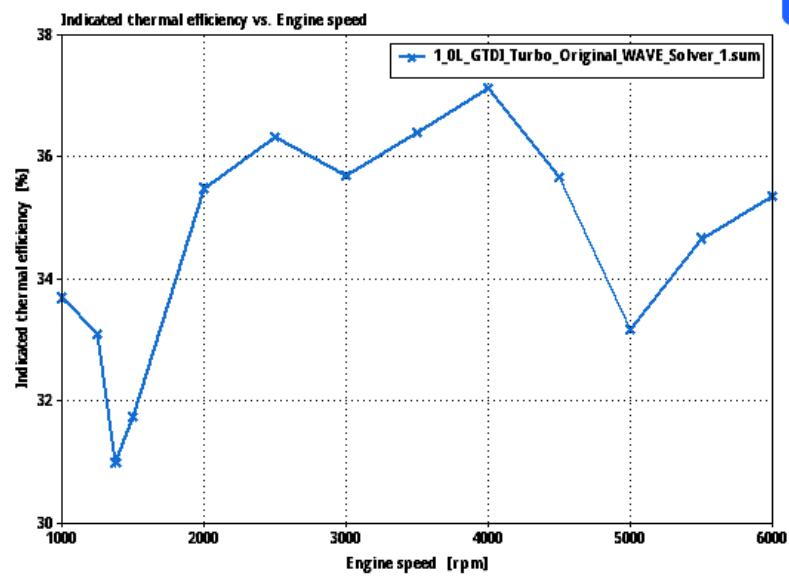


Figure 23 Ford Fox Indicated Thermal Efficiency

3.4.6 Emissions

The in-cylinder temperature profile (Figure 24) shows a typical four-stroke cycle trend for a turbocharged SI engine, with compression raising temperatures steadily before a sharp spike at top dead centre (TDC) during combustion. Peak combustion temperatures approach 2,600–2,700 K, followed by a decline during expansion and further cooling through the exhaust and intake strokes. These high temperatures directly explain the thermal NO_x trends seen in Figure 25, where NO_x concentrations peak around 2,000–3,000 ppm in the mid-range (≈ 2000 – 2500 rpm). According to the extended Zeldovich mechanism, NO_x formation increases exponentially with flame temperature above $\sim 2,000$ K (Heywood, 2018), making these temperature spikes the dominant factor in raw NO_x emissions.

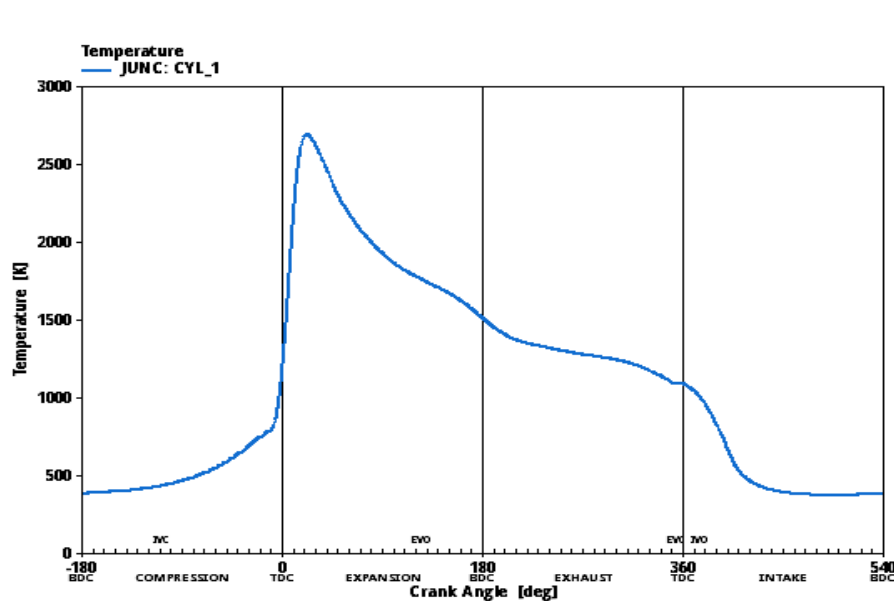


Figure 24 Ford Fox Temperature

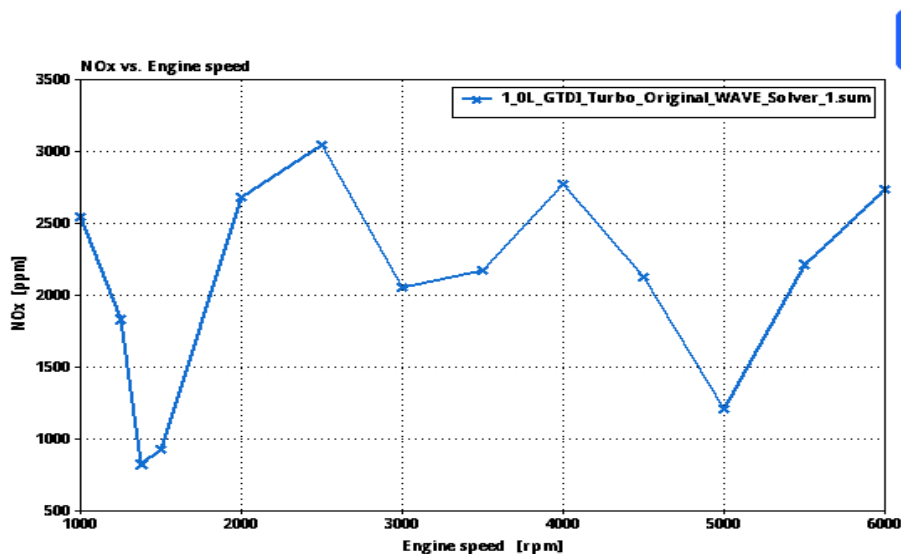


Figure 25 Ford Fox NO_x Emissions

The CO profile (Figure 26) instead reflects regions of incomplete oxidation. The high CO concentrations (~30,000–40,000 ppm at low rpm) coincide with fuelling strategies and mixture conditions that leave residual carbon species unoxidized. At higher speeds, CO falls toward ~10,000 ppm by 6,000 rpm, consistent with improved mixing, higher airflow, and shorter residence times. This trade-off between CO and NO_x is classic for SI engines: richer combustion suppresses NO_x by lowering flame temperature but produces excess CO, whereas closer to stoichiometric conditions reduce CO but promote NO_x.

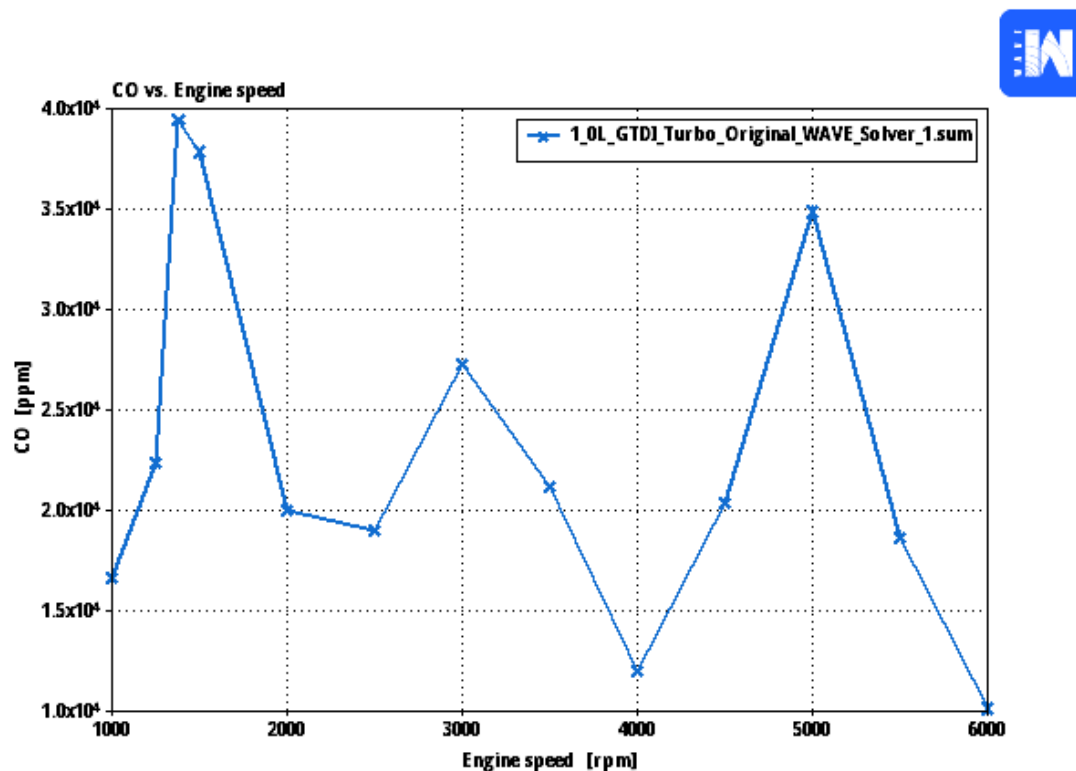


Figure 26 Ford Fox CO Emissions

Critically, the absence of aftertreatment or EGR in the Ricardo WAVE model exaggerates these engine-out values compared with production engines. In a real 1.0 L EcoBoost unit, a three-way catalyst (TWC) would reduce CO and NO_x by more than 90% under stoichiometric operation (Zhang et al., 2022). Similarly, cooled EGR could be used to suppress peak temperatures, lowering NO_x formation at source while maintaining combustion stability. The model therefore highlights the fundamental combustion–emissions trade-off but cannot be compared directly to legislated tailpipe figures without incorporating these technologies.

The large secondary peaks in NO_x around 4,000 rpm and 6,000 rpm suggest operating points where combustion phasing and in-cylinder temperature rise again, potentially linked to higher boost pressures and reduced knock margin. Without over boost control or advanced ignition

calibration in the model, these results may over-predict NO_x relative to the production engine, which would deliberately retard spark or enrich mixtures to protect against knock and reduce thermal loading.

The results are directionally correct and illustrate the direct coupling between in-cylinder temperature and NO_x formation, and between mixture richness and CO formation. However, they represent baseline, engine-out emissions and should be treated as diagnostic rather than predictive of compliance. The trends support the need for emissions-control technologies such as TWC, EGR, and lambda management, which in production smooth out the extreme spikes and bring emissions within Euro 6 limits.

4. Prechamber Model

The Ricardo WAVE model was further developed to incorporate a pre-chamber ignition system in place of the conventional single-spark configuration (Figure 27). In this arrangement, a small auxiliary chamber is ignited separately, producing high-energy jets of partially combusted gases that penetrate the main chamber and accelerate flame propagation. This strategy enables stable combustion under leaner air–fuel ratios, which lowers peak in-cylinder temperatures and can therefore suppress thermal NO_x formation compared with the baseline model. Literature suggests that pre-chamber systems not only extend lean-burn limits but can also deliver improvements in indicated efficiency of 1–3 percentage points, owing to faster and more complete combustion (Zhao et al., 2020). However, the approach introduces several trade-offs: incomplete oxidation of the fuel-rich jets may elevate CO and unburned hydrocarbon emissions at engine-out, and the system requires additional hardware, fuelling control, and calibration complexity (Bendu and Murugan, 2014). Compared with the baseline 1.0 L GTDI configuration, the pre-chamber concept is therefore expected to reduce NO_x by lowering combustion temperatures, improve thermal efficiency under lean operation, but at the cost of potentially increased CO and HC emissions without the application of a three-way catalyst.

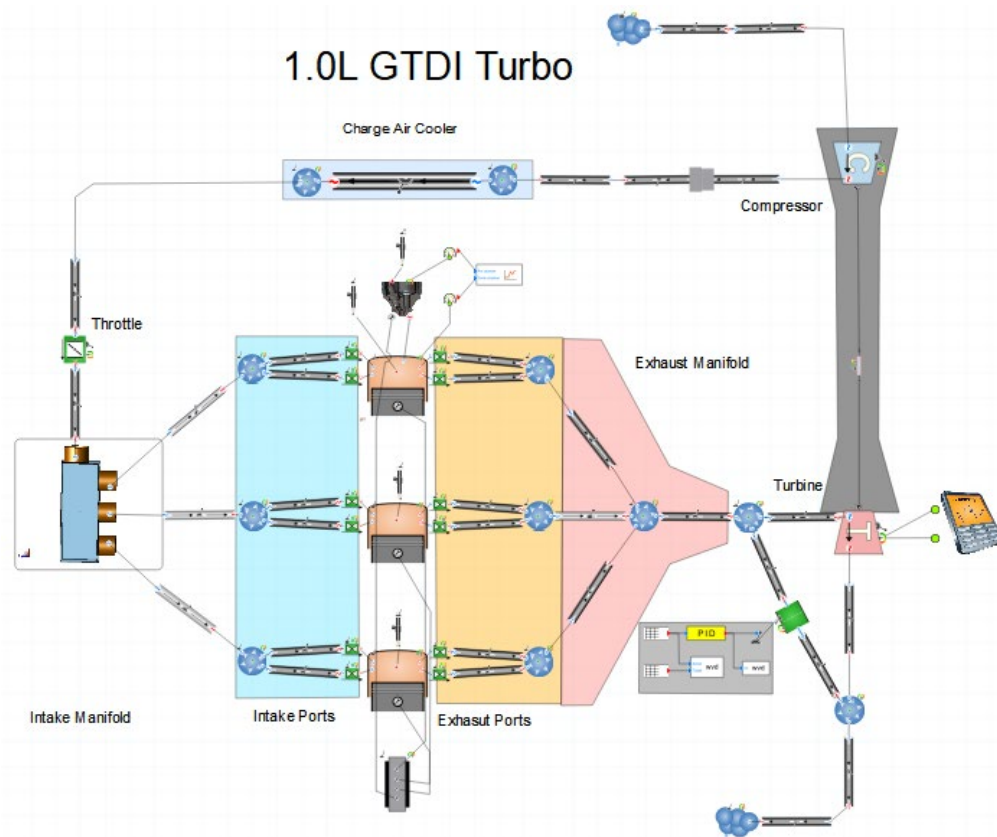


Figure 27 Prechamber fitted to Ford Fox Engine

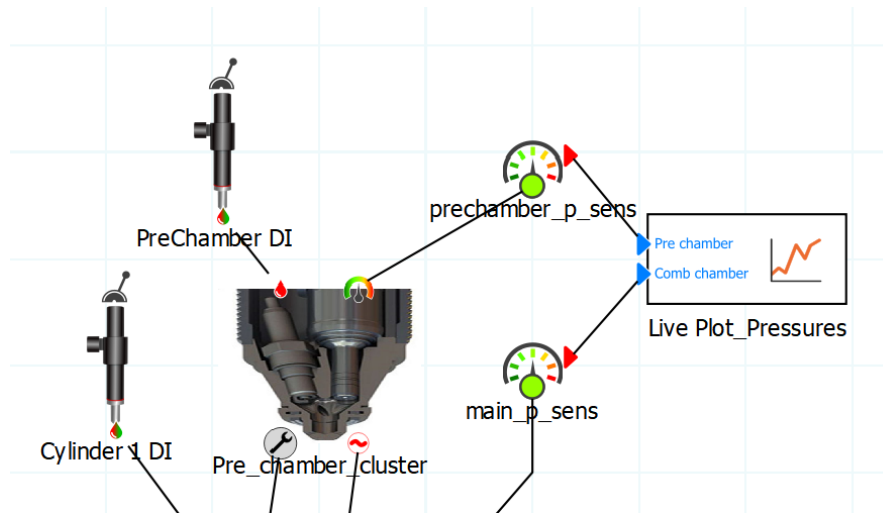


Figure 28 Prechamber Closer Look

Within the pre-chamber cluster (Figure 28), a position actuator is implemented to constrain piston motion within the pre-chamber domain. This modelling choice ensures that the pre-chamber volume remains fixed throughout the simulation, accurately reflecting the physical behaviour of pre-chamber ignition systems. In practice, pre-chambers are designed as quasi-stationary cavities that contain a small volume of mixture, which is ignited and then discharged into the main combustion chamber as turbulent jets. They do not undergo reciprocating motion, unlike the main cylinder volume. By fixing the piston position in the simulation, the model is therefore able to replicate this stationary behaviour and allow for a more realistic representation of ignition and jet formation processes. This approach is consistent with the literature, where pre-chambers are characterised as fixed-volume enclosures that enhance combustion stability and extend lean-burn operation (Mahesh et al., 2017).

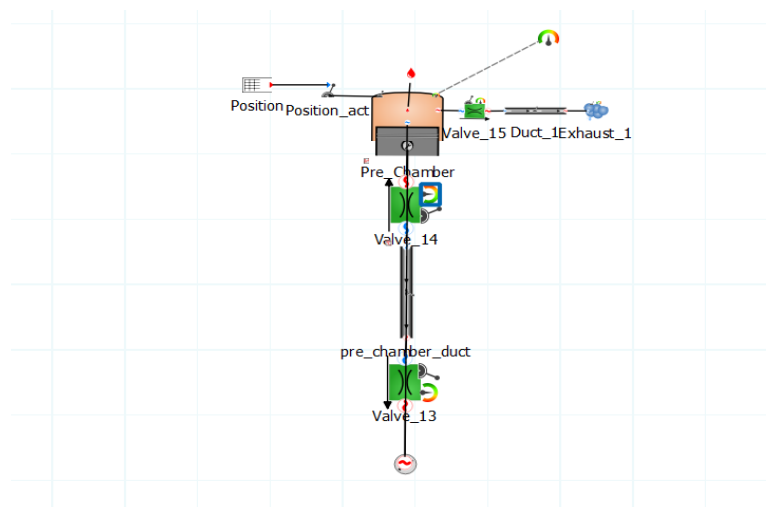


Figure 29 Inside Prechamber Cluster

4.1 Prechamber setup

4.1.1 Geometry

The active pre-chamber was represented as a 1.41 cm³ control volume ($\approx 1\text{--}2\%$ of clearance volume) coupled to the cylinder via calibrated restrictors. An auxiliary injector delivered 3–6% of the total cycle fuel to the pre-chamber, targeting $\phi_{pc} \approx 1.0$ at spark. Pre-chamber air mass was estimated from $m = \frac{PV}{RT}$ at pre-spark conditions, giving ≈ 20 mg air and $\approx 1.3\text{--}1.5$ mg gasoline per cycle at 30 bar / 750 K. A two-stage heat-release model captured the pre-chamber ignition and jet-driven main burn, with restrictor area and ϕ_{pc} tuned to reproduce the measured shoulder, peak pressure, and early expansion characteristics

4.1.2 Fuel calculation

A quick calculator was made within Wave as to quickly calculate fuel mass needed achieve a certain fuel equivalence value using the following formula:

$$m_{f,pc} [mg] \approx k(T_{pc}) \times V_{pc} [cc] \times P_{pc} [Bar]$$

Ideal gas law in Specific gas constant form $m_{a,pc} = \frac{P_{pc} V_{pc}}{R_{air} T_{pc}}$

Equivalence ratio:

$$\phi = \frac{\left(\frac{F}{A}\right)}{\left(\frac{F}{A}\right)_{stoich}} = \frac{m_f/m_a}{1/AFR_{stoich}}$$

So the fuel mass needed in the prechamber to achieve a target ϕ is:

$$m_{f,pc} = \phi_{pc} \frac{m_{a,pc}}{AFR_{stoich}}$$

$$m_{f,pc} = \phi_{pc} \frac{P_{pc} V_{pc}}{R_{air} T_{pc} AFR_{stoich}}$$

AFR_{stoich} is the stoichiometric air-fuel ratio by mass

$$R_{air} \cong 8314 \text{ J/Kmol}$$

4.2 Prechamber testing

The pre-chamber system was tested in Ricardo WAVE by representing the pre-chamber as an additional control volume coupled to the main combustion chamber through calibrated orifices. The geometry was defined with a bore of 10 mm and a length of 18 mm, giving a total pre-chamber volume of approximately 1.4 cm³ ($\approx 1\text{--}2\%$ of the clearance volume). This volume was connected to the main cylinder using restrictor elements, where the total effective discharge area and discharge coefficient were adjusted to reproduce realistic jet penetration and mixing behaviour.

An auxiliary fuel injector was specified for the pre-chamber volume. Initial fuelling was set to supply $\sim 3\text{--}6\%$ of the total cycle fuel to the pre-chamber, in line with values reported in the literature for active turbulent jet ignition systems (Mahesh et al., 2017). The pre-chamber mixture was targeted at stoichiometric conditions ($\phi_{pc} \approx 1.0$), while the main chamber operated under globally lean conditions ($\lambda > 1.5$). This ensured that the pre-chamber generated strong ignition jets, consistent with experimental observations.

4.2.1 Performance Graphs

The engine performance characteristics demonstrate broadly consistent behaviour with established combustion engine theory, with brake power and torque rising with speed and peaking at approximately 3500–4500 rpm before declining as frictional and gas-exchange losses dominate. Volumetric efficiency exceeds unity across the tested range, a phenomenon attributable to strong pressure-wave dynamics in the intake system as captured by the one-dimensional WAVE model. However, absolute values exceeding 200% are physically unrealistic and highlight the limitations of the model's wave action treatment. At low engine speeds (1000–2000 rpm), both torque and brake power predictions fall to near zero, with indications of negative torque. This anomalous behaviour is indicative of unstable combustion under low load fuelling conditions, consistent with poor mixture preparation within the pre-chamber system at these operating points.

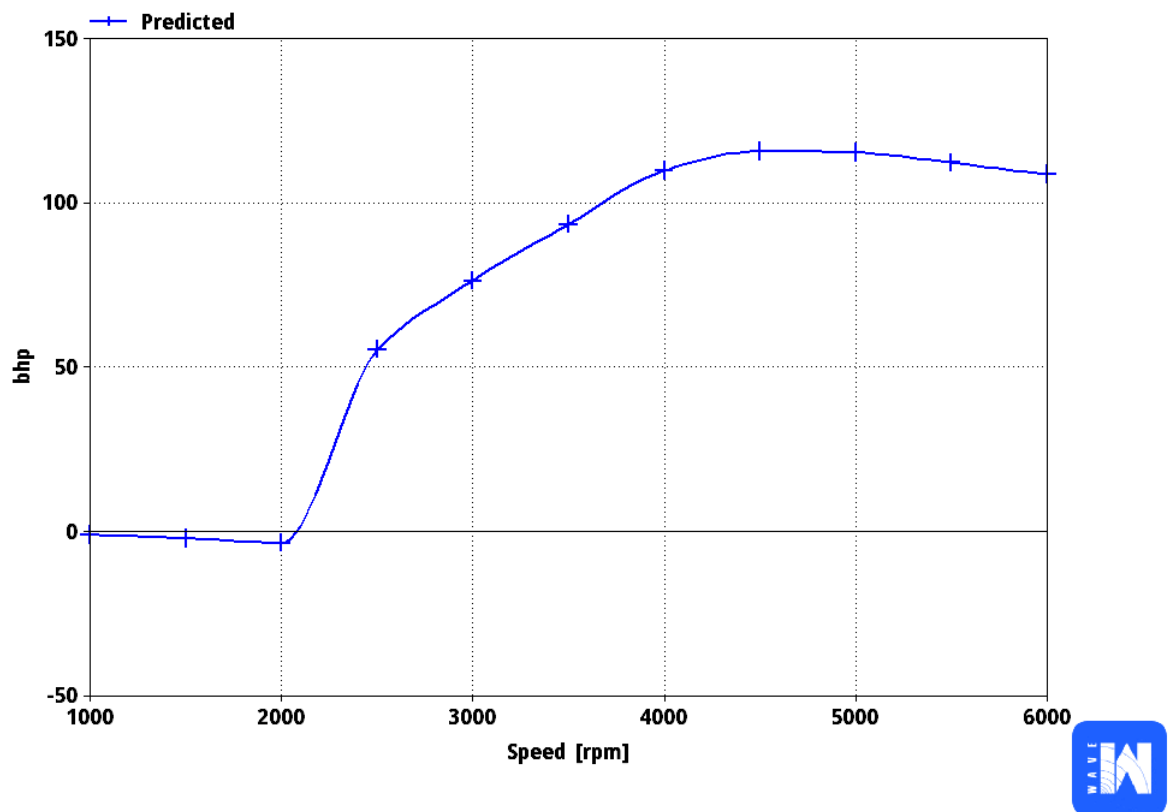


Figure 30 Bhp (PC)

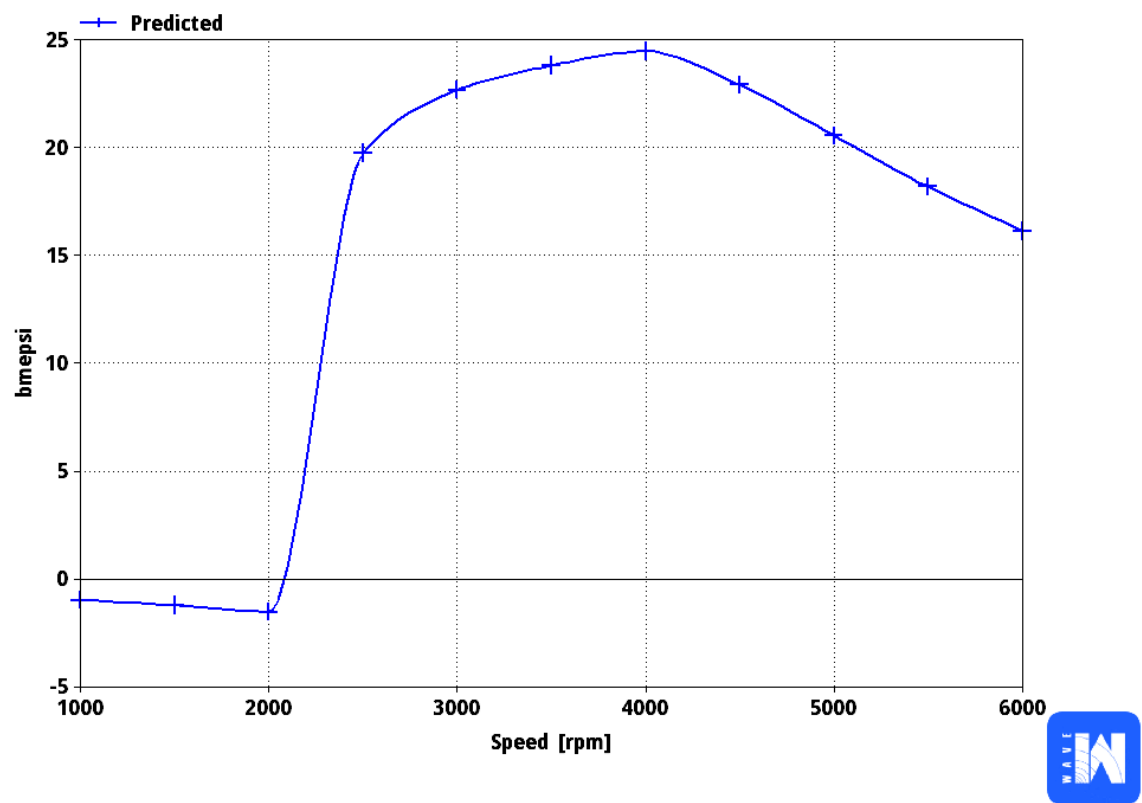


Figure 31 BMEP (PC)

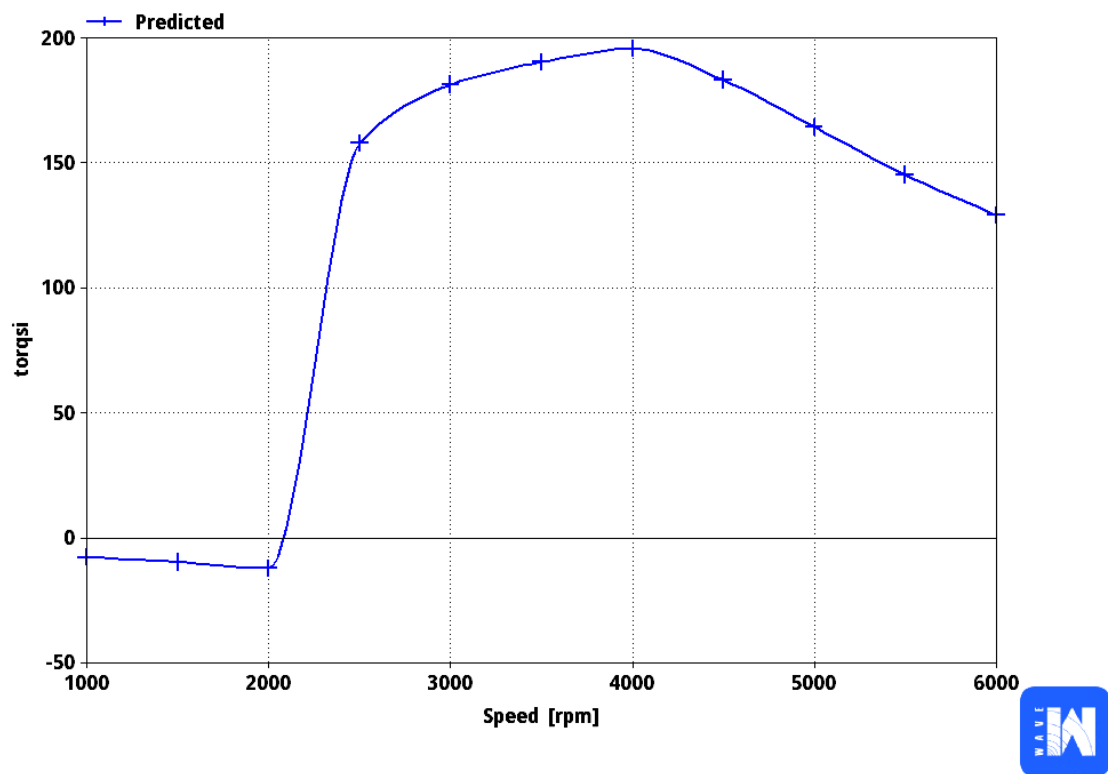


Figure 32 Torque (PC)

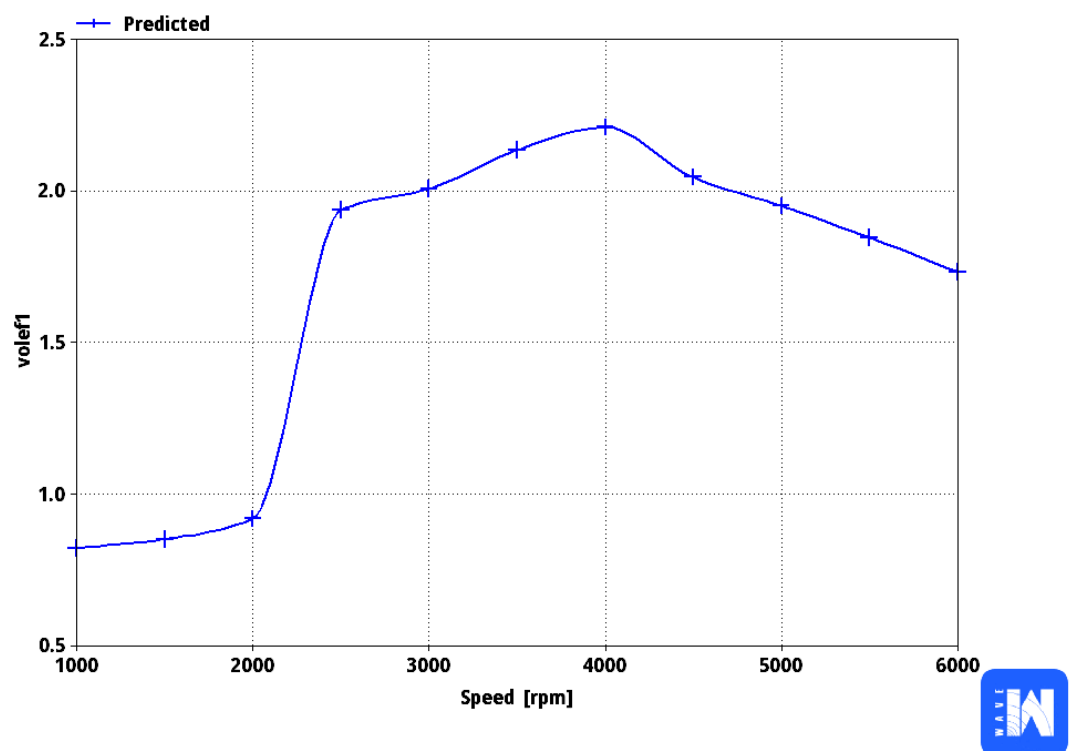


Figure 33 Volumetric Efficiency (PC)

4.2.2 Combustion Behaviour

The combustion-related parameters reflect the same inconsistencies observed in the performance outputs. Indicated mean effective pressure (IMEP) peaks above 25 psi near 4000 rpm, consistent with the torque maximum, before declining with further increases in speed. Friction mean effective pressure (FMEP) follows the expected rising trend with rpm due to mechanical and pumping friction but appears overestimated in the 3000–4000 rpm region, reducing the net brake output efficiency. Pumping MEP (PMEP) values are strongly negative beyond 3000 rpm, consistent with high intensity scavenging flows and the inflated volumetric efficiency predictions discussed previously. The 10–90% heat release duration curve exhibits unusually long combustion durations ($>20^\circ\text{CA}$) at 2000–3000 rpm, indicating delayed and unstable combustion. At higher speeds, combustion duration stabilises, reflecting improved in-cylinder mixing and pre-chamber performance. These findings again suggest that fuelling inconsistencies at low-speed compromise combustion stability, while high-speed predictions remain more robust.

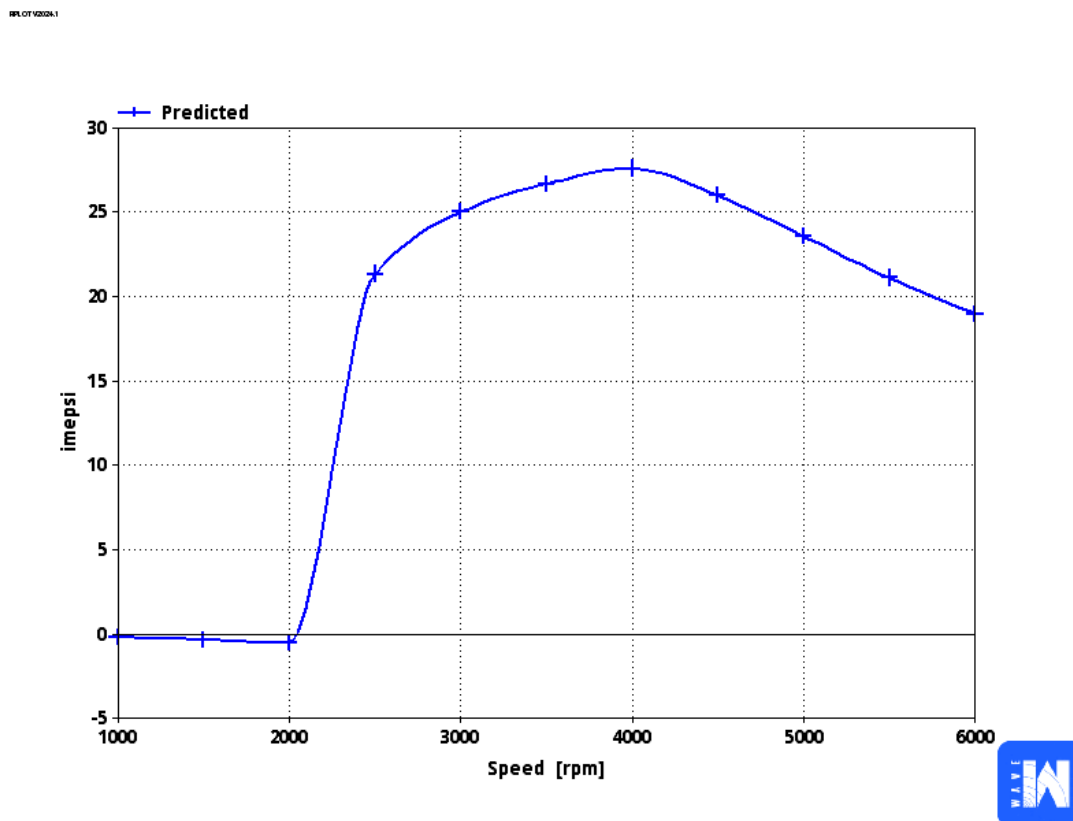


Figure 34 IMEP (PC)

WAVE

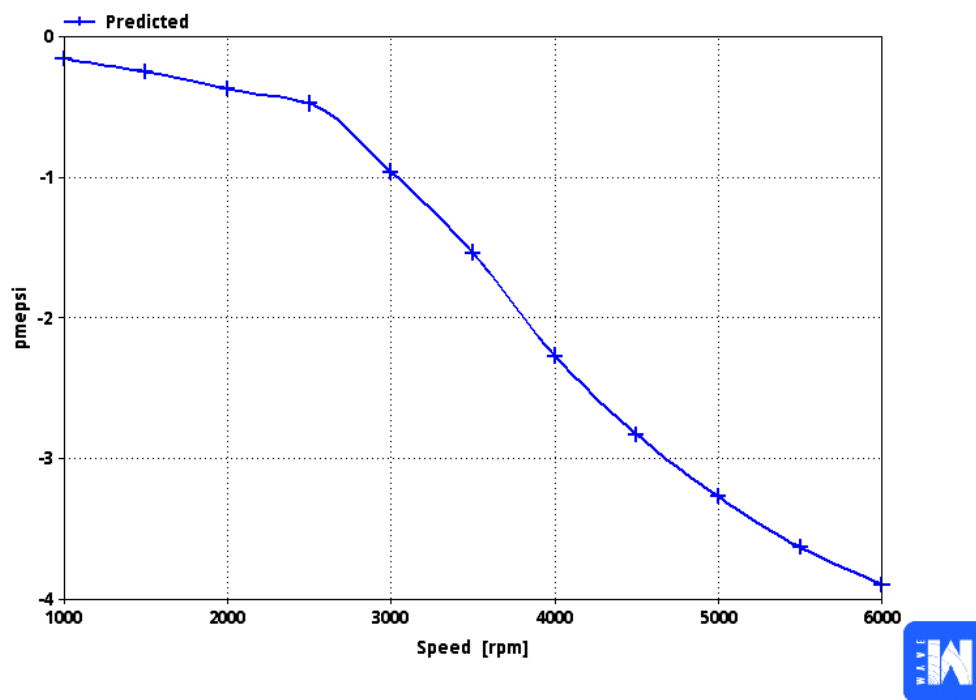


Figure 35 PMEP (PC)

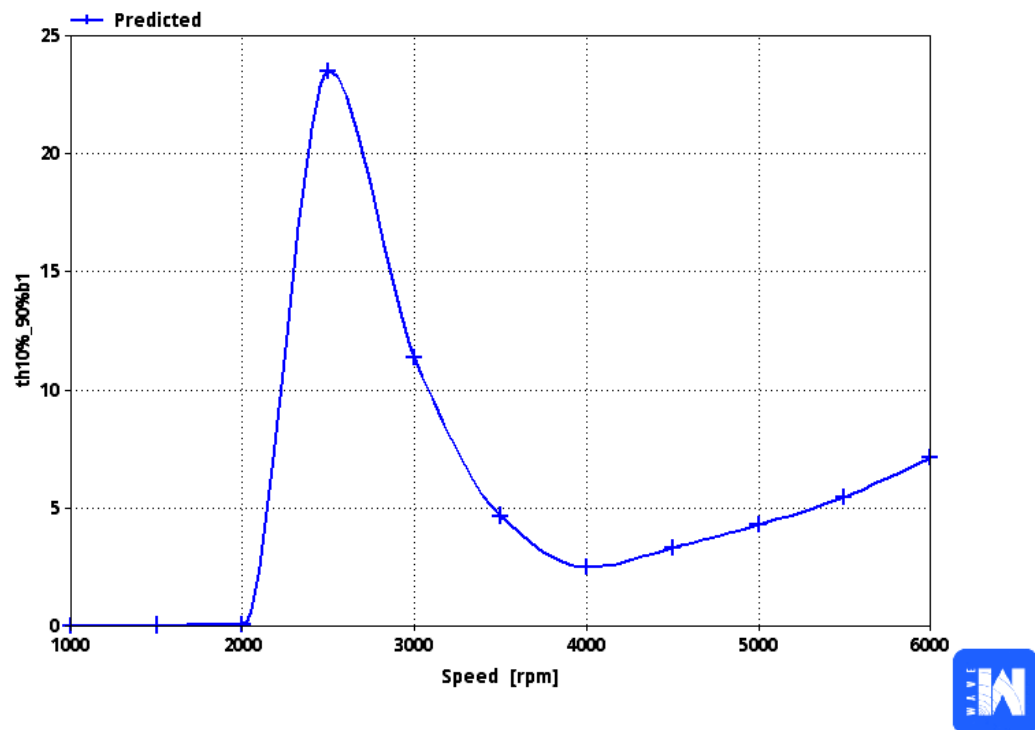


Figure 36 CA 10-90 (PC)

4.2.3 Fuel-air behaviour

Analysis of the fuel–air behaviour provides the clearest evidence of fuelling-related issues. The air–fuel ratio (AFR) and equivalence ratio (λ) show substantial deviations from expected stoichiometric values at 1000–2000 rpm, with AFR dropping below 3 and λ below 0.2. Such conditions correspond to an over-rich mixture that would not sustain stable combustion, explaining the poor torque and IMEP predictions in this regime. Beyond 2500 rpm, AFR and λ values recover to physically reasonable levels, with λ approaching unity by 5000 rpm, aligning with the assumed stoichiometric value of 14.02 for the selected fuel. Similarly, fuel mass flow predictions display sharp discontinuities around 2000–3000 rpm, further highlighting limitations in the fuelling strategy implemented in the pre-chamber. The results strongly suggest that the observed low-speed anomalies are attributable to inaccurate fuelling predictions, whereas higher-speed operation is captured with greater fidelity.

REF: COT120204.1

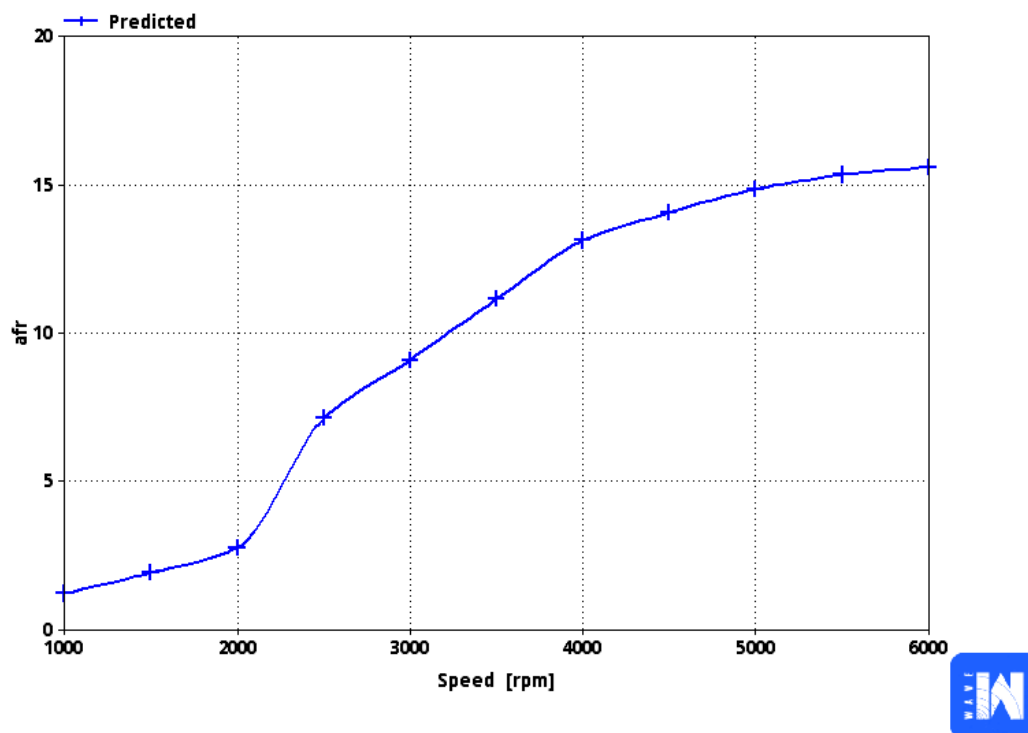


Figure 37 AFR Cylinder 1 (PC)

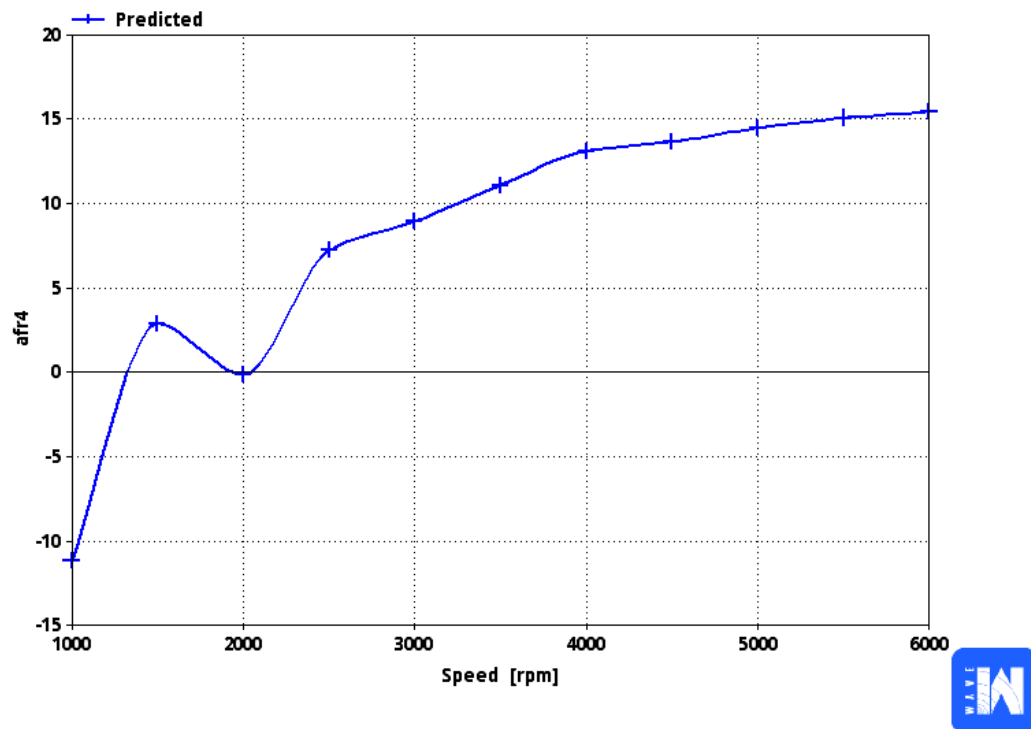


Figure 38 AFR Prechamber (PC)

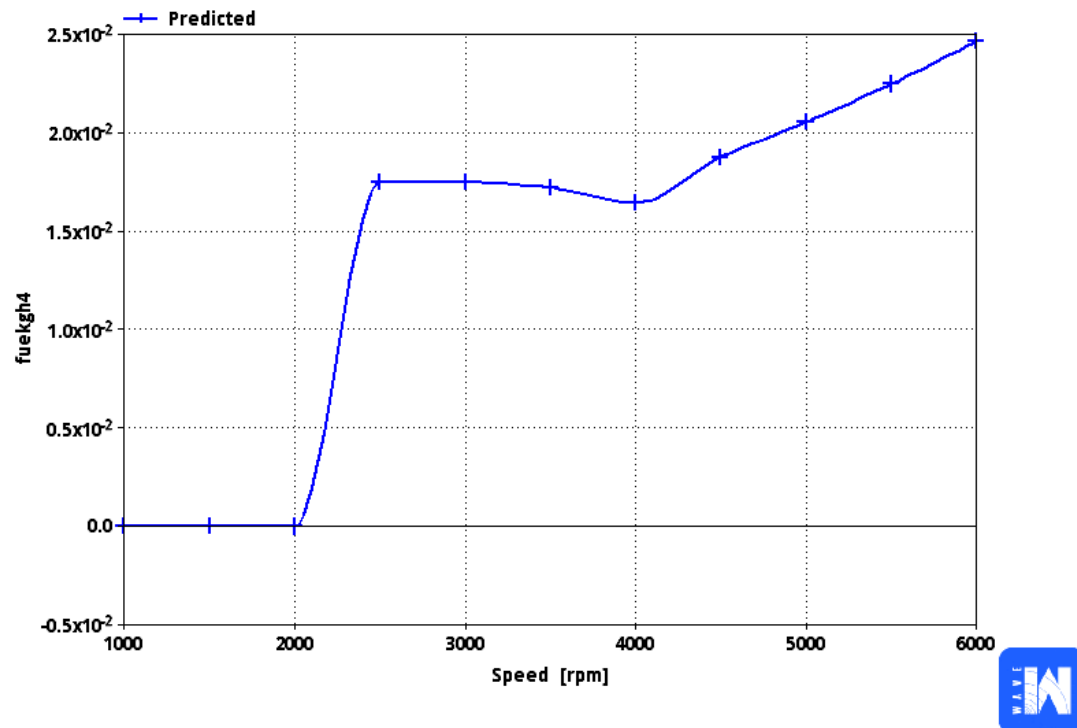


Figure 39 Fuel mass flowrate (PC)

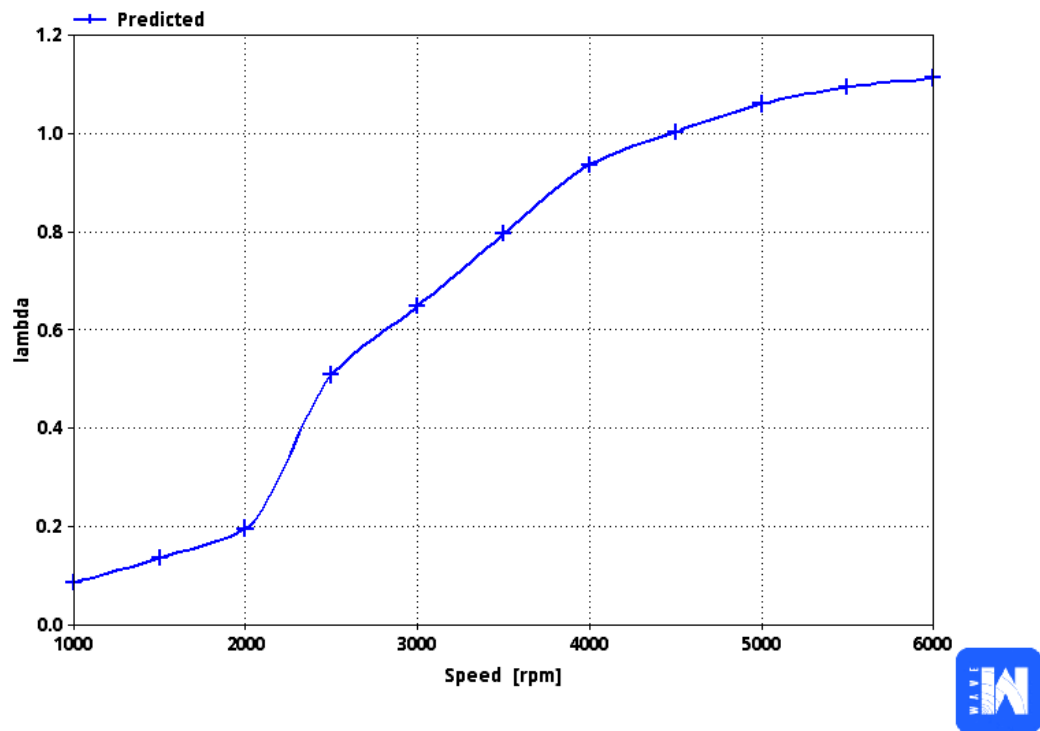


Figure 40 Lambda (PC)

4.2.4 Model Validation

The model validation metrics provide quantitative support for these conclusions. Coefficient of determination (R^2) values are consistently above 0.99 for AFR, λ , and torque across most of the speed range, indicating strong agreement between the design of experiments (DOE) surrogate models and the baseline simulation. However, localised deviations at low engine speeds are apparent, with R^2 values dropping significantly for AFR4 and IMEP, reflecting the combustion instability identified in the performance and combustion plots. Root mean square error (RMSE) results further substantiate these findings, with large spikes observed at 1000–2000 rpm for AFR and IMEP, while errors at medium-to-high speeds remain negligible. Together, these metrics demonstrate that the DOE methodology itself is robust and that the discrepancies at low rpm arise from limitations in the fuelling and combustion sub-models rather than from statistical inaccuracy.

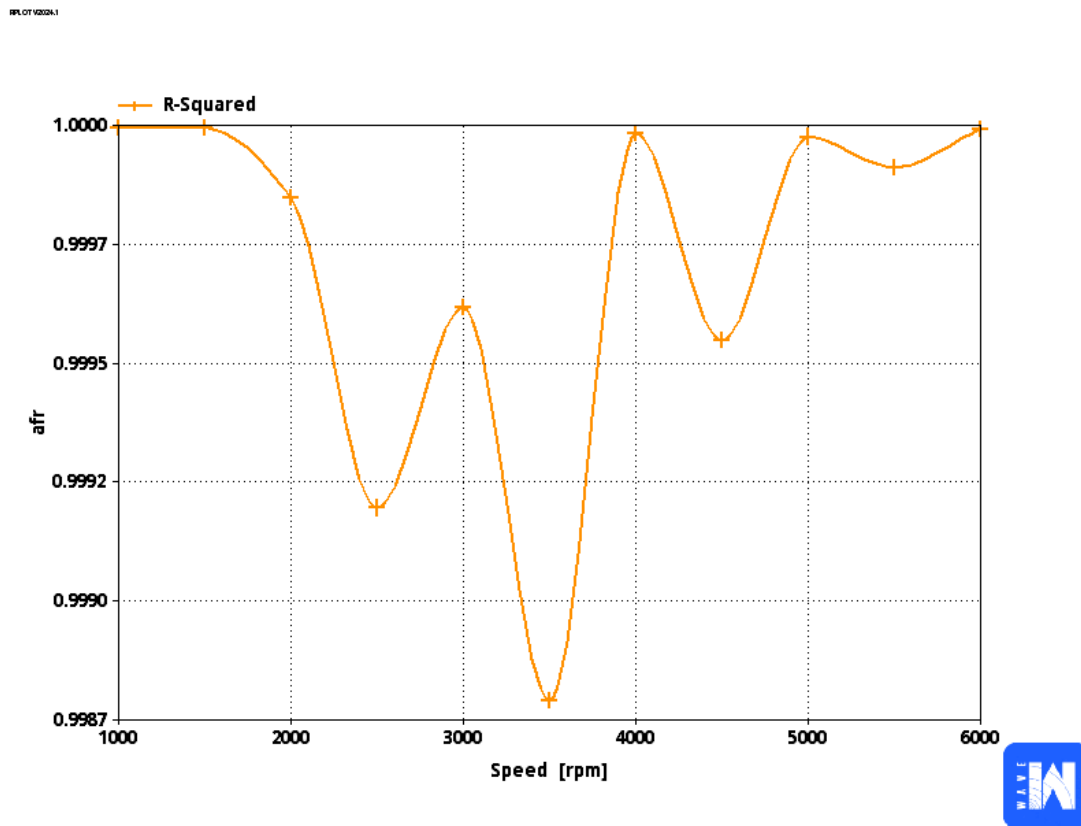


Figure 41 AFR R2

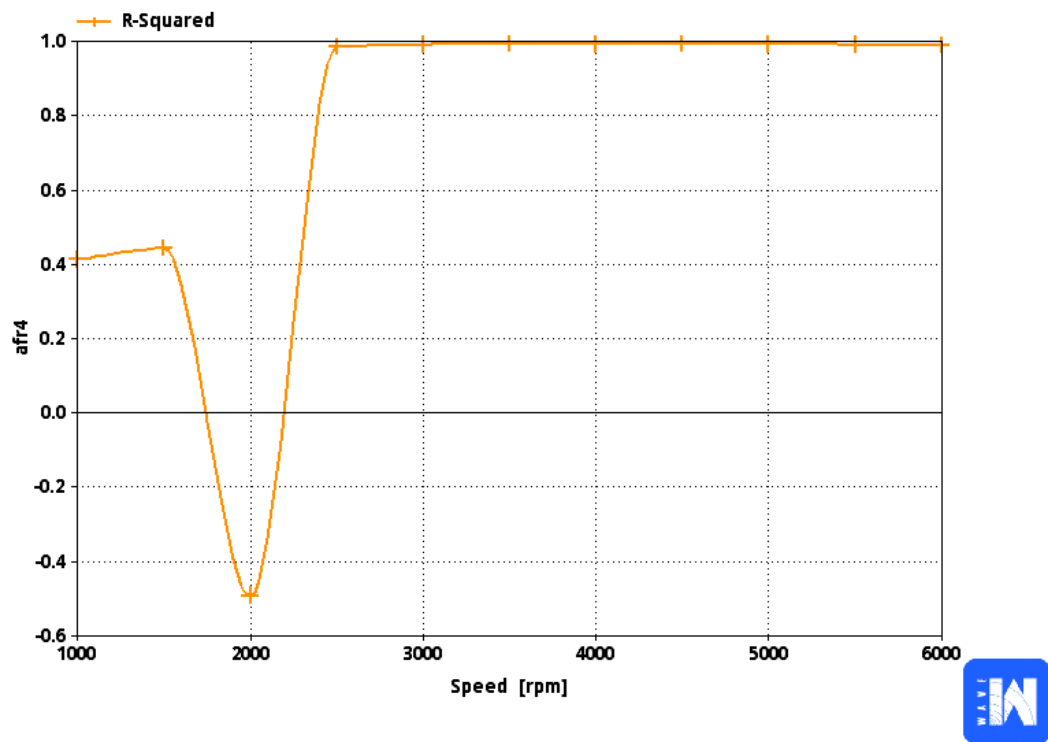


Figure 42 Prechamber AFR R2

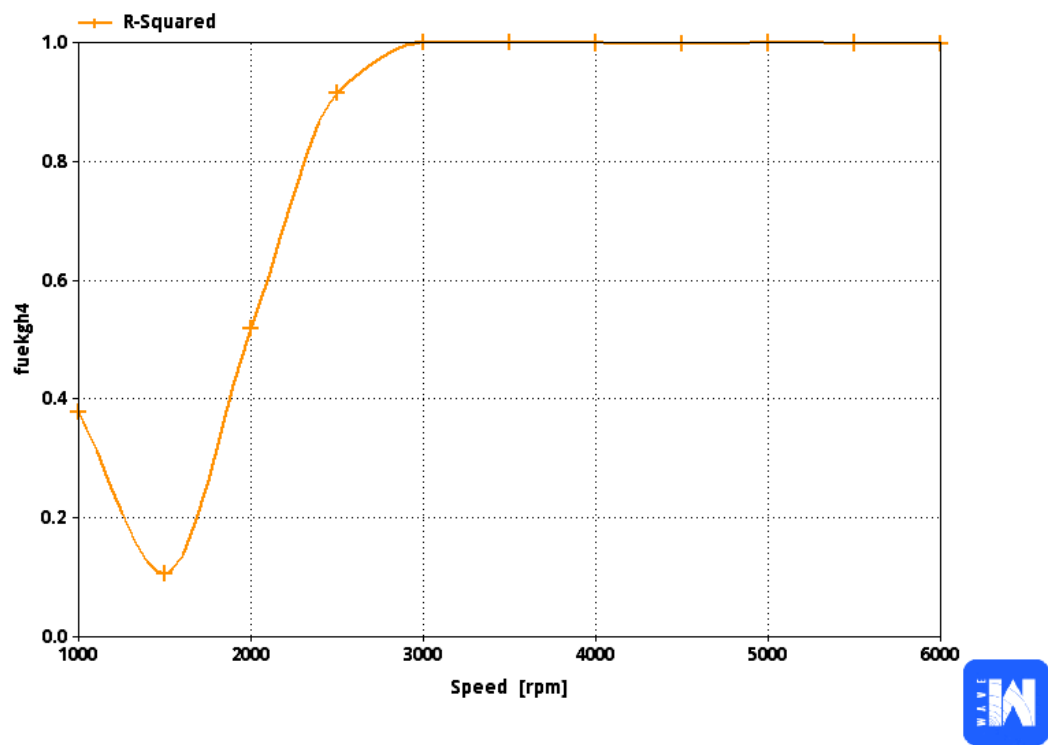


Figure 43 Fuel mass flowrate R2



Figure 44 IMEP R2

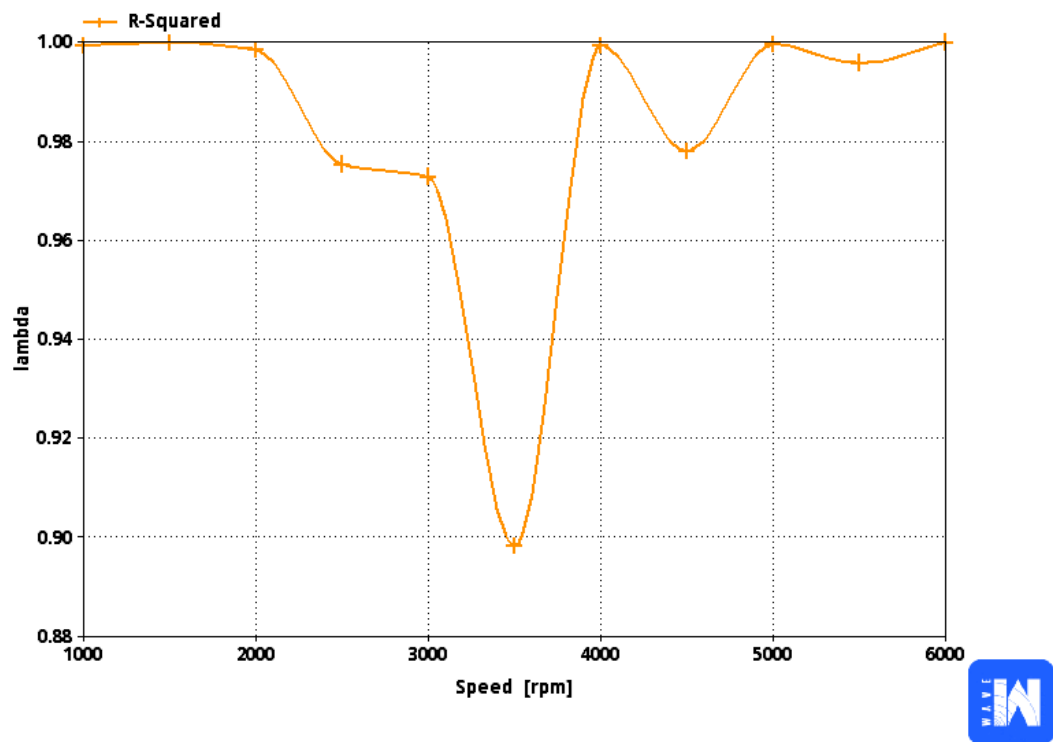


Figure 45 Lambda R2



Figure 46 PMEP R2

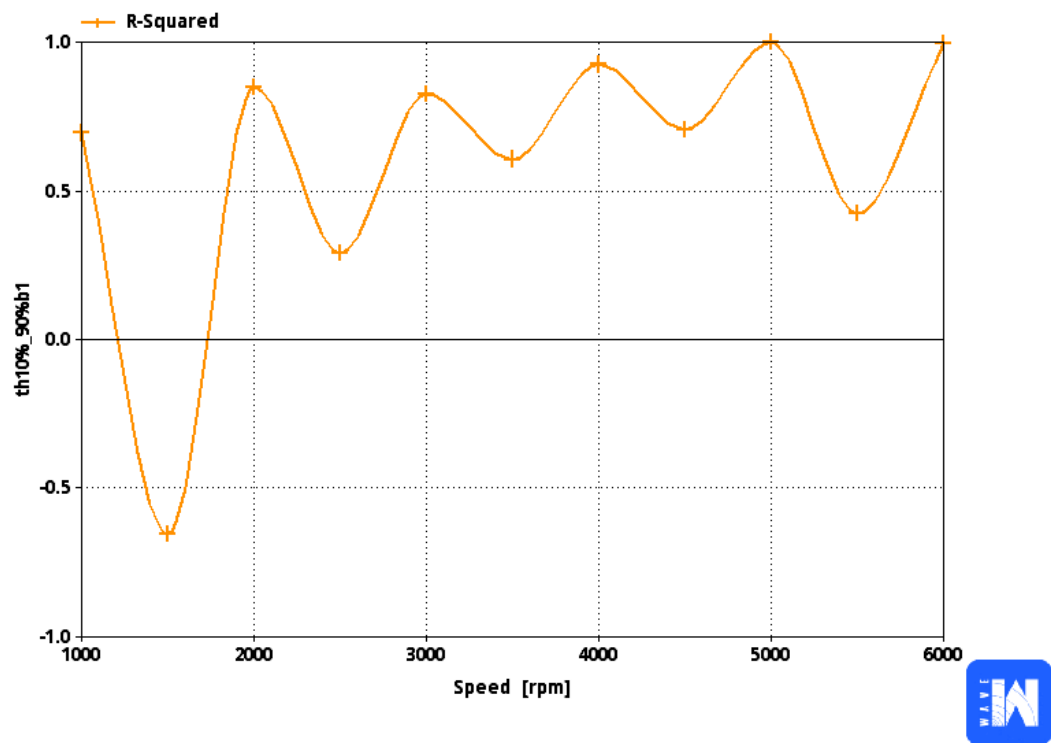


Figure 47 CA 10-90 R2

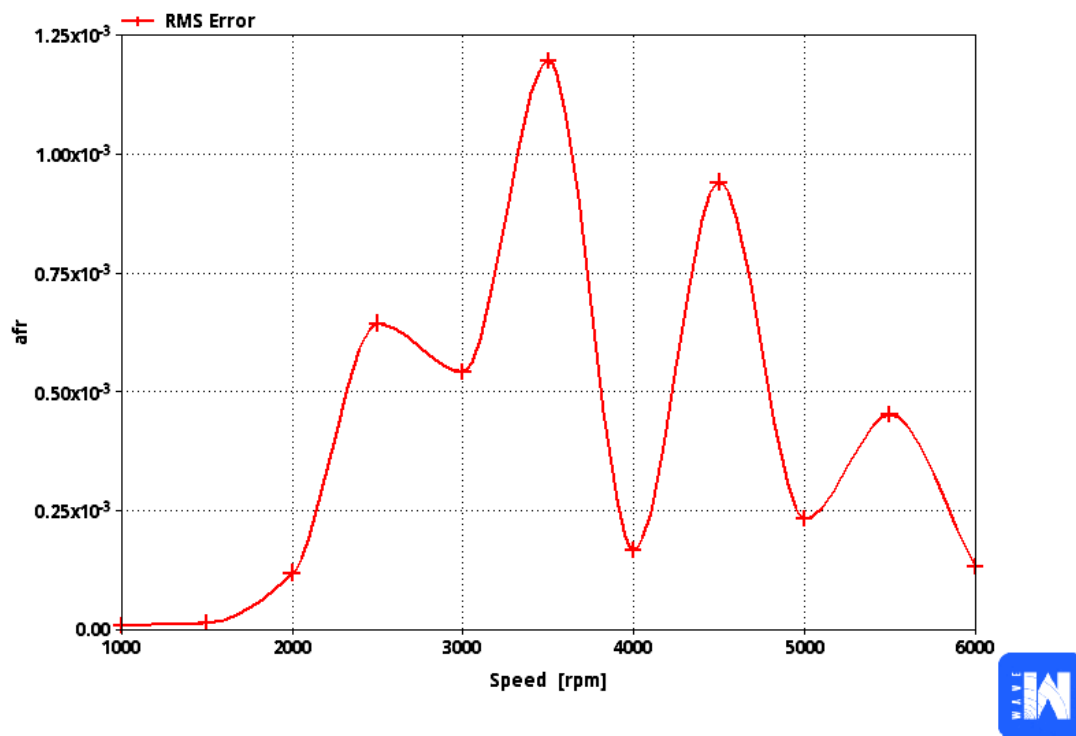


Figure 48 AFR RMSE

RPLOT120204.1

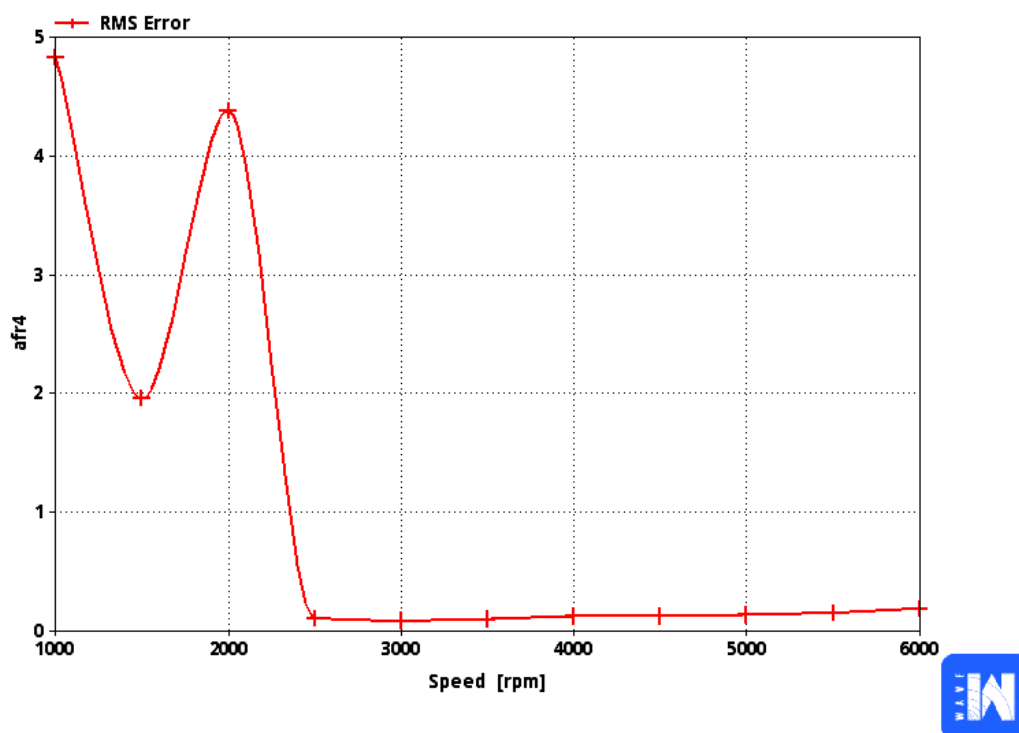


Figure 49 Prechamber AFR RMSE



Figure 50 Fuel mass flowrate RMSE

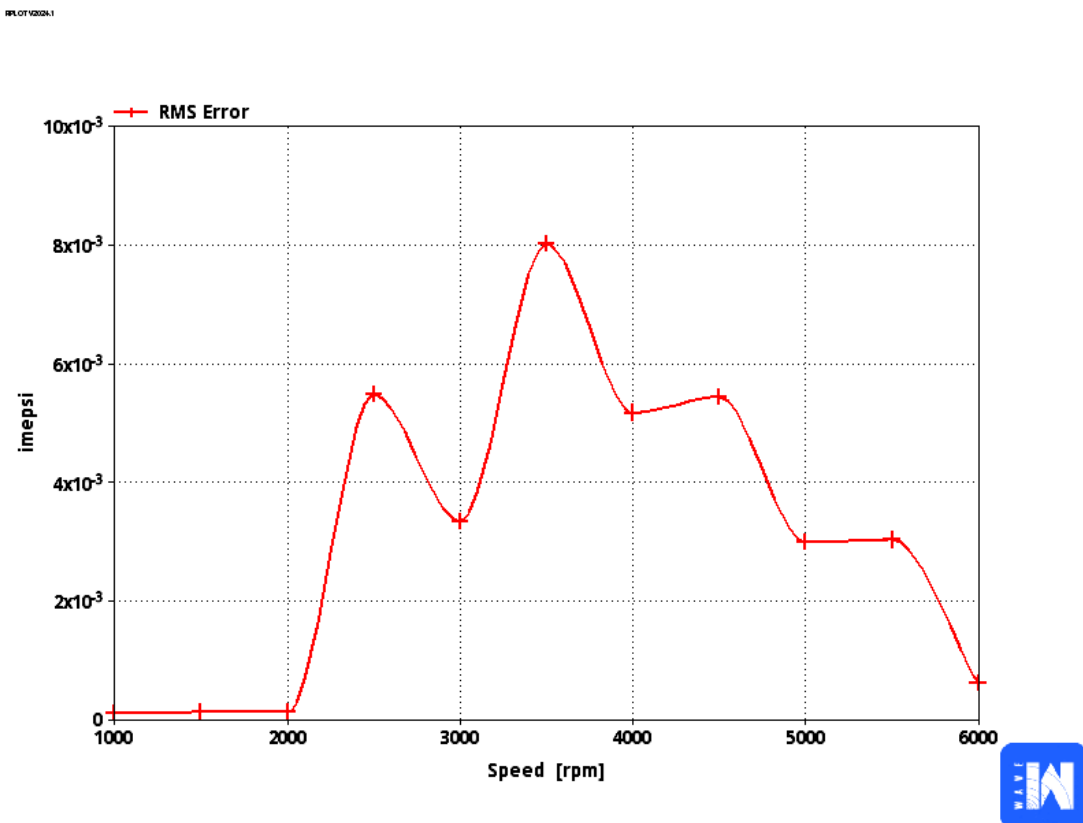


Figure 51 IMEP RMSE

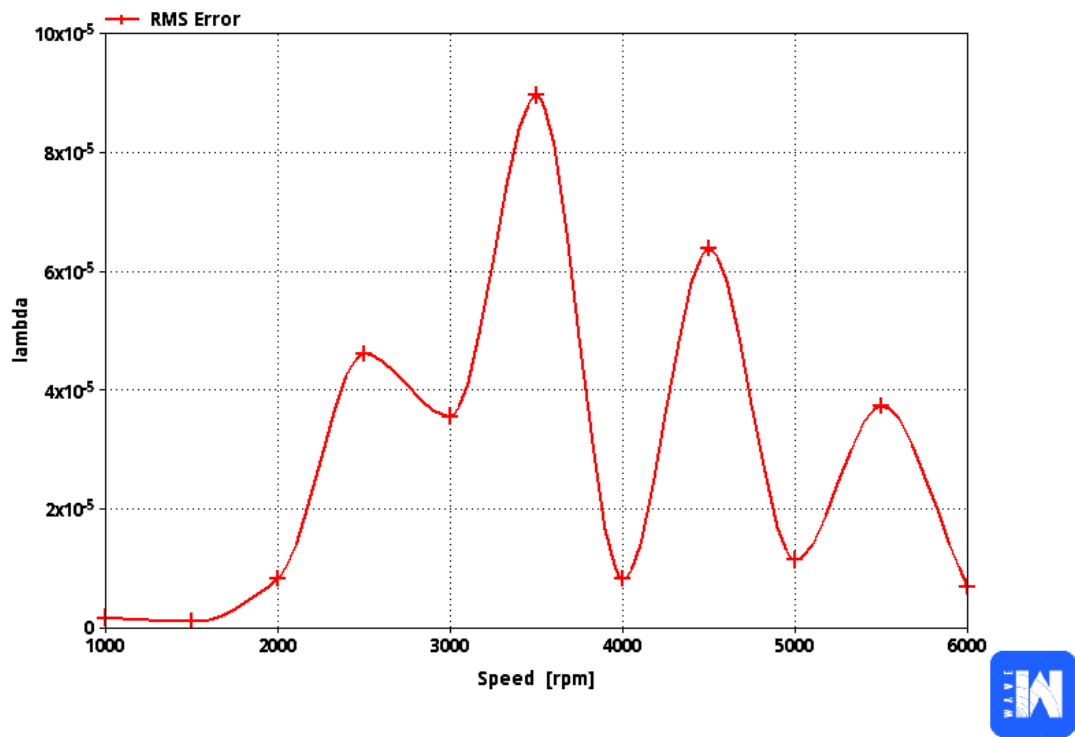


Figure 52 Lambda RMSE

RPL01102024.1



Figure 53 PMEP RMSE

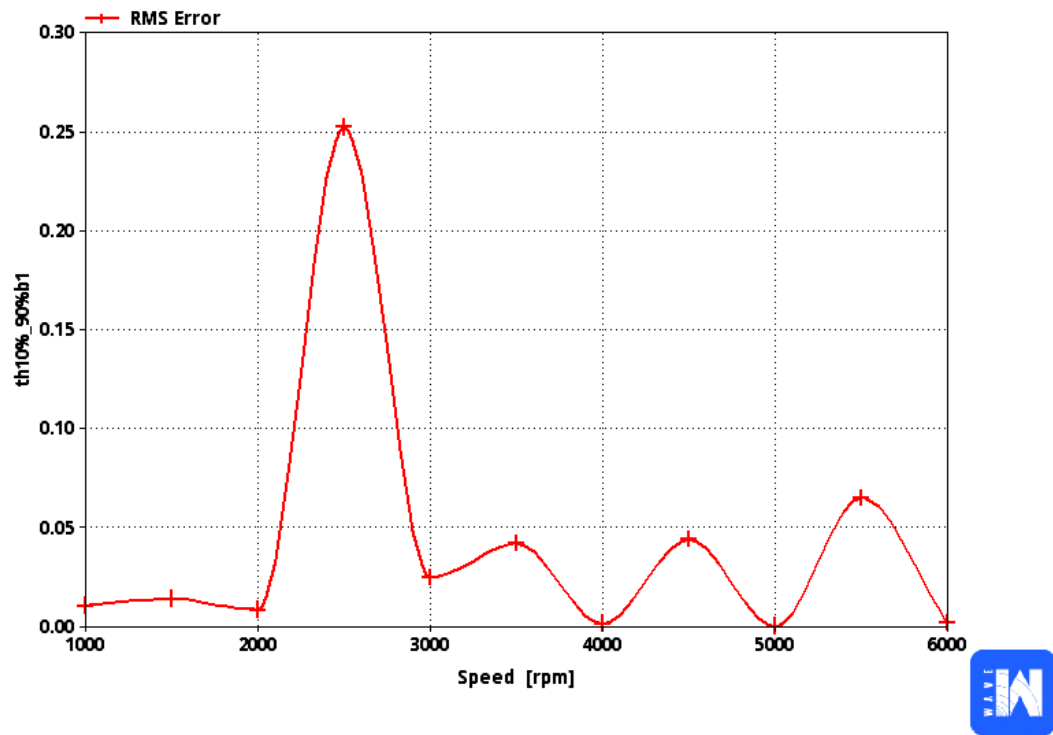


Figure 54 CA 10-90 RMSE

5. Prechamber analysis

5.1 Prechamber Results

The pre-chamber simulations highlight both the promise and limitations of the retrofit configuration. A notable outcome was the ~ 15 bar increase in peak cylinder pressure compared with the baseline spark-ignition case. This behaviour is consistent with the expected effect of turbulent jet ignition, where multiple ignition sites from the pre-chamber jets accelerate flame propagation and enhance combustion efficiency. The sharper pressure rise suggests improved thermodynamic efficiency, with the potential for leaner operation at equivalent torque output.

However, the performance graphs reveal inconsistencies at low speeds (1000–2000 rpm). Torque and brake power drop sharply, in some cases approaching negative values, while volumetric efficiency predictions exceed 200%. These results are physically unrealistic and point to shortcomings in the pre-chamber fuelling strategy and wave dynamics sub-model within Ricardo WAVE. At these operating points, the air–fuel ratio in both the cylinder and pre-chamber deviates significantly from stoichiometric, with λ values below 0.2. Such an over-rich mixture would not sustain stable combustion in practice, which explains the unstable and delayed burn behaviour (10–90% burn durations $>20^\circ\text{CA}$ at 2000–3000 rpm).

At higher speeds (>2500 rpm), the model predictions stabilise. AFR and λ values return to physically reasonable ranges, combustion durations shorten, and IMEP trends align more closely with theoretical expectations. This indicates that, under well-mixed conditions and with sufficient in-cylinder turbulence, the pre-chamber configuration behaves as intended, improving ignition robustness and overall combustion stability.

Critically, while the model demonstrates the qualitative benefits of pre-chamber ignition, the quantitative fidelity is limited by simplified assumptions: single-zone fuelling estimates, Wiebe-based heat release, and the absence of detailed turbulence–jet interaction. These limitations exaggerate volumetric efficiency, misrepresent fuelling transitions, and underpredict the sensitivity of combustion stability at low load. Thus, while the observed pressure gain validates the concept in principle, further refinement of fuelling control and multi-zone combustion modelling would be required before firm conclusions on efficiency or emissions benefits can be drawn.

6. Conclusion

This dissertation has investigated the modelling and validation of a Ford Fox engine using Ricardo WAVE, alongside the implementation and assessment of a pre-chamber ignition system. Validation against experimental cylinder pressure traces showed good overall agreement, particularly in the compression and expansion phases, with peak pressure predicted within a few bar of measurements. The two-zone combustion model provided a more detailed representation of heat release compared with a single-zone approach, but it still tended to overpredict the sharpness of combustion and failed to capture the pre-peak shoulder observed experimentally. This highlights the ongoing challenge of representing multi-stage heat release in simplified engine models and suggests the need for further refinements, such as double-Wiebe or multi-zone approaches, to more accurately capture combustion phasing.

The integration of a pre-chamber system within the Ricardo model demonstrated the potential of Turbulent Jet Ignition (TJI) to improve efficiency and lean-burn capability. Simulations produced elevated peak pressures and accelerated combustion at higher engine speeds, consistent with the literature on distributed ignition sites improving flame propagation and robustness under lean conditions. However, unrealistic fuelling behaviour and volumetric efficiency predictions at low loads revealed the constraints of the model's fuelling and jet representation. These limitations underline the need for refined sub-models that account for turbulence-jet interactions, fuel stratification, and cycle-to-cycle variability to ensure quantitative reliability.

The literature review further demonstrated that TJI represents a significant advancement in spark-ignition combustion technology. By enabling ultra-lean, stable operation ($\lambda > 2$) and achieving peak indicated efficiencies above 50% in research engines (Miao et al., 2025), TJI offers a pathway for spark-ignition engines to approach compression-ignition levels of performance while maintaining fuel flexibility. Even modest implementations can deliver 20–30% efficiency improvements and substantial NO_x reductions (Crosse, 2020; Nicklowitz et al., 2024). Importantly, TJI also provides a practical solution for overcoming ignition challenges associated with zero-carbon fuels such as ammonia and hydrogen, positioning it as a critical enabler in the transition toward net-zero internal combustion technology.

In conclusion, the modelling and analysis conducted here confirm both the promise and the practical challenges of TJI-equipped engines. Ricardo WAVE has proven to be a powerful tool for exploring pre-chamber concepts, though further refinements are needed for full predictive accuracy. When combined with the broader trajectory of research, these findings underscore that TJI can serve as a bridge between conventional SI engines and future low- or zero-carbon

combustion modes. By enhancing efficiency, reducing emissions, and enabling alternative fuels, TJI-equipped engines have the potential to play a key role in decarbonising transport and power generation sectors, particularly in hybrid powertrains, motorsports, and heavy-duty applications. The evidence to date strongly supports the view that, when properly designed and implemented, turbulent jet ignition delivers the fast, efficient, and clean combustion required to meet global carbon-neutral objectives.

Future improvements

While the present work has successfully validated a Ricardo WAVE model of the Ford Fox engine and demonstrated the potential of turbulent jet ignition (TJI), several areas for

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improvement remain. Future work can be grouped into short-, medium-, and long-term priorities:

Short-Term (model refinement)

Combustion modelling: Extend the current two-zone heat release approach with double-Wiebe or multi-zone formulations to capture the pre-peak shoulder and realistic CA10–90 combustion durations.

Low-speed operation: Resolve fuelling anomalies at 1000–2000 rpm by refining injector characterisation and implementing closed-loop λ targets for both cylinder and pre-chamber domains.

Gas exchange accuracy: Revisit intake/exhaust wave-action parameters to eliminate volumetric efficiency inflation (>200%) and cross-check PMEP predictions against measured flow and pressure data.

Medium-Term (prechamber optimisation)

Geometry and fuelling: Conduct systematic sweeps of nozzle diameter, number, and orientation, alongside pre-chamber fuel fraction mapping, to balance ignition robustness with heat-loss minimisation.

Purge strategies: Implement residual-scavenging or purge-air models to address pre-chamber quenching and durability concerns highlighted in the literature.

Multi-point validation: Extend baseline validation to pre-chamber-enabled cases across the speed/load range to ensure transferability of the model.

Long-Term (system integration and emissions)

Advanced sub-models: Incorporate CFD-informed jet–turbulence coupling and knock prediction to improve predictive fidelity under high-load and high-pressure conditions.

Aftertreatment and EGR: Add simple emissions-control models (e.g., TWC, EGR) to evaluate engine-out vs. tailpipe outcomes and capture NO_x/CO trade-offs under lean-burn operation.

Uncertainty quantification: Report validation results with statistical envelopes (RMSE/MAPE bands across crank angle) to strengthen the robustness of model–experiment comparisons.

By addressing these areas, the Ricardo WAVE model can evolve from a concept-validation tool into a quantitatively reliable platform for pre-chamber design, calibration, and emissions optimisation. This progression will be essential to assess the full potential of TJI as a net-zero enabling technology in practical engine systems.

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Appendix

Ford Fox Engine specification

Engine Parameter	Specification
Engine Type	3 Cylinder Gasoline Direct Injection
Engine Parameter	DOHC, VCT
Bore x Stroke / mm	71.9 x 82.0
Displacement	999 cc
Compression Ratio	10.0:1
Cylinder Head	Cast Aluminium
Cylinder Block	Cast Iron
Camshaft Drive	Belt
Max Power @ RPM	125 @ 6000 RPM
Max Torque @ RPM	170 lb-ft @1400 RPM (up to 200 lb-ft with transient over boost)

Mean Absolute Percentage Error Calculation

$$MAPE = \frac{100}{n} \sum_{i=1}^n \left| \frac{y_i^{model} - y_i^{exp}}{y_i^{exp}} \right|$$

y_i^{model} = model value at data point i

y_i^{exp} = experimental value at data point i

N = number of data points

Arrhenius equation

The Arrhenius equation describes the exponential dependence of the rate of constant of a chemical reaction on the absolute temperature.

$$k = Ae^{\frac{-E_a}{RT}}$$

K is the rate constant

T is the absolute temperature

A is the pre-exponential factor

E_a is the molar activation energy for the reaction

R is the universal gas constant

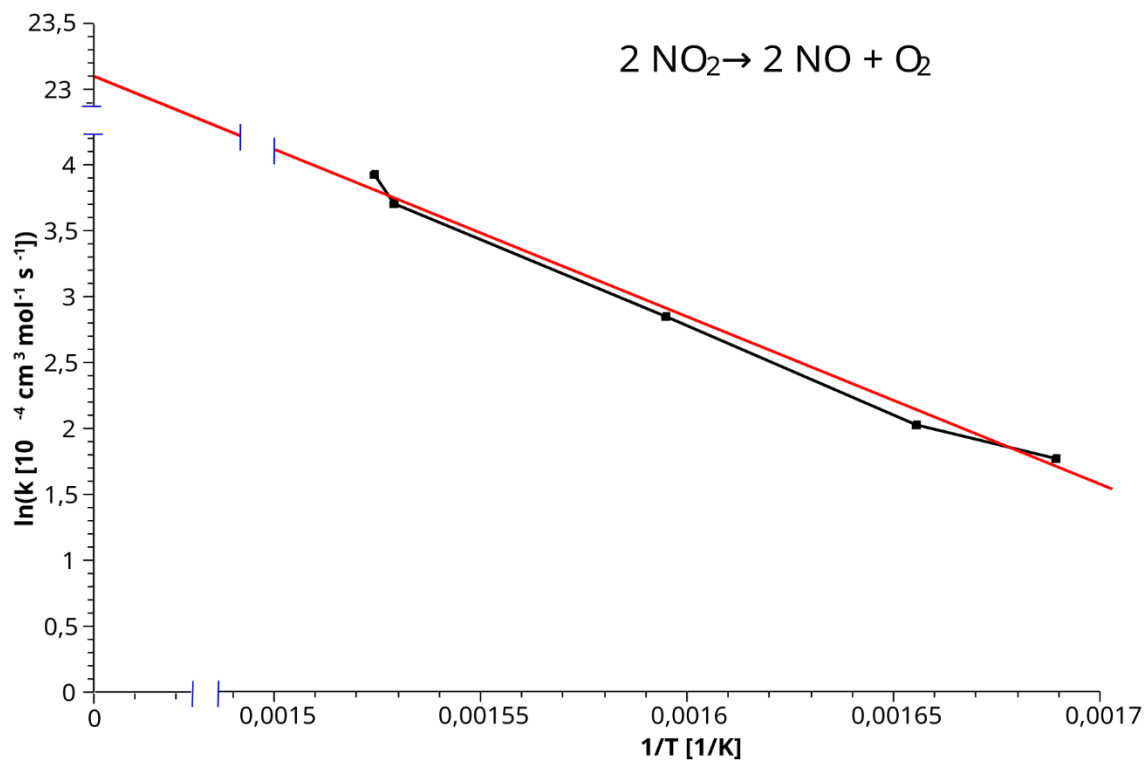
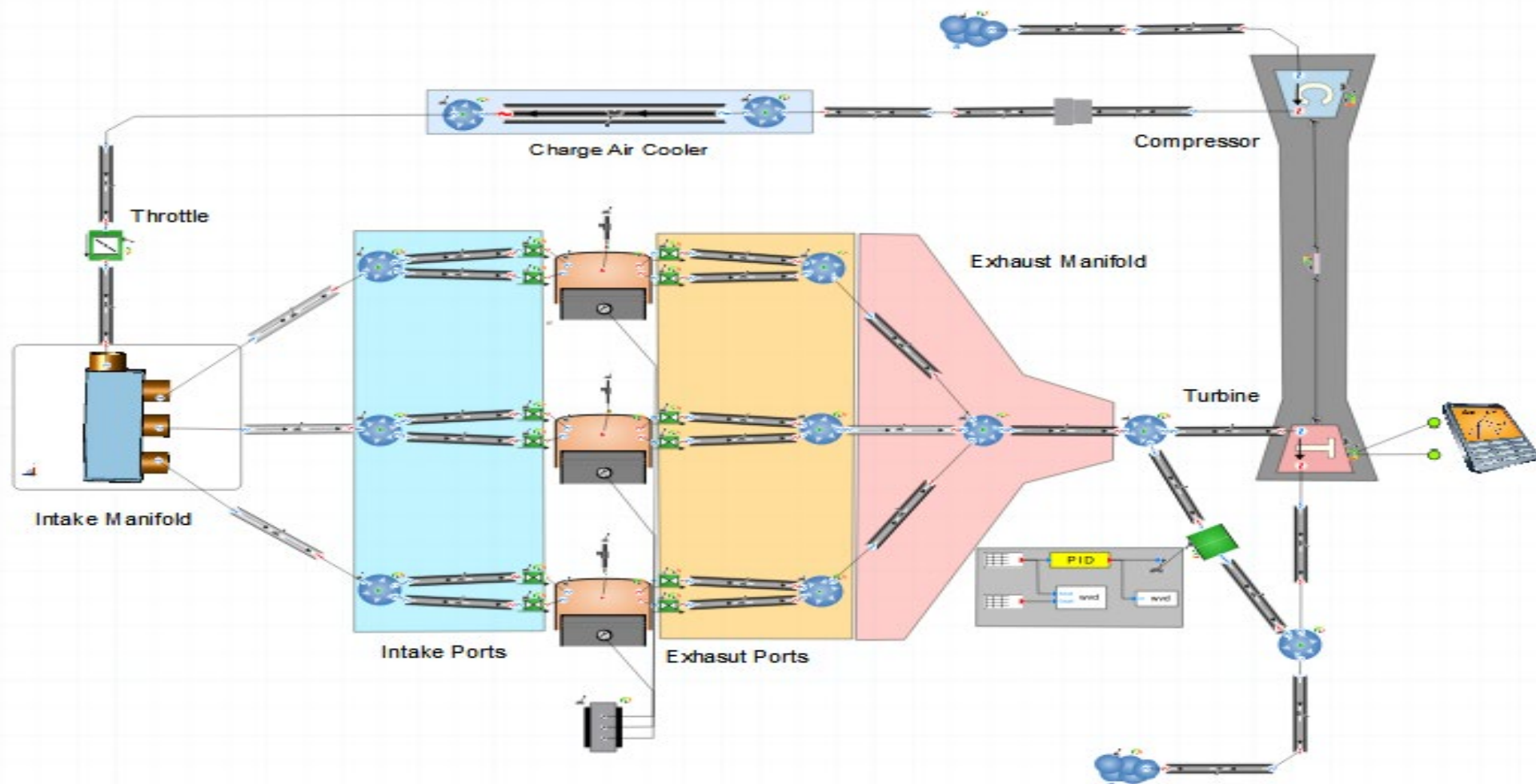


Figure 55 Arrhenius function graphed (Zivilverteidigung, M. 2013)

1.0L GTDI Turbo



FUELMASS kg/h

	800	1000	1250	1500	1750	2000	2250	2500	3000	3500	4000	4500	5000	5500	6000	6500
FL		3.24	5.22	7.52	9.39	11.35	12.92	14.34	18.08	18.92	21.54	22.34	25.55	26.61	27.62	25.11
200						11.34	12.96	14.25	17.78	19.47						
190						10.77	12.04	13.39	16.91	18.46						
180					9.11	10.05	11.26	12.52	15.24	17.76	20.61					
170				7.51	8.77	9.38	10.52	11.72	14.06	17.14	18.91	21.41				
160				7.10	8.08	8.74	9.77	10.88	13.05	15.06	17.46	19.80	23.68			
150				6.52	7.46	8.17	9.12	10.19	11.98	14.00	16.18	18.15	20.99	24.47		
140			5.19	6.04	6.84	7.63	8.45	9.44	11.01	12.93	14.87	16.65	18.92	21.63	25.58	
130			4.73	5.46	6.16	6.96	7.73	8.55	10.05	11.89	13.84	15.40	17.30	19.71	22.93	
120			4.35	4.93	5.75	6.39	7.07	7.79	9.15	10.95	12.86	14.24	15.88	18.00	20.45	23.30
110		3.22	3.91	4.52	5.24	5.78	6.43	7.13	8.42	9.97	11.65	13.06	14.58	16.31	18.66	20.90
100		2.97	3.43	4.13	4.66	5.20	5.78	6.48	7.71	9.03	10.53	11.88	13.38	15.00	16.78	19.21
90		2.56	3.00	3.60	4.11	4.55	5.13	5.77	6.90	8.05	9.36	10.70	12.11	13.59	15.15	16.95
80		2.20	2.66	3.15	3.59	4.05	4.55	5.13	6.11	7.13	8.39	9.56	10.95	12.32	13.64	15.25
70		1.89	2.33	2.77	3.21	3.60	4.07	4.56	5.40	6.32	7.45	8.44	9.71	10.92	12.18	13.69
60		1.63	2.02	2.42	2.78	3.12	3.54	4.02	4.75	5.54	6.50	7.39	8.51	9.60	10.76	12.04
50		1.38	1.73	2.08	2.38	2.70	3.03	3.41	4.09	4.80	5.60	6.41	7.31	8.25	9.33	10.42
40	1.04	1.16	1.46	1.76	2.01	2.28	2.57	2.87	3.46	4.10	4.77	5.48	6.27	7.10	7.94	8.92
30	0.89	0.95	1.19	1.42	1.67	1.88	2.13	2.36	2.85	3.41	3.94	4.55	5.21	5.94	6.62	7.51
20	0.61	0.73	0.93	1.10	1.30	1.48	1.65	1.84	2.21	2.67	3.13	3.64	4.18	4.76	5.37	6.08
10	0.44	0.51	0.65	0.78	0.93	1.05	1.16	1.29	1.63	1.92	2.38	2.72	3.18	3.64	4.18	4.79
5	0.36	0.42	0.53	0.63	0.79	0.88	0.98	1.09	1.32	1.60	1.99	2.33	2.73	3.14	3.56	4.08
2	0.32	0.39	0.46	0.56	0.66	0.78	0.86	0.98	1.22	1.44	1.83	2.09				

