



Charge Dilution Effects in a Direct-Injection
Gasoline Engine with Advanced Spark Ignition
and Valvetrain Systems

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Abstract

Current legislative trends and customer demand drive the need for clean and efficient vehicles. Homogenous lean combustion is widely recognised within the industry as the viable mean of pushing efficiency of gasoline engines to the territory which was previously reserved for higher compression diesel engines. Diluted mixtures are thermodynamically very attractive, however they present challenges related to limited ignitability and often require assistance to realise their theoretical potential. Such assistance can be related to either enhanced flow motion inside the cylinder or a higher energy ignition system.

Experimental work was carried using the single cylinder engine test facility which was designed and commissioned in partnership with the project sponsor – Jaguar Land Rover.

The aim was set on understanding the limitations of a current combustion system, and implementing advanced technology packages in the proof of concept studies to increase engine tolerance towards diluted mixtures.

An advanced valvetrain system with capabilities of continuously variable valve lift was used to assess effects of deactivating one of the intake valves. It was found that increased TKE at ignition led to significant improvements in combustion stability and shorter burn durations, however the benefits were largely offset by heat transfer penalty. Heat transfer effects were quantified at various speeds and loads to provide comprehensive understanding of implementing the valve deactivation strategy. An optimisation study followed based on the findings and staggered lift was used on one of the valves. By utilising advanced, optimised valvetrain strategies dilution tolerance was significantly increased and indicated thermal efficiency of the engine surpassed 40%.

Prototype high energy ignition systems from two suppliers were also tested with the use of very high sampling rate data acquisition system built for the purpose of this work. It was found that significant benefits in dilution tolerance could be achieved, especially when combining the valvetrain and ignition systems potential.

Declaration of Originality

I, Pawel Okroj, declare that the work presented in this thesis is my own. Any materials taken from other sources are appropriately referenced.

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Nomenclature

Abbreviations

AFR	Air/Fuel Ratio
aTDC	After Top Dead Centre
bTDC	Before Top Dead Centre
BDC	Bottom Dead Centre
BMEP	Brake Mean Effective Pressure
BW	Borg Warner
CA	Crank Angle
CEI	Controlled Electronic Ignition
CFD	Computational Fluid Dynamics
CLA	Chemilluminescent Analyser
CO	Carbon Monoxide
CO ₂	Carbon Dioxide
CoP	Coil on Plug
CoV	Coefficient of Variance
CR	Compression Ratio
CVVL	Continuously Variable Valve Lift
DCP	Dual Coil Pack
DOE	Design of Experiments
DOHC	Double Over Head Cam
DCI	Dual Coil Ignition
DI	Direct-Injection
DISI	Direct-Injection Spark-Ignition
DSG	Dynamometer Services Group

DVA	Dual Valve Actuation
ECU	Engine Control Unit
EGR	Exhaust Gas Recirculation
EMOP	Exhaust Maximum Opening Point
EVC	Exhaust Valve Closed
EIVC	Early Intake Valve Closure
EVO	Exhaust Valve Open
FID	Flame Ionisation Detector
FFID	Fast Flame Ionisation Detector
GFR	Gas/Fuel Ratio
GMEP	Gross Mean Effective Pressure
GSFC	Gross Specific Fuel Consumption
HC	Hydrocarbon
ICE	Internal Combustion Engine
IMEP	Indicated Mean Effective Pressure
IMOP	Intake Maximum Opening Point
IVC	Intake Valve Closed
IVO	Intake Valve Open
JLR	Jaguar Land Rover
LIVC	Late Intake Valve Closure
LIVO	Late Intake Valve Opening
MFB	Mass Fraction Burned
MPA	Magnetopneumatic Analyser
NDIR	Non-Dispersive Infra-Red
NMEP	Net Mean Effective Pressure
NO	Nitrogen Oxide

NO _x	Oxides of Nitrogen
NSFC	Net Specific Fuel Consumption
PDF	Probability Density Function
PFI	Port Fuel Injection
PRT	Platinum Resistance Thermometer
Re	Reynolds Number
RMS	Root Mean Squared
SCRE	Single Cylinder Research Engine
SI	Spark-Ignition
SOI	Start of Injection
SVA	Single Valve Actuation
SwRI	Southwest Research Institute
TDC	Top Dead Centre
TDP	Thermal Dilution Parameter
THC	Total Unburned Hydro-Carbons
WOT	Wide Open Throttle

Greek Symbols

η	Efficiency [%]
λ	Air-fuel Equivalence Ratio
μ	Dynamic viscosity [(N s)/m ²]
ρ	Density [kg/m ³]
ϕ	Fuel-Air Equivalence Ratio
γ	Ratio of Specific Heats – C _P /C _V
ψ	Mole fraction

Subscripts

C	Compression
E	Expansion
f	Firing
ge	Gas exchange
m	Motoring
v	Volumetric

Chapter 1

Introduction and Literature Review

1.1. Preliminary Remarks

Since its introduction in 1876 the Spark-Ignition (SI) engine proves to be the most convenient, affordable and reliable mean for powering passenger and sports/ performance cars all over the world. Due to its efficiency of converting chemical energy into motive power, which is currently at about 35% for modern Direct-Injection (DI) engines, and well-established fuel delivery infrastructure, it can be said with high degree of confidence that SI engines will continue to be the source of power generation in transport for the foreseeable future.

Since the break of the Volkswagen emissions scandal (also known as “dieselgate”) in 2015, media coverage of legislative efforts into lowering emissions of road transport has been more pronounced than ever, whilst car manufacturers are subjected to tighter controls. European Commission estimates that light-duty vehicles are responsible for around 15% of the EU's emissions of CO₂. The only solution for years to come, is to improve efficiency and produce cleaner engines. Current legislation focuses on tackling the air pollution, and recent years have shown that gasoline and hybrid (combining gasoline and electric motors) cars steadily gain their market share compared to diesel powered vehicles. SMMT (2019) reported that gasoline

cars registrations (as percentage of all new passenger vehicles registered in the UK) rose from 47.57% in January 2015 to 64.08% in January 2019.

Recent efforts within engine development in automotive industry are mainly focused on challenges related to lowering fuel consumption, whilst meeting increasingly stringent exhaust emission standards.

European emission standards (Euro6 introduced in September 2014) put limits on the following tailpipe pollutants: oxides of nitrogen (NO_x), carbon monoxide (CO), unburned hydrocarbons (THC) and particulate matter (PM).

European Commission (2019) set 2021 target, which states that the fleet average to be achieved by all new cars is 95 grams of CO₂ per kilometre, whilst its data shows that average emissions level of a new car sold in 2017 was 118.5 g CO₂/km which is significantly below the current 2015 target set at 130 g CO₂/km. If the average CO₂ emissions for manufacturer's fleet exceed target values, it must pay "excess emission premium" which is currently set at €5 for the first g/km of exceedance, €15 for the second g/km, €25 for the third g/km and €95 for each subsequent g/km. From 2019, the cost will be €95 from the first gram of exceedance onwards.

Alongside the aforementioned targets and limits, the automotive industry has recently seen changes in engine certification process with introduction of Real Driving Emissions (RDE) conforming to recently revised EURO 6 legislation (EU6d) and Worldwide Harmonised Light Vehicle Test Procedure (WLTP) that replaced outdated New European Driving Cycle (NEDC). WLTP procedure was designed to better match real on-road performance and provide customers with more reliable data compared to NEDC cycle which was widely criticised for notoriously underestimating real-world fuel consumption.

To ensure that gasoline engines are well positioned in today's marketplace and can effectively meet both customer demands and legislative requirements, new technologies are making their way into their mass production. Lowering CO₂ emissions and strive for increasing thermal efficiency of Direct-Injection, Spark-Ignition (DISI) engines are being achieved through downsized turbocharging, utilising heavy dilution techniques by air and Exhaust Gas Recirculation (EGR), variable valvetrains, advanced ignition systems, high pressure stratified injection and increased compression ratio among others.

Stokes *et al.* (2000) identified four key categories that can be exploited in order to reduce losses and improve the overall engine efficiency:

- Exhaust energy reduction
- Heat loss reduction
- Friction reduction
- Reduction of gas exchange losses

Recent studies focused on realising high thermal efficiency, indicate that pushing into 40% region should be possible with the use of recent technology advancements. Takahashi *et al.* (2016) created a plot (Figure 1.1) which shows how actual engine efficiency is being affected by mechanical loss, pumping loss, cooling heat loss, exhaust loss and unburned loss at 2000 Revolutions per Minute (RPM).

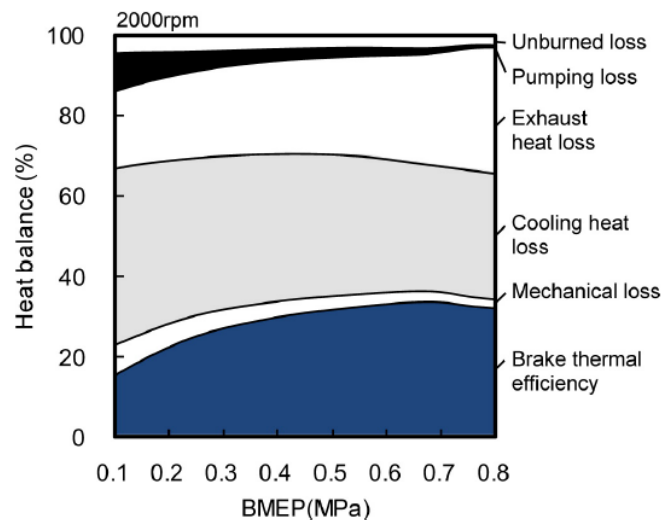


Figure 1.1 - The relation between engine load and heat balance. (Takahashi *et al.*, 2016)

It can be seen in the above that pumping loss is most severe at low load conditions, when engine is being heavily throttled, and that exhaust heat loss increases with engine load. Cooling heat loss is the most significant contributor of all losses and it accounts for up to 40% at low load and goes down to about 30% at high loads.

Analysis presented by Takahashi *et al.* (2016) is in-line with the early study, conducted at Ricardo, on increasing thermal efficiency of gasoline engine by introducing *The Lean Boost System*. Stokes *et al.* (2000) presented variation in typical losses for part and full load conditions.

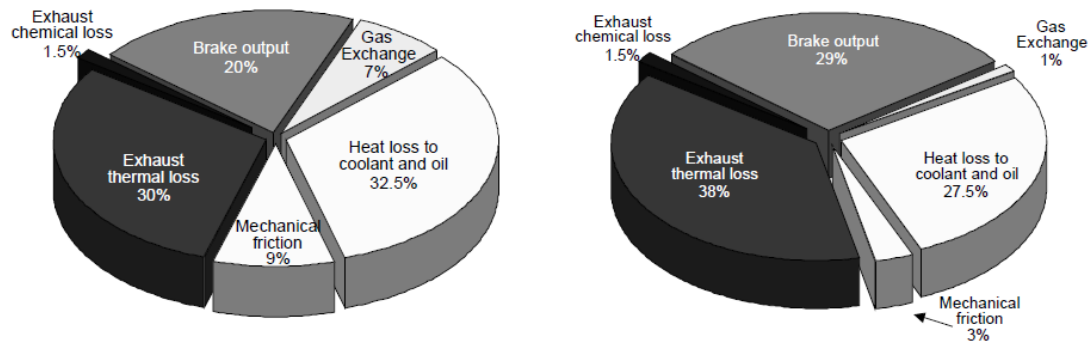


Figure 1.2 - Typical engine losses at part and full load. (Stokes et al., 2000)

It is believed that automotive industry trends which emerged in the last few years, seeing production engines using high charge dilution concepts, will dominate the marketplace along with new concepts that are in their research and validation phases. Bosch (Brandt et al., 2016) presented vision of the combustion concepts roadmap and highlighted homogenous lean combustion as a strongly promising and attractive concept due to its high potential of lowering NO_x emissions and therefore reduced cost for exhaust gas after-treatment.

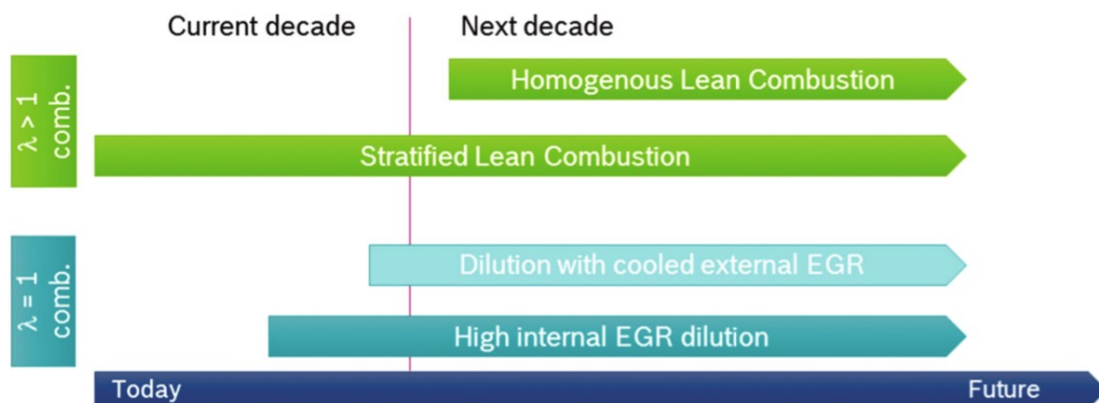


Figure 1.3 – Roadmap of combustion concepts presented by Bosch (Brandt et al., 2016)

1.2. Thesis outline

In Chapter 1, an introduction to spark ignition engines and their role in the current mix of powertrain choice for passenger vehicles is being discussed. Emphasis is made on the importance of emissions control and main trends towards lowering fleet average CO₂ emissions in the automotive industry are presented. Literature review discussing how charge dilution, being one of the main drivers to improving gasoline engine efficiency, plays an important role in the research efforts across the globe. Various dilution methods are discussed in detail, highlighting their benefits and shortfalls, together with the latest technologies serving as “enablers” to extending dilution tolerance in practical applications. Areas of limited or ambiguous conclusions reached up to date are identified and summarised. These findings are then used to set objectives of this study.

In Chapter 2, the experimental set-up built for the purpose of carrying out experiments is presented. Detailed information on hardware and software selected to commission world class Single Cylinder Research Engine (SCRE) facility is discussed in detail, together with measurement methods employed in the studies. Closed loop control systems developed at Imperial College and secondary voltage and current measurement systems are presented with discussion on challenges faced during design phase. Lastly, a set of preliminary studies carried out using the experimental set-up and matching of SCRE to its multi-cylinder counterpart are presented.

In Chapter 3, study of single valve deactivation strategy performed using the aforementioned experimental set-up is presented for three test points. Detailed analysis of combustion metrics and emissions formation is aided by in-cylinder flow field investigation. Benefits of shorter burn durations and penalty in heat transfer effects are thoroughly explained for Single Valve Actuation (SVA) strategy. Hypothesis on NO_x formation based on experimental findings is evaluated with the aid of numerical simulations.

In Chapter 4, valvetrain study is taken to another level by introducing staggered lift on the second valve in the form of either Late Intake Valve Opening (LIVO) or Early Intake Valve Closing (EIVC) strategy. Both variations are precisely evaluated in the pursuit of optimisation of fuel efficiency and lowering pollutants formation.

In Chapter 5, thorough investigation of increasing dilution tolerance by the use of advanced ignition systems is presented for low load 1500 RPM, 3.6 bar GMEP scenario. Stock coil

system is compared against two prototype, high energy designs provided by Bosch and BorgWarner. Dedicated measurement system designed for this study is used to evaluate secondary voltage and current traces, linking them with combustion parameters.

In Chapter 6, conclusions of all studies presented in Chapters 3-5 are summarised. They are critically evaluated and recommendations for future work are presented.

1.3. Charge Dilution Techniques

1.3.1. Lean Combustion

Engine efficiency can be increased when running lean with excessive amounts of oxygen, λ greater than 1, compared to stoichiometric conditions when amount of fuel injected in each cycle is exactly matched to the amount of oxygen that can burn all hydrocarbons.

Ayala *et al.* (2005) attributed higher net indicated efficiency when operating with air dilution to the following:

- Increased ratio of specific heats, γ ; similar to a diesel engine, the increased presence of air in lean mixtures increases the ratio of specific heats, which increases the expansion work by allowing the burned gases to expand through a larger temperature ratio prior to exhaust.
- Reduced pumping losses, achieved through running less throttle since higher intake pressures are required to allow more diluent to flow into the cylinder.
- Higher combustion efficiencies, because an increasing amount of the fuel is converted into useful energy instead of being lost in the exhaust.

Ayala *et al.* (2006) created a plot (Figure 1.4) that outlined individual contributions to increasing thermal efficiency when running lean at a certain engine operating condition – 1500 RPM, 3.5 bar Net Mean Effective Pressure (NMEP). It can be seen that engine efficiency peaks at a certain λ value and then after this point slower combustion can no longer be made up by other positive factors.

Efficiency curve under lean operation also proved to be more sensitive to load changes than stoichiometric conditions due to impact of reduced burn durations.

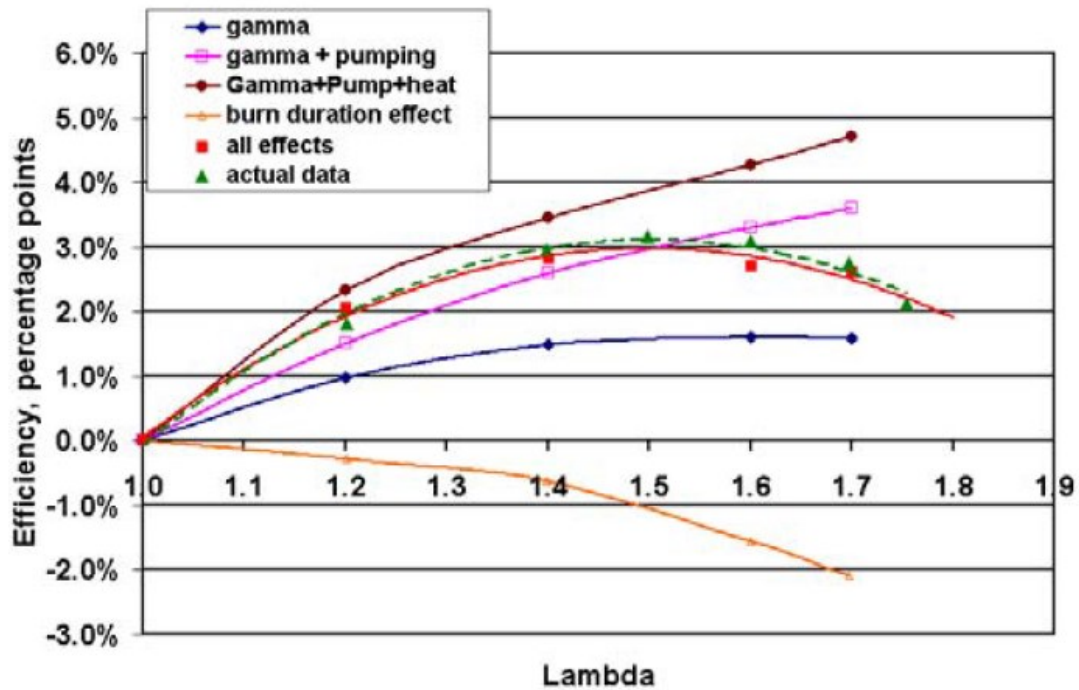


Figure 1.4 – Air-fuel ratio effect on engine efficiency; comparison of simulation results and actual data at 1500 RPM, 3.5 bar NEMP (Ayala et al., 2006)

Leaning the mixture can also be effective mean of lowering NO_x emissions. As outlined by Heywood (1988) production of nitrogen oxide emissions is a function of the combustion temperature, being greatest near stoichiometric conditions when temperatures are the highest, and equivalence ratio. Peak NO_x emissions occur at slightly lean conditions, where the combustion temperature is high and there is an excess of oxygen to react with the nitrogen. After passing the described peak, NO_x emissions drop due to lowering combustion temperatures, and therefore running lean mixtures has great potential for lowering NO_x engine emissions (shown in Figure 1.5).

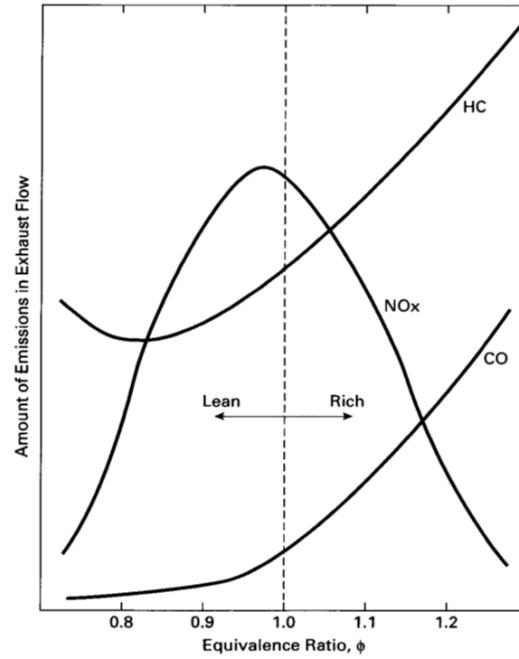


Figure 1.5 – Emissions from SI engine as a function of equivalence ratio. (Heywood, 1988)

Lean mixtures, however present challenges compared to running at stoichiometric condition. Many researchers (Ayala *et al.*, 2009, Grandin *et al.*, 1999, Grunden *et al.*, 1979) concluded that fundamental factor that determines the decrease in engine efficiency at high air-fuel ratios is a lengthening burn duration and unfavourable conditions for flame propagation.

As the combustion becomes longer, gradient of the rate of heat release (ROHR) curve decreases and cycle departs from the idealised constant-volume Otto process. Aforementioned researchers also attributed that phenomena to flame development which takes place early in the combustion process.

Air dilution is known to be limited by lean misfire limit. Heywood (1988) defined it as the leanest mixture stoichiometry at which the engine could be stabilized to operate at maximum brake torque (MBT) spark timing with a misfire frequency below a specified value. Once the engine starts misfiring, its efficiency drops due to discontinuity in fuel conversion.

As outlined by multiple investigations, lean burn limit can be extended by introducing higher levels of turbulence in the combustion chamber.

Scarcelli *et al.* (2013) concluded that increase in turbulent kinetic energy achieved through implementing design modifications to combustion chamber can be effective in enhancing flame propagation in the early stage, hence improving combustion duration and its stability.

1.3.2. Exhaust Gas Recirculation

Exhaust Gas Recirculation (EGR) technique was originally introduced to diesel engines in order to suppress NO_x formation by reducing in-cylinder temperatures. Method was later adopted in gasoline engine development not only to reduce NO_x formation but also to reduce throttling losses at its part-load operation.

Kinetics of NO_x formation are well described by Heywood (1988) and they take form of the extended Zeldovich mechanism. It is well known that highest NO_x formation is achieved for near-stoichiometric fuel-air mixtures and the process is heavily temperature dependent. NO forms in both the flame front and the postflame gases, however NO formation in the postflame gases almost always dominates any flame-front produced NO due to higher temperatures reached immediately after combustion (shown in Figure 1.6 below).

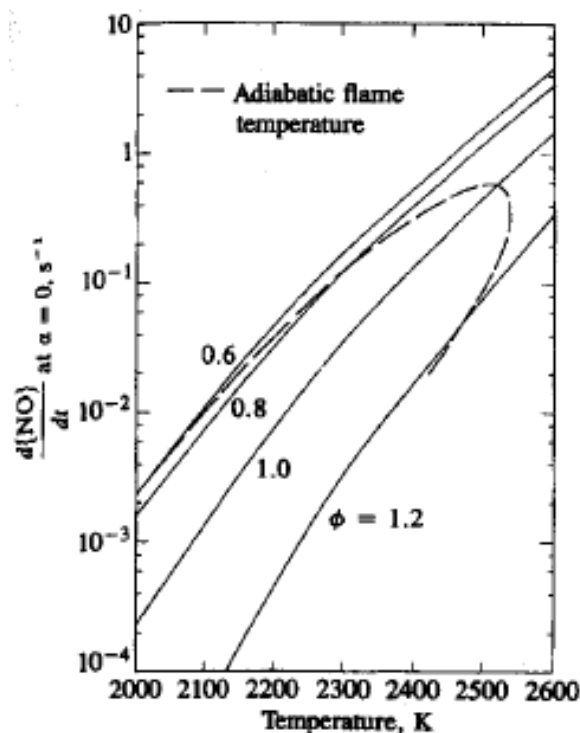


Figure 1.6 – NO formation rate as a function of gas temperature and fuel/air equivalence ratio in postflame gases. (Heywood, 1988)

EGR acts as a diluent in the intake mixture, and as a result the absolute temperature reached after combustion varies inversely with the burned gas mass fraction, hence it reduces NO emission levels.

CO₂ is much more effective in increasing the heat capacity of the cylinder charge, per unit mass of fuel, than N₂ and therefore EGR is preferred diluent when it comes to lowering NO emissions, as illustrated in Figure 1.7.

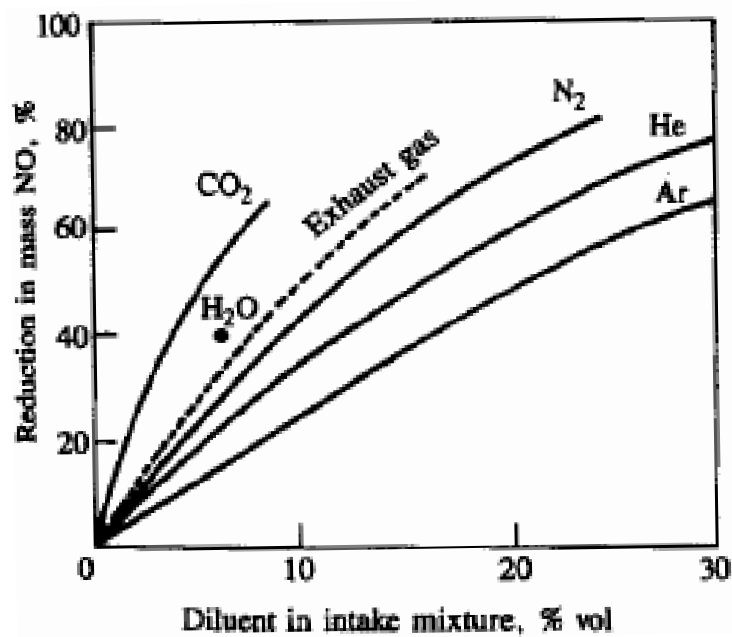


Figure 1.7 – Percentage reduction in mass NO emissions with various diluents (Heywood, 1988)

External EGR is usually fed to the engine cooled after going through a heat exchanger. Benefits of cooling the EGR were outlined by Wei *et al.* (2013) in their review of EGR applied to gasoline engines, and are as follows: lower NO_x values, better knock suppression and increased intake density which results in increased volumetric efficiency of the engine.

It should also be noted that mitigating knock at higher engine loads contributes to lowering exhaust heat loss. Looking at Takahashi *et al.* (2016) Figure 1.1 on page 29, it can be noted that exhaust heat loss is the one of the most significant losses contributor in the heat balance at high engine loads, especially when ignition timing needs to be delayed due to knock

occurrence. It also reduces pre-turbine temperatures and eliminates need for fuel enrichment as reported by Cairns *et al.* (2006).

EGR technique was extensively studied at various operating conditions and multiple researchers published data that outlines EGR benefits and shortfalls.

During engine part-load operating conditions it was reported by Wei *et al.* (2012) that DISI engines benefit from three main effects that result in improved fuel consumption:

- Reduced pumping work due to increased intake pressure which is required to maintain same torque output level (i.e. more diluted charge needs to enter the combustion chamber in order to maintain constant amount of oxygen).
- Reduced heat loss transferred to the cylinder wall due to lowered burned gas temperature.
- Reduced degree of dissociation in the high temperature burned gases that allows more fuel's chemical energy to be converted to sensible energy near TDC.

Cairns *et al.* (2006) proved using 2.0L turbo-charged (TC), DI engine that at high speed, high load (peak power region) cooled external EGR was a more effective suppressant of knock than excess air, with increased charge specific heat capacity, combustion phasing further advanced towards the optimum timing and improved combustion stability over a wider engine operating range. Compared to conventional fuel dilution (running rich mixture for charge cooling), EGR allowed savings of up to 17% in CO₂, 70% in CO and 80% in HC.

Duchaussoy *et al.* (2003) arrived at similar conclusions to Cairns *et al.* (2006) when testing effects of air and EGR dilution on 2L, 4-cylinder, TC engine running full load at 5000 RPM. The same study also identified EGR to be beneficial for turbine operation. When operating points were plotted on the compressor map, EGR dilution positioned itself in the top direction and on the pressure ratio axis with slight increase of the mass flow. That result indicates that smaller turbine could be used which in turn could give better engine responsiveness at low speed. On the contrary, lean burn conditions displaced the operating point in the right direction that would lead to choke conditions. Lean burn would therefore require bigger turbocharger which would lead to lower responsiveness at low speed and decreased vehicle driveability.

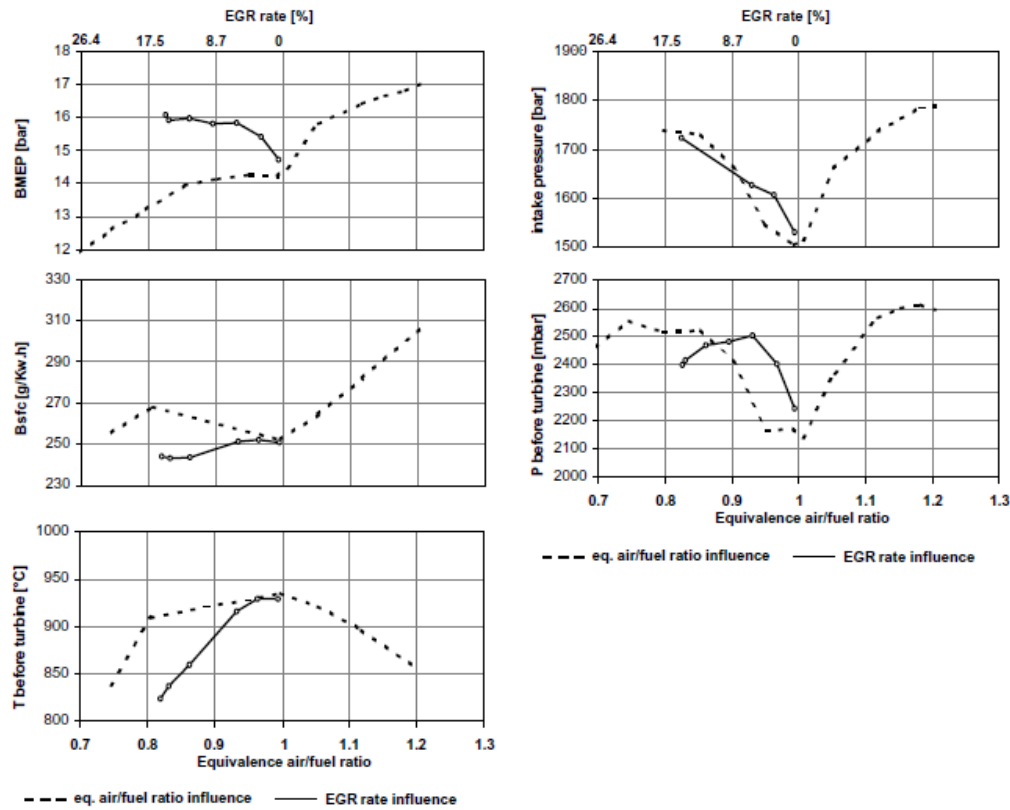


Figure 1.8 – Equivalence air/fuel ratio and cooled EGR sweeps comparison at full load; 5000 rpm (Duchaussoy *et al.*, 2003)

As feeding EGR to the engine replaces given portion of fresh charge with burned gases, the mixture becomes more difficult to ignite, cycle to cycle variation increases and speed of combustion is being compromised, resulting in practical limits to the maximum EGR dilution which can be tolerated in an engine.

Aforementioned drawback can be minimised by optimising combustion chamber design. Takahaski *et al.* (2016) found that EGR tolerance can be significantly extended by increasing turbulence intensity through the use of high tumble technology. Strong correlation was found between bore diameter divided by 10-90% combustion duration and turbulence intensity. As it affects main combustion event, such optimisation of the combustion chamber can have significant effect on increasing thermal efficiency of the engine by reducing negative effects of slow combustion.

An interesting idea of introducing stratified EGR was proposed and investigated by Xu *et al.* (2000) in a numerical study. It suggested separating EGR from the air-fuel mixture and pushing it towards the liner (illustrated in Figure 1.9). Benefits of such behaviour were analysed for

1500 RPM running with various EGR levels and swirl ratios. It was only a theoretical concept used in Computational Fluid Dynamics (CFD) studies, however it showed that fuel consumption could be reduced, together with NO_x , in-cylinder temperatures and heat losses through engine walls. According to this concept EGR does not mix with combustible air-fuel mixture, and therefore typical drawbacks of employing EGR discussed earlier could be eliminated. It was also concluded in this study that due to overmixing between EGR and air-fuel mixture, maximum EGR rate was set at 50%. Whilst the concept was encouraging due to theoretical advantages, it is duly noted that feasibility of practical introduction is very limited, especially in the automotive engine which needs to be operated across a wide range of speeds and loads.

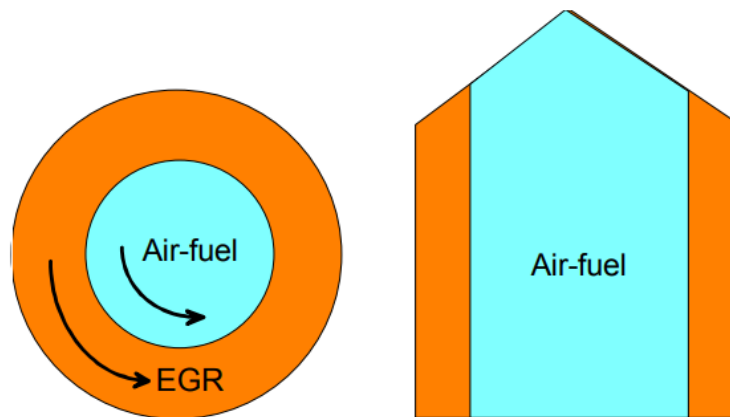


Figure 1.9 – Schematic of radially stratified EGR combustion process (Xu et al., 2000)

1.3.3. Residual Gas Fraction

The residual gas fraction (RGF) is often being referred to as hot or internal EGR. It is the burned gas remaining in the combustion chamber during compression that was not cleared during exhaust stroke of the previous cycle.

Engine running diluted with hot residuals can achieve thermal efficiency gains through heating the intake and thus promoting faster combustion duration and reducing cycle-to-cycle variations. According to Wei et al. (2013) hydrocarbon emissions are also lowered compared to cooled, external EGR.

Knowing that hot residuals speed up the combustion, engine can experience higher dilution level when subjected to RGF, compared to cooled EGR (given that operation point is clear of knocking region which is better suppressed by cooled EGR).

The amount of RGF trapped in cylinder is a function of engine breathing and is therefore controlled through inlet and exhaust processes. Heywood (1988) listed inlet and exhaust pressures, engine speed, compression ratio, valve timing and exhaust system dynamics as parameters that govern RGF trapping.

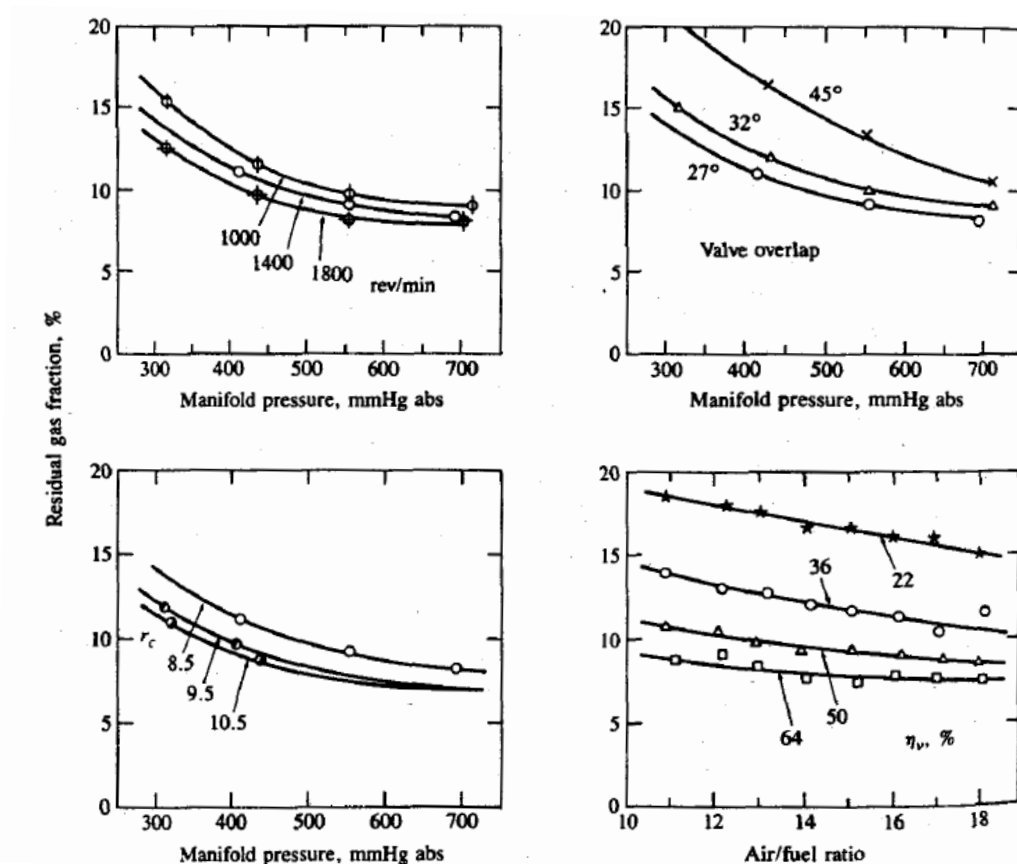


Figure 1.10 – RGF comparison subject to changes in speed, manifold pressure, compression ratio, valve overlap, air/fuel ratio (Heywood, 1988)

Compared to external EGR, which concentration is relatively easy to measure using two CO₂ analysers simultaneously – one measuring concentration in the intake and the other in the exhaust, accurate determination of RGF proves to be very challenging.

Heywood (1988) provided equation that can be used to calculate residual gas mass fraction x_r .

$$x_r = \frac{(\tilde{x}_{CO_2})_C}{(\tilde{x}_{CO_2})_e}, \text{ where} \quad (1.1)$$

$\tilde{x}_{CO_2}$ mole fraction in the wet gas

C compression

e expansion

Equation (1.1) above requires measurement of gas extracted from the cylinder during compression stroke (as the newly formed CO₂ after the start of combustion distorts the results) which makes it quite complicated and challenging from engine instrumentation perspective.

Schwarz *et al.* (2003) used flash-mounted fast gas sampling valve (GSV) on the combustion chamber wall and a GSV controller to open the valve during defined moment of the engine cycle. To prolong the timeframe during which sampling could be performed without distorting results, ignition was skipped during the measurement cycle, so that sampling could take place close to TDC. This presented the benefit of sampling more homogenous charge at an elevated pressure, maximising the sampling process.

Sample was taken every eighth cycle and then passed through CO₂ analyser. Process run for about 400 cycles (50 of which were sampled) after which CO₂ signal reached stationary values (Figure 1.11).

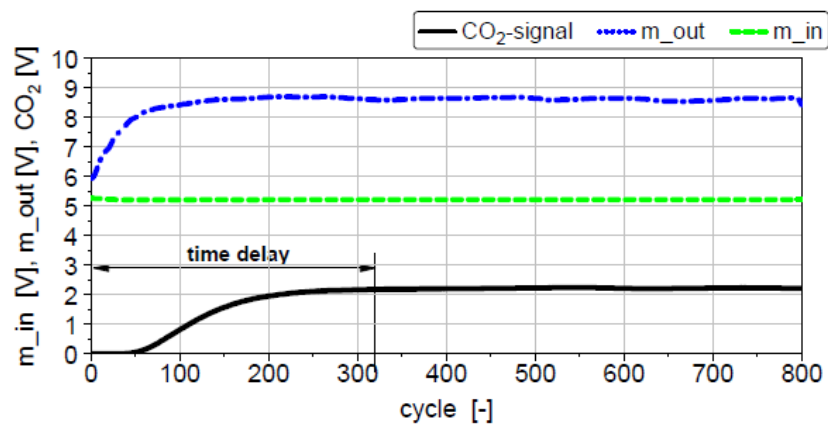


Figure 1.11 – Recorded signals of CO₂, inlet and outlet flow of the gas sampling valve (Schwarz *et al.*, 2003)

Measured values were then input to the following equation:

$$X_{RG} = \frac{M_{BG} \left[\Psi_{CO_2,C} \left(\frac{M_A}{M_F \cdot \lambda_{L_{St}}} + 1 \right) - \Psi_{CO_2,A} \right]}{M_A \left[\left(1 + \frac{1}{\lambda_{L_{St}}} \right) (\Psi_{CO_2,BG} - \Psi_{CO_2,C}) + \frac{M_{BG}}{M_A} (\Psi_{CO_2,C} - \Psi_{CO_2,A}) + \frac{1}{\lambda_{L_{St}}} \frac{M_{BG}}{M_F} \Psi_{CO_2,C} \right]} \quad (1.2)$$

M_{BG}, M_A, M_F	molar masses of exhaust gas, air, fuel [kg/kmol]
$\Psi_{CO_2,BG}, \Psi_{CO_2,A}, \Psi_{CO_2,C}$	CO ₂ concentration in exhaust gas, air, cylinder charge [Vol-%]
L_{St}	absolute air-fuel ratio [kg _{air} /kg _{fuel}]

In the same study at University of Karlsruhe, Schwarz *et al.* (2003) used simulation based on the gas exchange analysis to estimate RGF. Models used for such study were very detailed and modelling of valve lifts and coefficients was required for numerical analysis (solved with Runge-Kutta method).

Correlation exercise showed that gas sampling method and numerical analysis were in good agreement. Biggest differences were found at low residual levels. Researchers used various valve lifts (shown in Figure 1.12) to prove that presented methods could be implemented with modern engines that run variable valvetrain systems.

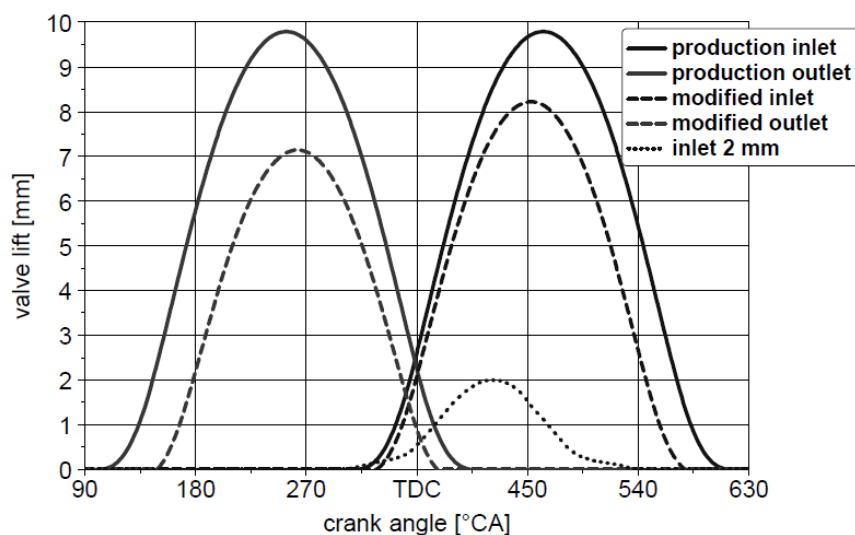


Figure 1.12 – Valve lifts used by Schwarz *et al.* (2003) in their study.

Alger *et al.* (2004) used a single cylinder engine with optical access and employed spontaneous Raman scattering diagnostic to measure in-cylinder burned gas residual to validate proposed

model for calculating RGF. Approach proved successful, however it required an optical engine and performing such validations using thermal engine is not possible.

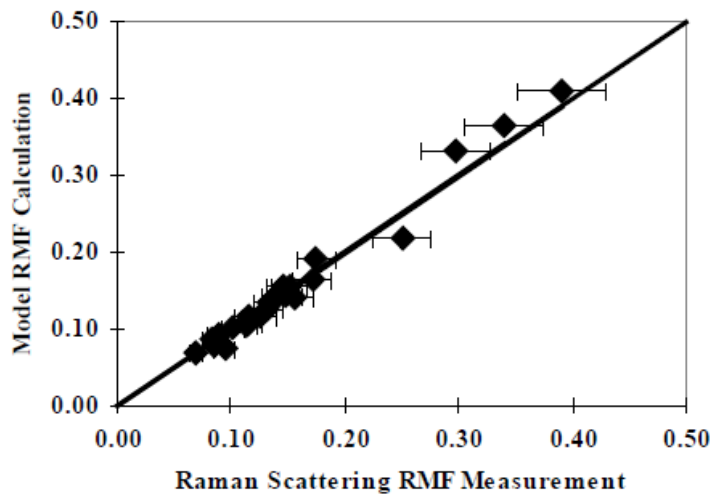


Figure 1.13 – Comparison of analytical RGF to RGF measured using Raman scattering (Alger *et al.*, 2004)

All the aforementioned studies looked into correlating experimental data against numerical models. It was done due to the fact that experimental measurements require significant set-up effort and cannot be used during daily running procedures in engine development testing (on the contrary to standard exhaust species measurements using Horiba or AVL exhaust emission benches).

1.3.4. Overall dilution

Dilution methods have been investigated by numerous researchers in the past, however most of the literature is focused on the individual assessment of either running with excess oxygen (lean), using External Gas Recirculation (EGR) or attempting to quantify Residual Gas Fraction (RGF) left in cylinder from previous work cycle.

Some back-to-back comparison tests were performed in order to characterize performance differences when using different dilution methods. It was confirmed by Ayala *et al.* (2006) that Net Indicated Efficiency follows very similar trend line when given engine is running with either EGR or excess air and that in both cases it achieved its peak efficiency with the same 10-90% MFB duration, which tends to be irrespective of any shifting of the 10-90% burn duration curve (shown in Figure 1.14).

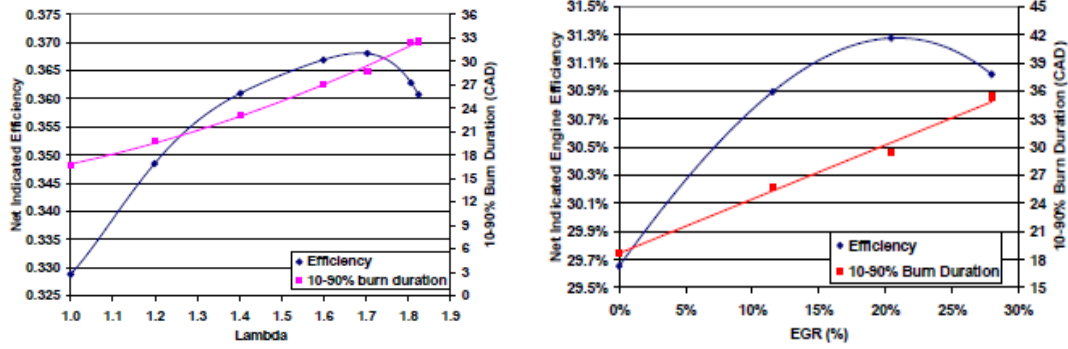


Figure 1.14 – Effect of EGR and Lambda on efficiency and burn duration at 1500 rpm; 3.5 bar NMEP for EGR and 6.0 bar NMEP for air-dilution (Ayala *et al.*, 2006)

There is however very little information available on how overall charge dilution can be measured and attributed to any of the aforementioned components in a modern DISI engine. Shayler *et al.* (2000) introduced parameter called gas/fuel ratio (GFR) and conducted series of experiments using two Port Fuel Injection (PFI) Ford engines with homogenous charge designs in his studies – 1.25/Zetec SE and 1.8/Zetec.

$$GFR = \frac{(m_a + m_f + m_b)}{m_f} = \frac{(AFR + 1)}{(1 - x_b)}, \text{ where} \quad (1.3)$$

m_a mass of air

m_f mass of fuel

m_b burned mass

x_b burned gas fraction, $x_b = \frac{m_b}{(m_a + m_f + m_b)}$

It was found that stability limits and misfire likelihood can be defined as a limit on the GFR with limit of 24:1-26:1 for the fully warm operation and slightly lower 23:1 for cold operating conditions. In the publication authors claim that DISI engines should have much higher tolerance that may approach 60:1, however there are no experimental results to back that statement.

Tang *et al.* (2013) attempted to conduct direct comparison of lean vs. EGR operation using 1.6L NA engine. In-Cylinder Dilution Factor (α) was introduced. It is defined as the total gas

mass trapped in cylinder at the end of the gas exchange process, divided by the air mass required to burn the fuel mass under stoichiometric condition:

$$\alpha = \frac{m_{Gas, total}}{AFR_0 m_f} \quad \alpha = \frac{\lambda}{1 - RGF} \quad (1.4)$$

Due to the fact that RGF is always >0 , the in-cylinder dilution factor α is always $>\lambda$.

When running experiments Tang *et al.* (2013) attributed α values for two instances, one running with excess air and the other running with EGR (Figure 1.15). Once dilution factors were calculated, direct comparison between two cases was performed in terms of emissions and combustion stability. It was found that under same dilution conditions EGR consistently performed slower burn (from $\alpha=1.2$ onwards) and combustion fell over at $\alpha=1.45$ when CoV of IMEP went up to 25% (compared to 4% at $\alpha=1.40$). Lambda sweep reached α of almost 1.7 and did not exhibit significant increase in CoV of IMEP (well under 5%). Sweep did not go any further than 1.7 due to lowering combustion efficiency (attributed to slow burn) and increasing ISFC.

Researchers in this study also identified in-cylinder heat loss as main contributor to ISFC drop. It was concluded that almost 50% increase of in-cylinder gas mass from a dilution factor of $\alpha=1.13$ to $\alpha=1.65$ for lambda sweep (Figure 1.16), gas temperature rise by combustion (for the same amount of fuel) should be expected 50% lower, hence reducing the in-cylinder temperature. Such reduction was responsible for significantly reduced heat transfer to the engine walls and as a result lower ISFC.

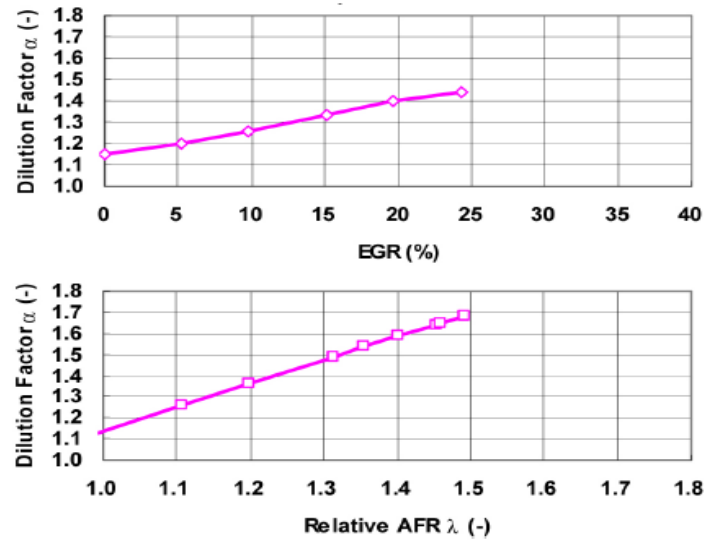


Figure 1.15 – In-cylinder dilution factor (α) as a function of EGR and relative AFR (λ), 2000 rpm, 2.0 bar BMEP (Tang et al., 2013)

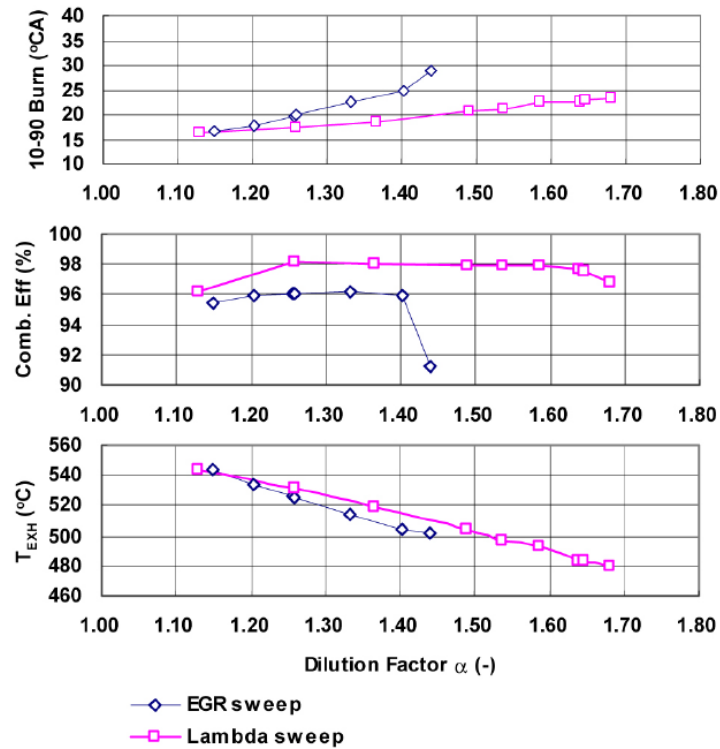


Figure 1.16 – Effects on combustion duration and efficiency of EGR vs Lean burn at 2000 rpm; 2.0 bar BMEP (Tang et al., 2013)

In other study, Ivanic et al. (2005) introduced Thermal Dilution Parameter (TDP) to compare engine operating parameters on a common basis when plasmatron gas was added to the charge

mixture. TDP accounted for heat capacities of all diluents present in the charge mixture, and therefore certain value of TDP can be given to any EGR or excess air dilution level.

$$TDP = \frac{\Delta T_{stoichiometric}}{\Delta T} \quad (1.5)$$

ΔT – chemical energy released during combustion per constant volume heat capacity of the unburned cylinder charge:

$$\Delta T = \frac{\sum_{i=1}^5 m_{Xi} \cdot Q_{LHV, Xi}}{\sum_{j=1}^9 m_{Xj} \cdot C_{v, Xj}} \quad (1.6)$$

with

$$X_i \in (C_7H_{14}, H_2, CO, CH_4, C_2H_2)$$

$$X_j \in (H_2O, H_2, CO, CO_2, N_2, C_7H_{14}, C_2H_2, O_2, CH_4)$$

It was outlined that since change in specific heat of gases was ignored, ΔT only approximated the adiabatic constant volume temperature difference between the burned and unburned gas. In their analysis Ivanic *at al.* (2005) evaluated specific heats of all species at a temperature of 600K which approximated the temperature at the time of spark. $\Delta T_{stoichiometric}$ was then determined experimentally to be around 3000K for a baseline configuration at $\lambda=1$ and no hydrogen dilution.

In the same study, air dilution proved to be more effective mean of increasing engine efficiency at low speed, low load condition by about 3 percentage points, whilst EGR consistently outperformed air dilution for lowering engine NO_x emissions (Figure 1.17). Air dilution proved to be effective in lowering NO_x emissions only after it passed a certain level of TDP – about 1.4 (shown in Figure 1.19) which ties up with theory of NO_x formation presented by Heywood (1988).

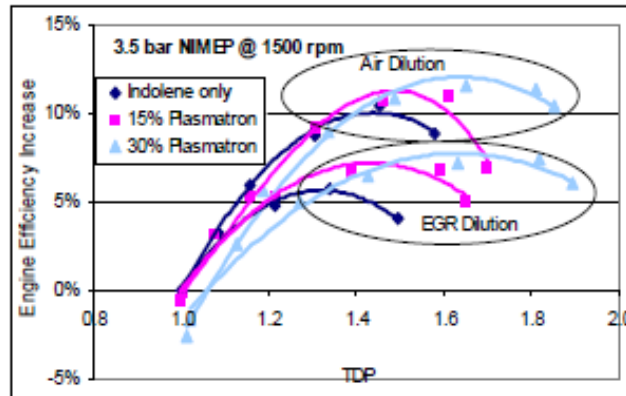


Figure 1.17 – Comparison of net indicated efficiency increases between air and EGR-diluted mixtures (Ivanic *et al.*, 2005)

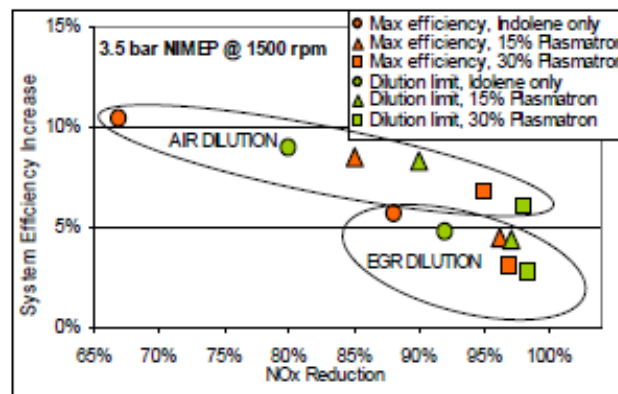


Figure 1.18 – Normalised NO_x emissions of diluted mixtures versus TDP (Ivanic *et al.*, 2005)

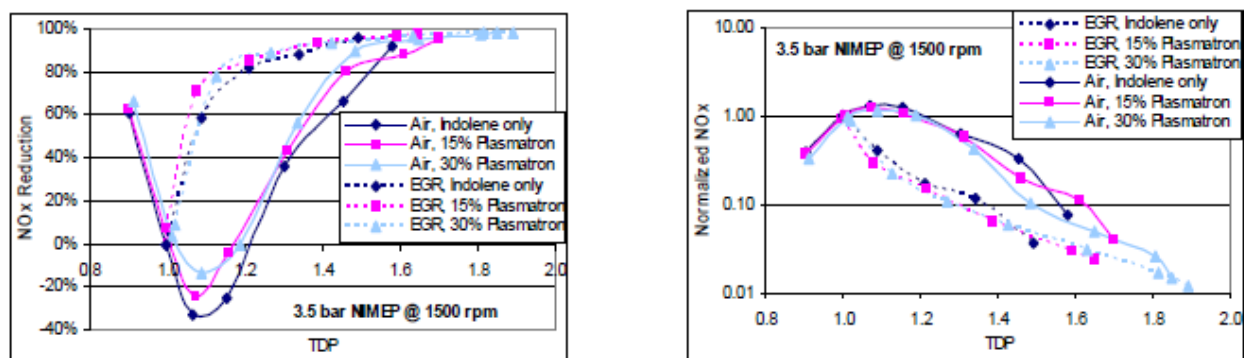


Figure 1.19 – NO_x reduction comparison of dilution strategies (Ivanic *et al.*, 2005)

Both Shayler *et al.* (2000) and Tang *et al.* (2013) used engines with fixed Variable Valve Timing (VVT) in order to simplify their analysis. Such an approach greatly simplified calculation of RGF. Shayler *et al.* (2000) used a simplified calculation to obtain residual gas fraction which was dependent on engine design and operating condition. It provided an

estimate of the RGF for the few cases run on the engine with relatively few variable parameters, however it was impractical method for estimating residuals content on the engine that achieves load control through the use of CVVL, hence using valve throttling. Correlation between the modelled and measured RGF was also not presented.

1.4. Ignition Systems

1.4.1. Types of Ignition Systems

Ignition systems are often considered to be one of the key components that can be leveraged to enhance dilution tolerance of an engine. Novel continuous inductive discharge ignition systems were developed and tested on the SI engines in the recent years. Single Coil on Plug (CoP) design was replaced by more sophisticated dual-coil design. Such approach enables discharging coils in a number of different ways, creating continuous discharge or multi-strike operation in addition to possibility of achieving higher energy single strike events.

In addition to dual-coils, more radical ignition system designs such as jet ignition, laser ignition or microwave-based (plasma ignition system) have also been extensively studied in the recent past. Mahle (2019) claimed that the jet ignition system they developed enables >42% brake thermal efficiency and fuel consumption lower than 200 g/kWh for ultra lean operation ($\lambda \approx 2.0$). Whilst such claims are very attractive, the system requires use of two injectors and pre-chamber which significantly alter design of cylinder head (Figure 1.20). Additional direct injector within the jet igniter assembly delivers a small amount of fuel (<5% of the total system fuel) to the pre-chamber which is then ignited by the spark plug. The ignition chamber is connected to the main chamber by a number of orifices which allow jets of partially combusted products to ignite the main charge. Small orifice sizes cause turbulence in the hot gas jets which penetrate deep into the main combustion chamber and cause an evenly distributed ignition effect.

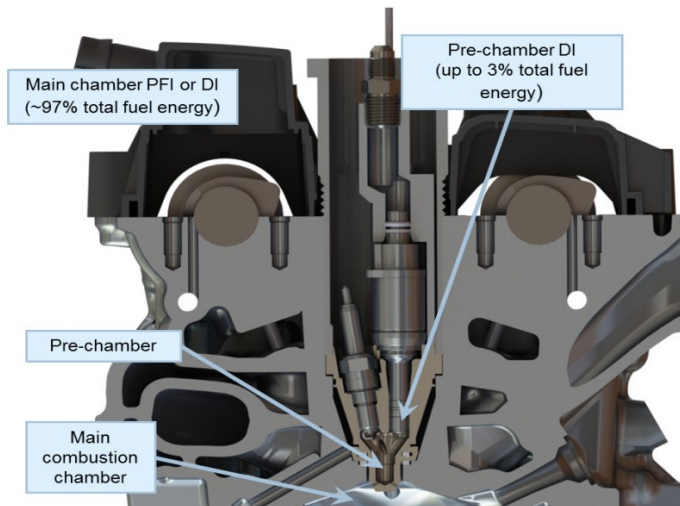


Figure 1.20 – Cross section of MAHLE Jet Ignition System (MAHLE, 2019)

Another alternative system is the microwave assisted plasma ignition system. It is the structure composed of direct ignition and microwave from a spark plug tip by mixing microwave of 1kw class 2.45 GHz in the pathway of a high voltage supplied to the spark plug from the second coil of the ignition coil, as shown in Figure 1.21.

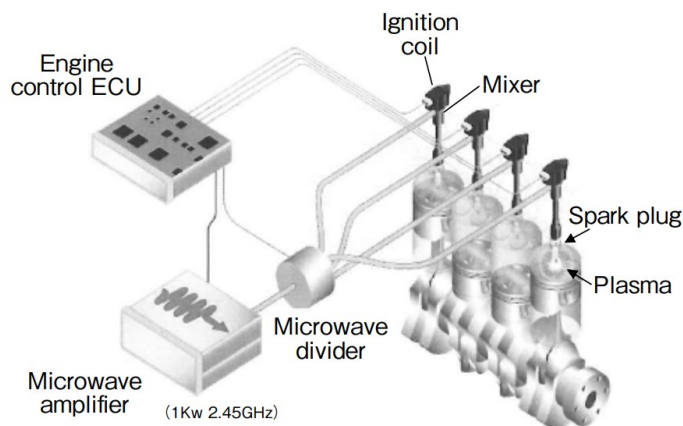


Figure 1.21 – Configuration of Plasma Ignition System (Oharara, 2017)

In 2017 Denso summarised their three and a half years of research on Plasma Ignition System and Okahara (2017) reported that add-on microwave assisted system did not establish compatibility between ignition performance and durability performance. It was caused by adding the microwave irradiation function to the spark ignition system and too big current was passed through electrode with overheated status to support plasma volume.

Due to technical limitations of experimental studies, such designs were not considered as viable alternatives to CoP solution for the purpose of this Thesis.

Igniting lean and diluted mixtures proves to be more difficult as they have lower burning velocity compared to stoichiometric conditions. Peterson *et al.* (2010) outlined that if high spark discharge energy is deposited in the spark plug gap, large early flame kernels can be formed even at very lean conditions due to significant heat supply from the spark plasma. Such kernels need to be developed quickly into a turbulent flame front, and then propagate throughout the fuel-containing regions within the cylinder. Alger *et al.* (2011) identified amount of energy delivered by the ignition circuit and the discharge duration as two main factors that have most significant impact on igniting diluted mixtures. Dual coil systems were introduced to provide combustion system with high energy source of ignition that would not suffer from the durability and electronic noise issues of high-power systems such as plasma jet or railplug. Alger *et al.* (2011) reported that testing performed on 4-cylinder 2.4L PFI engine with 8.6:1 CR using Dual Coil Offset system developed at Southwest Research Institute (SwRI) could significantly extend EGR tolerance of their engine, especially at part load when running in continuous discharge mode.

Results presented in Figure 1.22 show that the same engine stability limit of 4% CoV of IMEP could be reached at 10% EGR with standard CoP or 17.5% EGR when applying continuous discharge mode. EGR tolerance increase of 75% was quite promising for engine calibration that could achieve significant BSFC reduction whilst maintaining good engine stability levels required during throttle tip-out condition (reducing the likelihood of misfiring).

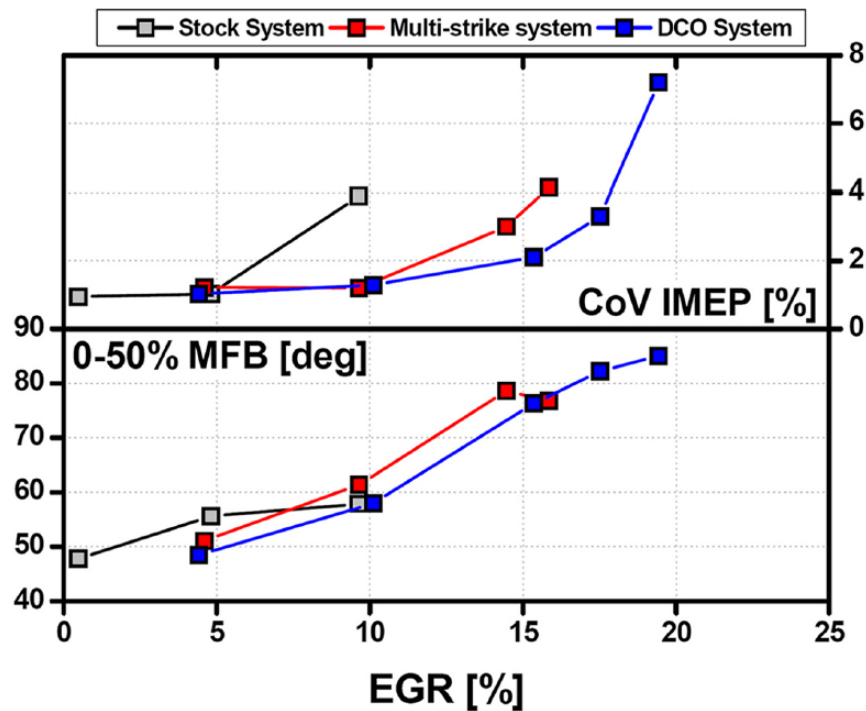


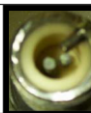
Figure 1.22 – Stability and burn rate results for 2000 rpm, 2 bar BMEP condition (Alger *et al.*, 2011)

Further studies conducted by Alger *et al.* (2013) on the performance of DCO system proved that a higher energy operating mode, achieved by replacing low power coils with High Current Density (HCD) ones, further improved burn rates. This effect was best observed at high engine speeds, where the higher current helped to resist blow out and improved initial kernel formation, enhancing initial burn duration. Higher power systems (such as DCO developed at SwRI) use more energy, however it was proven through multi-cylinder engine testing that fuel consumption benefit was much greater than increase in ignition energy across all tested conditions.

Similar studies were performed by other researchers. Jung *et al.* (2017) presented their findings on running a single-cylinder engine equipped with a ten coil ignition system and intake port adapter designed to enhance in-cylinder turbulence levels. It was reported that air-fuel ratio was successfully increased by introducing the ten coil, high energy ignition system, and then improved even further by the use of intake port adapter. Study concluded that introducing advanced ignition system with high spark discharge energy was a practical and effective mean for increasing dilution tolerance of an engine, however it did not provide any explanation as to why spark discharge energy variations and combustion variations were not related.

Briggs *et al.* (2014) conducted series of experiments using constant-volume combustion chamber and single cylinder research engine in order to assess various ignition systems (presented in Table 1.1) when operating with EGR diluted mixtures.

Table 1.1 – Ignition systems tested by Briggs *et al.* (2014)

Igniter	Delivered Energy	Photo	Igniter	Delivered Energy	Photo
Conventional 3 mm nickel-plated copper electrode	30 mJ		Iridium #4 with very high energy coil	300 mJ (inductive)	
Iridium #1 0.6 mm iridium tipped electrode	30 mJ		Plasma jet plug	2.4 J	
Iridium #2 0.5 mm iridium tipped electrode	30 mJ		Rail plug	2.4 J	
Multi-torch	30 mJ		Platinum spark plug with extended duration circuit 0.8 mm platinum electrode	1.2 J	
Surface #1	30 mJ		Dual spark open chamber plug	2.4 J	
Surface #2	30 mJ		Dual spark torch plug	2.4 J	
Iridium #2 with high energy coil	100 mJ		Cold plasma plug	4 J	
Iridium #3 0.6 mm iridium center electrode with 0.8 mm platinum ground electrode	100 mJ		Laser ignition	80 mJ	
Iridium #4 0.6 mm iridium center electrode with 0.6 mm platinum ground electrode	100 mJ				
Surface #2 with CD ignition system	55 mJ (capacitive)				

It was proven across various operating conditions that the most effective systems for stable operation at high EGR rates were systems which created a large effective kernel and/or a long kernel lifetime. High energy spark discharge systems performed better and could ignite a flame under EGR dilution as high as 40%.

1.4.2. Ignition Event Characteristics

First truly comprehensive survey of the principles and basic properties of spark ignition was published by Maly (1984), covering its physics and applications in IC engines. It is recognized that spark initiating event – electrical breakdown across the spark plug gap caused by voltage rise between its electrodes is very short (nanoseconds), and therefore extremely challenging to be properly measured. The breakdown phase is then followed by arc and glow phases that are

few orders of magnitude longer. Heywood (1988) presented these typical three phases and they are shown in Figure 1.23 plotted vs time.

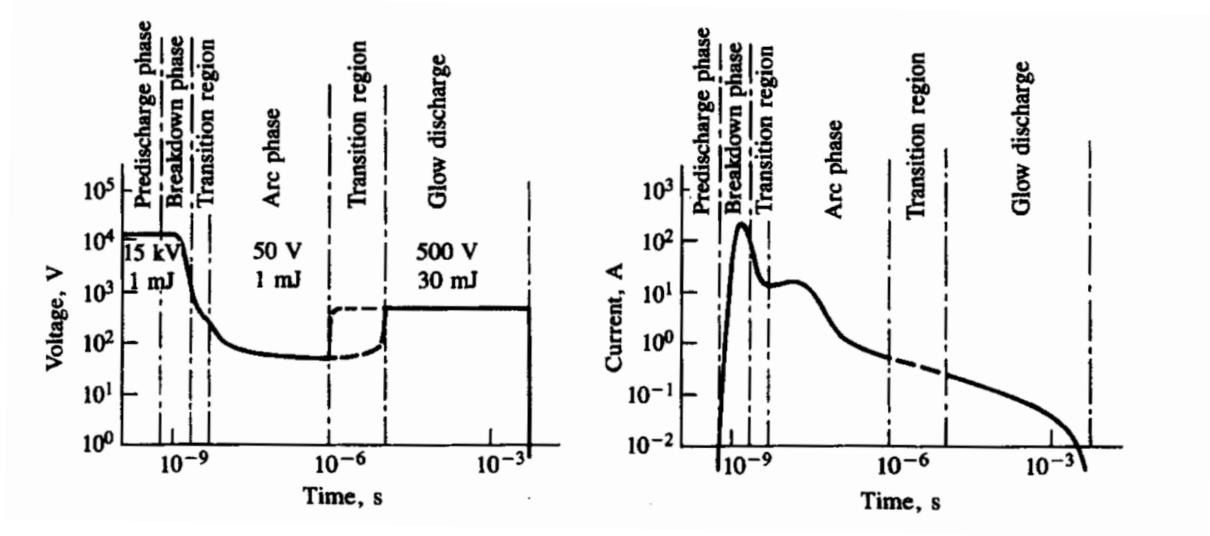


Figure 1.23 – Typical voltage and current traces variation with time, specifying three main phases (Heywood, 1988)

Aforementioned breakdown phase is preceded by the prebreakdown/ predischage phase during which gas volume within the electrode gap represents a perfect insulator. Once the spark pulse is applied and electric field reaches sufficiently high values (50-100 kV/cm), field-accelerated electrons ionize the gas molecules by collision, rapidly generating additional electrons and ions (avalanche effect; Raetcher, 1964). Electrons are collected on the anode, however additional new starting electrons are needed near or at the cathode. As long as the ionizing process produces less electrons than required for rendering the discharge, process is contained within the predischage phase which length is inversely proportional to the voltage rise rate.

Having enough feedback electrons produced, an overexponential increase in discharge current (at approximately $I > 10$ mA according to Maly, 1984) originates, marking a start of breakdown phase. Current rises within nanoseconds to hundreds or thousands of amperes, whilst voltage can exceed 10 kV. Cylindrical plasma channel forms between electrodes (initially at about 40 μm in diameter) and temperature rises up to 60,000 K due to electronic excitation.

Extremely short breakdown phase is then followed by arc phase which is few orders of magnitude longer, usually lasting hundreds of microseconds (Maly and Vogel, 1978). It is characterised by currents in excess of 100 mA where the upper limit is set by the external electrical impedances. Voltage drops to less than 100 V, and plasma temperature rapidly

reduces to approximately 6,000 K, whilst plasma kernel grows in size. In the arc phase spark event starts showing sensitivity to in-cylinder flow field. Velocity can stretch the discharge channel and increase its length. De Soete (1983) showed that higher velocities result in shorter spark duration.

Glow phase is the last phase of spark event, following the arc. Glow discharge is very similar to the arc phase with the most significant difference being the cold cathode. It gets its feedback electrons from the very inefficient electron liberation by ion impact, resulting in low current densities and currents (currents higher than 100 mA cannot be sustained in this phase). This phase was identified by Heywood (1984) to be fully dependent on the type of ignition system, typically lasting for several milliseconds for inductive systems. As the current eventually goes to zero in this phase, voltage can sustain levels of approximately 500 V. Regardless of the fact that this phase is low power (compared to breakdown and arc phases), it remains responsible for more than 90% of the energy release (Kalghatgi, 1987) thanks to its duration being orders of magnitude longer than the phases preceding it.

High local flow velocities were reported to lead to long plasma arcs, which help to distribute the thermal spark power in the mixture (Brandt *et al.*, 2016). It also leads to high energy transfer efficiencies.

Systems using elongated sparks were reported to experience so called “restrikes” during current decrease. Restrikes were found to occur when extended length of the spark lead to formation of a short circuit arc between plasma channels (shown in Figure 1.24). When the aforementioned decrease can be avoided by keeping the current level at constant, high level, positive impact on spark deflection can be achieved. However, the deflection process has its limit and cannot be sustained endlessly.

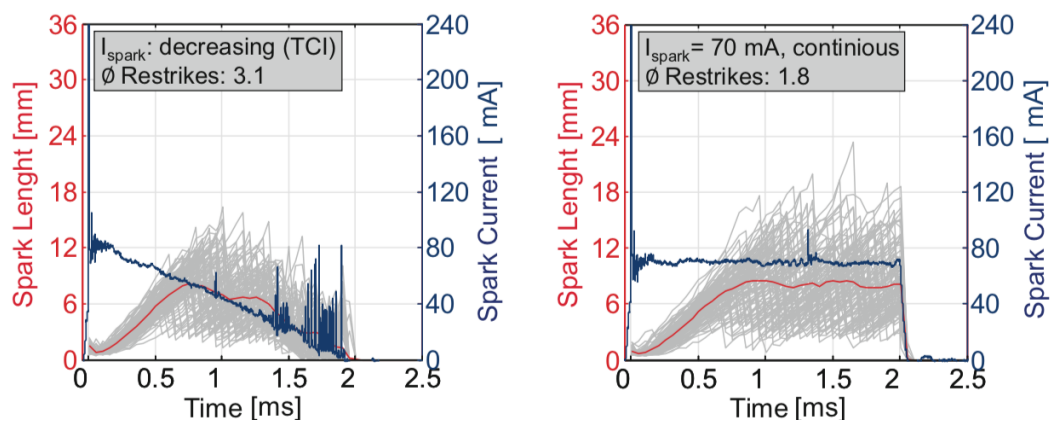


Figure 1.24 – Evolution of spark length for decreasing and continuous spark current for 50 cycles at $v=10$ m/s. Red line showing mean spark length (Brandt *et al.*, 2016)

High flow velocities were also reported to cause “spark blowout”. In this event, the spark plasma is detached from the electrodes, therefore requiring new spark breakdown between the electrodes. If the blowout occurs, ignition system must establish high voltage build up and provide a new spark breakdown within a short time.

Effects of low flow velocities in the vicinity of spark plug were reported by Schneider et al. (2016). Their optical measurements on a single cylinder research engine showed that misfires and delayed combustion cycles can be expected, leading to detrimental combustion stability.

1.5. Modern valvetrain design & Constantly Variable Valve Lift (CVVL) systems

Valvetrain systems have been studied in context of increasing gasoline engine efficiency for decades. Variable valvetrain technology has been used extensively to reduce pumping losses and many researchers reported their findings on stoichiometric operating conditions due to benefit of retaining well-established three-way-catalyst for exhaust after treatment. One of the first advanced systems introduced to the market was BMW’s VANOS which enabled shift of the valve lift to continuously control the valve closing. Flierl and Kluntig (2000) reported that employing this fully mechanical valvetrain system offered 10% fuel consumption reduction and an additional peak torque of 5% in the aforementioned $\lambda=1.0$ operation.

Constantly Variable Valve Lift (CVVL) systems are gaining popularity within the automotive industry – especially in the higher end, premium sector where additional cost of equipping engine with advanced technology packages can be more easily justified, however most of the development and research work is confidential and there are very few publications which share research findings and suggest which modes of running CVVL – Early Intake Valve Closure (EIVC) - Miller, Late Intake Valve Closure (LIVC) – Atkinson or hybrid mode (combining both EIVC and LIVC) are beneficial under part load conditions. Miller and Atkinson cycles have their fundamental principles based on the difference between geometrical and effective compression ratio. Engines built with high geometrical compression ratio (contributing towards high thermal efficiency of the working cycle), use valve action to effectively lower it during operation i.e. needed for knock mitigation (Ferrey *et al.*, 2014) or valve throttling. It also allows for adjusting compression/expansion ratio which was originally fixed with traditional valvetrains.

The 2.0L TFSI EA888 Gen.3B engine from Audi (Wurms *et al.*, 2015) with increased geometric CR serves as a good example of how Miller cycle was implemented to gain efficiency improvements (shown in Figure 1.25). When running under part load conditions shorter intake opening event (140°CA vs conventional 200°CA) combined with increased geometric compression ratio offered so called *extended expansion* mechanism. Thanks to significantly advanced IVC engine could be dethrottled which lowered pumping losses. Subsequently combustion occurred within a smaller volume and higher pressures were reached.

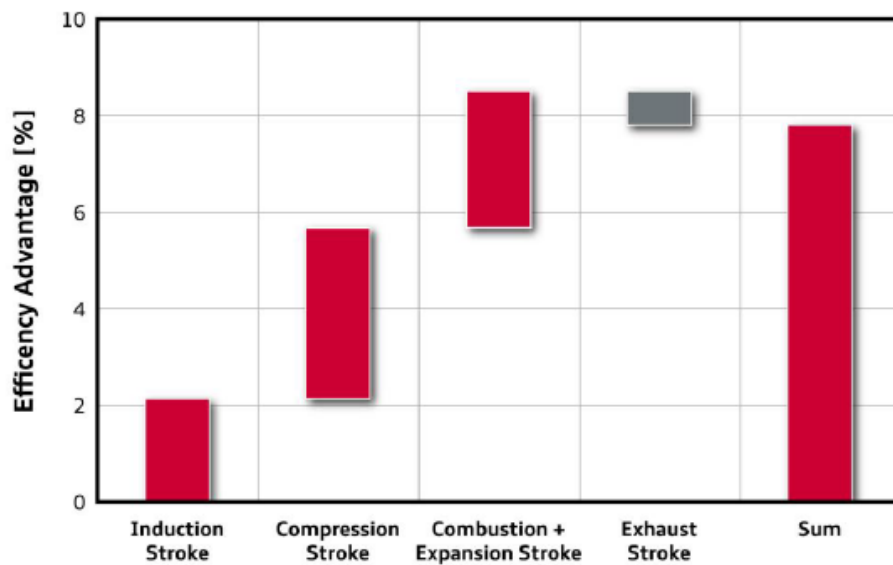


Figure 1.25 – Efficiency gains of 2.0L EA888 Gen 3.B (CR=11.7) vs EA888 Gen 3 (CR=9.6) (Wurms *et al.*, 2015)

Use of variable lift system enables switching to longer intake event, allowing larger cylinder charger under high load operation. Wurms *et al.* (2015) claimed that EIVC presented benefits over LIVC (especially under part load operation), however due to confidential character of the studies and engine development work, no results backing that statement were released in the public domain. There are no conclusive studies that would determine how mixture formation is being affected by either using EIVC or LIVC and it is believed that benefits related to application of either Atkinson or Miller cycle are engine-specific.

Bernard *et al.* (2009) claimed that in conventional gasoline engines, throttle-based air control wastes about 10% of the input energy in pumping the air from lower intake pressure to the atmospheric exhaust pressure, hence there is great potential to be explored using this

technology. These claims appear to be in-line with the Flierl and Kluntig (2000) findings mentioned earlier, as well as Cleary and Silvas (2007) who reached 7% net-specific fuel consumption improvement by employing EIVC variable valve actuation strategy on GM's 3.4L DOHC engine under part-load operation (1300 RPM; 3.3 bar NMEP). Same study also reported 25% lower engine-out NO_x emission levels, however HC emission was 25% higher.

In addition to fuel consumption benefit, Patel et al. (2008) explored effects of running in EIVC mode on engine heat release. Tests run on both thermodynamic and optical engines led to the conclusion that higher fuel consumption benefits were expected at low load (5%) compared to medium load (2.5%). All investigated cases showed that faster heat release rates were observed for single intake valve operation compared to traditional two intake valve operation. It was also reported that position of DI fuel injector relative to the airflow could be an important parameter in influencing the mixing process, based on the analysis of Mie images obtained by the use of optical engine. It is also worth noting that Lotus engine used for this study had a centrally mounted six equi-spaced streams with its axis inclined towards the spark plug and was operated at 120 bar injection pressure.

Scheidt *et al.* (2014) showed in their analysis that significantly different behaviour under part load conditions is expected based on the intake port design. It should be noted that applying either EIVC or LIVC had significant effect on the TKE, and therefore noticeable changes of flammability limit are expected which results in extended tolerance to air or EGR dilution.

Shortfalls of employing EIVC strategy, resulting in slower combustion were discussed by Ferrey *et al.* (2014), and the same conclusions based on the reduced TKE during ignition for EIVC were presented in the study of Scheidt *et al.* (2014) pointing to more favourable LIVC strategy from the burn duration perspective.

Number of investigations using numerical simulations discussed impact of employing non-conventional valve strategies on in-cylinder quantities and mixture formation, but Diana *et al.* (2001) presented that employing EIVC strategy, instead of traditional throttling, resulted in increased in-cylinder TKE levels and tumble ratio. Such conclusions did not agree with the other studies discussed earlier, pointing towards the fact that use of EIVC strategy remains engine specific and its benefits or shortfalls cannot be easily generalised for any engine.

In addition to producing TKE profiles, Jovanovic *et al.* (2011) provided an insight into spatial distribution of TKE within the cylinder when using cylinder deactivation i.e. running the engine with one of the valves completely shut. Figure 1.26 shows y and z plane views depicting

that one valve actuation gained significantly higher TKE values (various scales for two plots) and shifted the highest TKE region closer to spark plug which was beneficial for combustion.

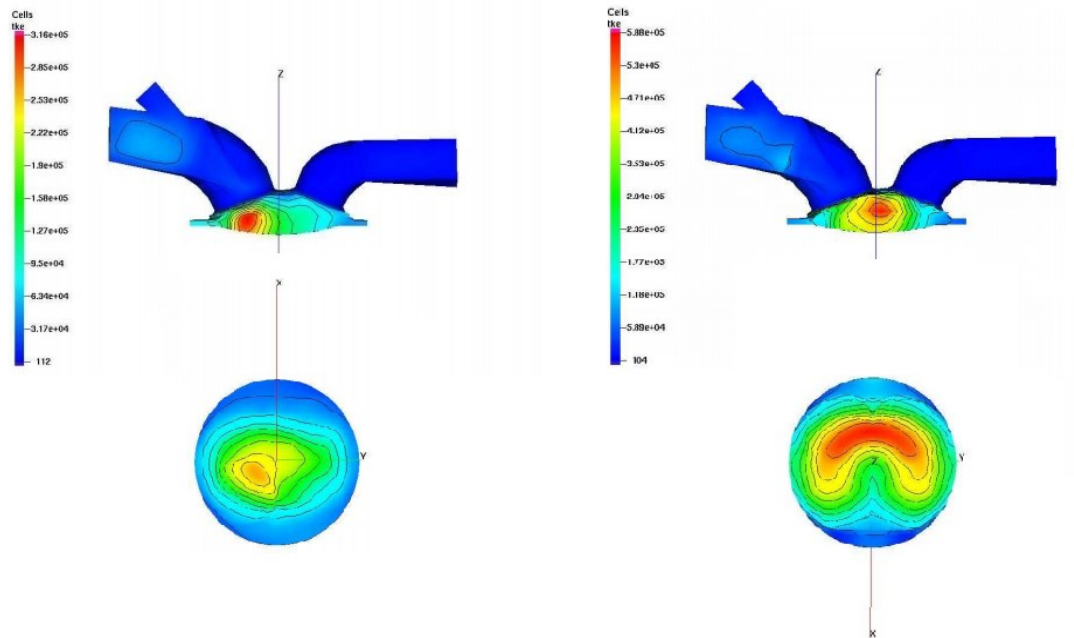


Figure 1.26 – Spatial distribution of in-cylinder TKE for two valve and one valve engine operation (Jovanovic *et al.*, 2011)

Strategy of Late Intake Valve Opening (LIVO) was introduced to reduce the amount of valve overlap and opening of intake valves can be retarded well into the induction stroke. It was found that due to unfavourable pumping losses attributed to this strategy, i.e. retarded opening of the inlet valves collapses vacuum generated by the piston travelling from TDC to BDC after exhaust valves closed, which is an irreversible process that carries pumping loss penalty. LIVO strategy was predominantly used in the studies discussing reduction of cold-start emissions (Pischinger *et al.*, 2000, Saber *et al.*, 2002 and Trask *et al.*, 2003). THC production at 20°C could be reduced up to 60% according to these studies.

Early attempts of implementing the technique of increasing the charge motion by operating the Port Fuel Injection (PFI) SI engine with differential valve lift, resulting in an increased lean operating limit and a reduction in Brake Specific Fuel Consumption (BSFC) were reported by Horie *et al.* (1992). In this study authors used VTEC-E Honda engine to arrive at conclusion

that significant, 12% fuel consumption benefit could be realised at part load operation, however they recommended switching to traditional two valve operation at 2500 RPM for high load (WOT) and 3200 RPM at lower loads. VTEC-E engine was also used by Hardalupas et al. (1995) to study injection timings influence on the fuel distribution, and it was shown that lifting previously deactivated valve incrementally admits fuel which would otherwise reside in the port. Such study which looked into formation of fuel-rich clouds for PFI engine is believed not to be well correlated with mixture formation of modern DI engines, which can be controlled via injection timing and duration within the cylinder.

Mixture preparation strategies using various combustion systems achieved improved combustion stability (mostly at low loads) by introducing combination tumble and swirl motions. More et al. (2011) combined experimental data with CFD simulations and reported improvements in combustion stability and lowering BSFC at low engine loads. It was also shown through CFD simulations by Ramasamy et al. (2016) that TKE and swirl motion were increased when operating asymmetric valve lift in a DI Compressed Natural Gas engine under Wide Open Throttle (WOT) condition.

However, based on all of the above, there is an ambiguity as to how valve deactivation affects in-cylinder temperature and associated heat transfer effects under single valve operation and no clear understanding as to at which engine loads, the effects of valve deactivation stop being beneficial and what causes the aforementioned preference of using the strategy at lower engine loads.

1.6. Synopsis and present contribution

Recent efforts within engine development in automotive industry are mainly focused on challenges related to lowering fuel consumption, whilst meeting increasingly stringent exhaust emission standards. Fleet average targets for 2021 have been published by European Commission and lack of compliance will be penalised by the need to pay “excess emission premium” for each CO₂ g/km exceedance.

Increasing engine thermal efficiency is fundamental and four key categories were identified by Stokes et al. (2000):

- Exhaust energy reduction
- Heat loss reduction

- Friction reduction
- Reduction of gas exchange losses

Three of the above (excluding friction reduction) can be addressed through the means of charge dilution and corresponding combustion optimisation.

Charge dilution can be achieved through three various approaches:

- Increasing air/fuel ratio – running lean.
- Feeding external, cooled EGR.
- Trapping in-cylinder residuals (hot, internal EGR).

Extensive research was conducted up to date and benefits of running each diluent were described in detail. Whilst diluting mixture offers efficiency increase benefits due to lowering pumping losses, having cooler combustion and therefore lowering heat losses and suppressing NO_x formation, it also extends burn duration and has detrimental effect on combustion stability.

Implementation of heavy dilution techniques requires changes to engine design that can promote better flame propagation under less favourable conditions. Techniques involve increasing compression ratio, enhancing in-cylinder turbulence and providing higher energy ignition.

Most of the studies performed up to date were done on port fuel injected (PFI) engines. Shayler *et al.* (2000) claimed that gas/fuel ratio can be potentially extended from the levels of about 25:1 to 60:1 for DISI engines compared to PFI, however no proof was given.

In the reviewed literature tests were usually performed at either part load or full load conditions without comprehensive analysis of the full speed/load map. Ayala *et al.* (2006) performed testing at various conditions and found that there is optimum/limiting 10-90% burn duration that always corresponds to peak efficiency which is irrespective of any other parameter, however engine used in that study had a very basic, fixed valvetrain.

Studies focused on implementing Miller or Atkinson cycle for control of effective CR lack details on the valve operating strategies due to confidential character of engine calibration. It was found that EIVC or LIVC strategies depend on intake port design and tumble/swirl characteristics, therefore experimental study on JLR's AJ20 is required to establish most favourable valvetrain operation. There seems to be a general conclusion pointing towards EIVC being more beneficial at part-load operating conditions, mainly due to more favourable pumping losses offset (introducing valve throttling), whilst LIVO strategies are predominantly

used for cold-start operation to minimise engine out emissions, while suffering from pumping penalty. Also valve deactivation strategies investigated in the past failed to provide conclusive answers as to why they stop being beneficial at higher engine loads and are preferred for low and part load BSFC optimisation.

The main objectives of the work packages designed for this study were based on the literature review presented earlier and their relevance in aiding fundamental gasoline engine research that could be applied to engine development of Jaguar Land Rover who sponsored this project. Specifically, JLR provided a prototype thermal Single Cylinder Research Engine (SCRE) that had to be set up in a dedicated test cell along with all the peripheral and ancillary systems from the ground up. Therefore, objectives identified for this project were both of engineering design/application type, and related to fundamental understanding of in-cylinder thermo-physical processes:

- Set up the SCRE state-of-the-art test facility in such a way that it can be used flexibly for proof of concept combustion system engine testing, including prototype equipment. Industry standard data acquisition methods need to be reviewed and enhanced where necessary to suit testing of highly diluted mixtures and capability of very high sampling rates when necessary, particularly for ignition systems measurement and analysis.
- Automate SCRE data acquisition processes to ensure high repeatability and quality of experimental findings which is of paramount importance given that experimental data are then used to establish boundary conditions and validation for numerical simulations that were run in parallel in a separate PhD project under the same sponsorship agreement with JLR.
- Identify key parameters and understand dilution limits of the AJ20 combustion system (high tumble design) and test its tolerance to running across engine conditions representative of low and high loads across different engine speeds.
- Decouple the benefits and shortfalls of running the engine in single valve lift mode and provide fundamental understanding of the respective effects of the altered in-cylinder flow motion.
- Provide conclusive answers as to why single valve deactivation strategies are claimed to stop being beneficial at higher engine loads and are preferred for low and part load fuel consumption optimisation with parallel focus on pollutants formation and their sensitivity to running with single valve opening.

- Utilise the capabilities of a constantly variable valve lift system on the intake side and introduce various staggered lift operations to further optimise valve deactivation modes of operation, beyond typical single valve deactivation strategies. In effect, draw conclusions on which characteristics are mostly beneficial to increasing combustion stability, and hence increasing dilution tolerance of the engine.
- Design and commission a new data acquisition system for secondary current and voltage measurements that can be used on ignition systems of different designs to gain insights into delivered spark energy and duration for individual cycles.
- Investigate if claimed benefits of running advanced, high energy ignition systems with a capability of continuous discharge or multi-strike operation, are also applicable to the AJ20 combustion system under charge diluted conditions and assess their magnitude.
- Provide understanding of how enhanced in-cylinder charge motion brought about by valve deactivation strategies affects the ignition event, and whether synergies with continuous and multi-spark operations exist and can be utilised in practice.

Chapter 2

Experimental Set-Up

2.1. Preliminary Remarks

A Single Cylinder Research Engine (SCRE) test facility was set-up at Imperial College London as part of collaboration with project sponsoring company – Jaguar Land Rover – Centre of Excellence for Spark-Ignition Engine Combustion Research. A thermal engine with combustion chamber representative of Jaguar Land Rover AJ20 gasoline family was commissioned to carry out experimental work. An engine test bed facility was designed with focus on flexibility and tight control of all engine-operating parameters.

There was also a prototype optical engine with a combustion system representative of JLR's AJ-20 thermal engine and it was envisaged to be used for some parallel CFD validation and enhancing understanding of the combustion behaviour inside the thermal engine. Unfortunately, the optical engine was tied up in other JLR funded projects that were running in parallel at Imperial College, hence this optical engine was not used to the benefit of the current thesis and its operation is not described here.

2.2. Engine Configuration and Hardware

The SCRE was mounted on the seismic mass and coupled to an elastic motor – 65 kW DC dynamometer, controlled by UNICO drive unit. Torque was measured using the Tedea Huntleigh 615 compression load cell with tolerance on the torque reading of $\pm 0.25\%$ of full scale (calibrated to 125 Nm).

Dynamometer Services Group (DSG) delivered software and interface for engine control and data acquisition – DaTAQ Pro.

The engine was based on Ricardo's Hydra bottom-end and cylinder head was supplied by Jaguar Land Rover. Since its first installation in 2015, the engine went through four iterations/ rebuilds that are outlined in the Table 2.1.

Table 2.1 – SCRE specification changes throughout the builds (2015-2018)

Characteristic Parameter	Build No 1	Build No 2	Build No 3	Build No 4
Commissioning Date	Jan-2015	Jul-2015	Dec-2017	June-2018
Displacement Volume [cm ³]	499.02	499.02	499.02	499.02
Compression Ratio	10.620	11.006	10.860	11.272
Bore x Stroke [mm x mm]	83 x 92.23	83 x 92.23	83 x 92.23	83 x 92.23
Valvetrain	RFF (intake & exhaust)	CVVL intake & RFF exhaust	CVVL intake & RFF exhaust	CVVL intake & RFF exhaust
No of Valves / Cylinder	4 (2 Inlet, 2 Exhaust)			
IMOP [° CA aTDC]	144	161	161	161
IVC [° CA bTDC]	96	87	87	87
EMOP [° CA bTDC]	121	119	119	120
EVO [° CA aTDC]	124	126	126	125
Conrod length [mm]	152.72	152.72	152.72	153.00
Pin offset [mm]	11.5	11.5	11.5	11.7

Intake valve close position and exhaust valve open position were defined after calculation of the first and second derivative of the valve profiles. Valve lift curve and its second derivative are shown in Figure 2.1 below.

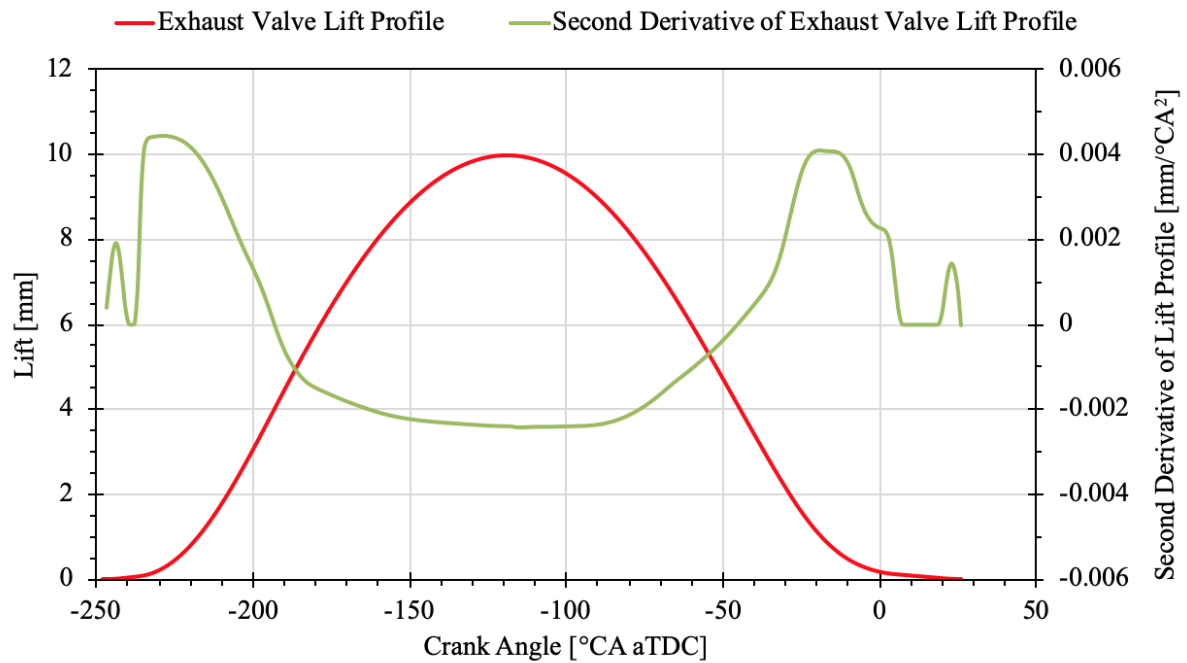


Figure 2.1 – Exhaust Valve Profile (RFF, Build No2) and its second derivative

Two aforementioned engine configurations are presented in Figures 2.2 and 2.3. These are not fully labelled for clarity purposes. Detailed schematic diagrams of engine sub-systems can be found later in this chapter.

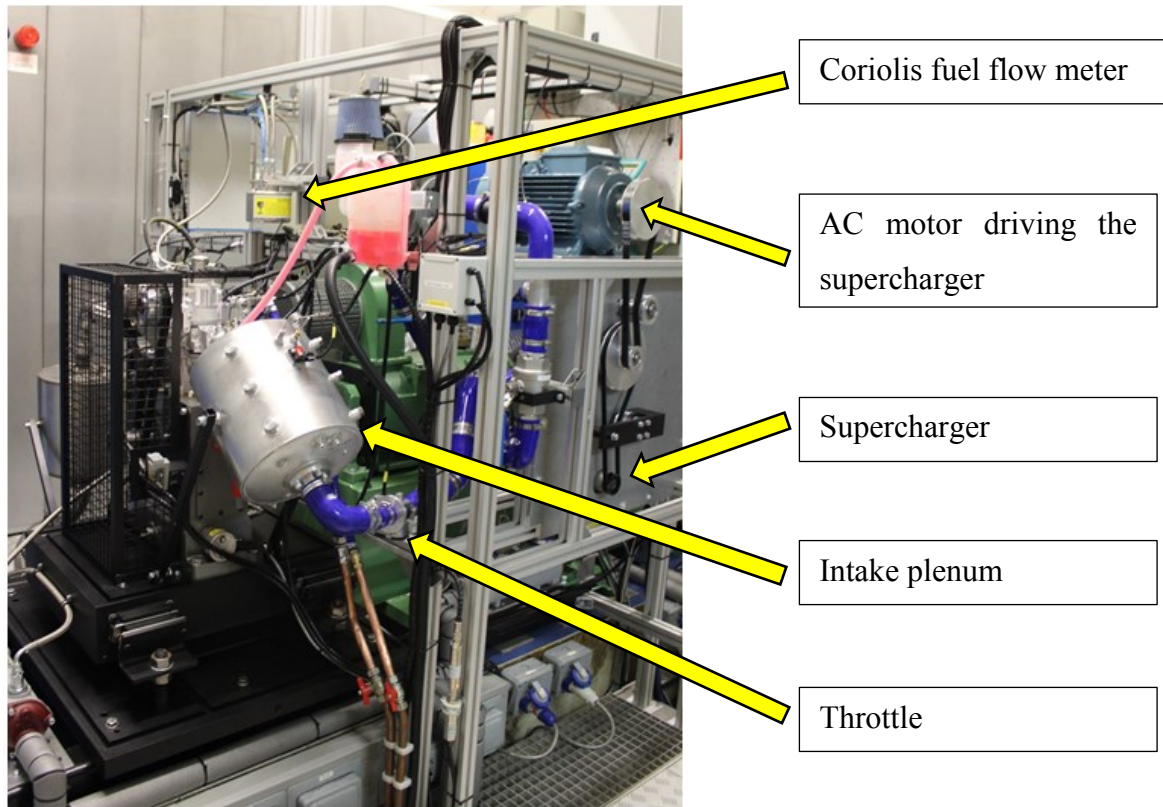


Figure 2.2 - Fully instrumented single cylinder engine in RFF (Build 1) configuration.

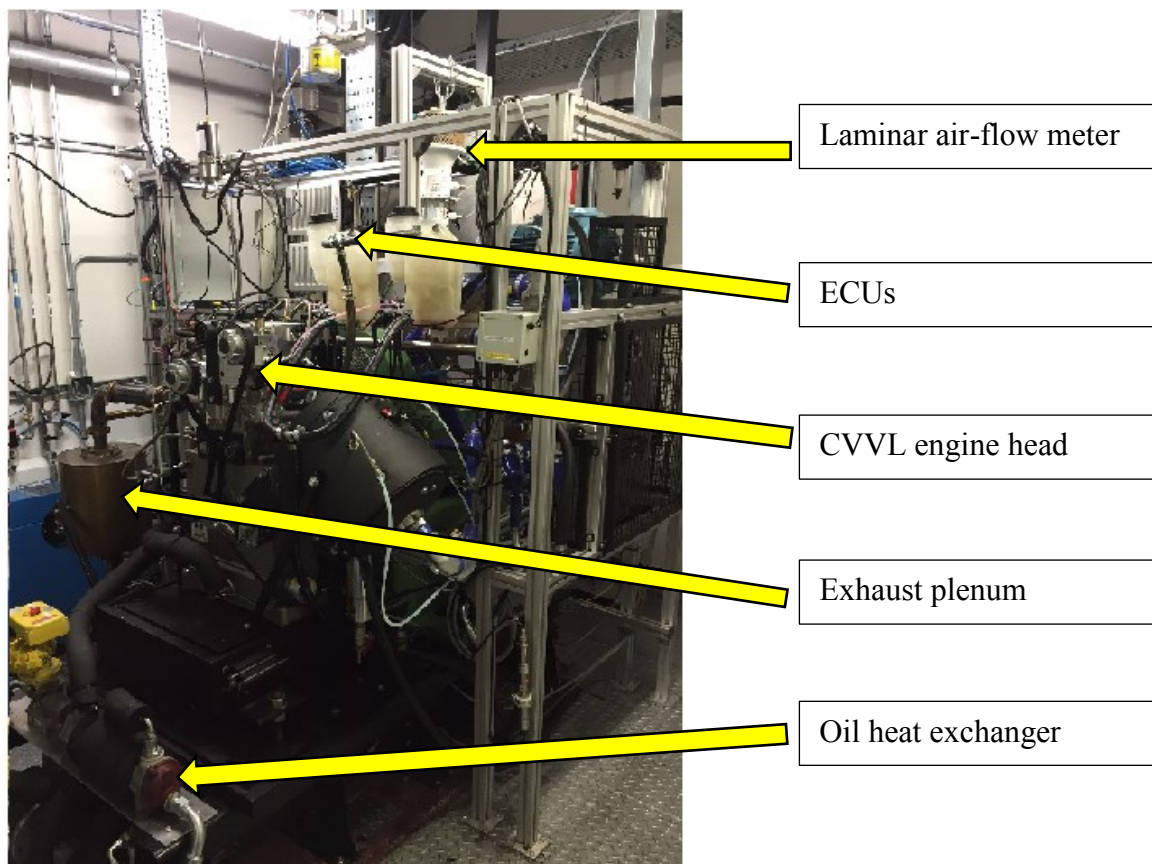


Figure 2.3 - Fully instrumented single cylinder engine in CVVL (Build 2) configuration

Each rebuild of the engine was successfully followed by the “break-in and power pass off” (BIPO) procedure aimed to friction stabilise the system and ensure that any measurements taken afterwards were repeatable and consistent.

2.2.1. Valvetrain

The valvetrain used on the engine changed between first and subsequent builds. Originally engine was installed with Roller Finger Follower (RFF) type valvetrain on both intake and exhaust sides. RFF system was equipped with phasers that gave 50 °CA range on both the intake and exhaust. Intake Maximum Opening Point (IMOP) could be advanced by 50°CA, whilst Exhaust Maximum Opening Point (EMOP) could be retarded by 50°CA relative to the base position, creating valve overlap (Figures 2.4 and 2.5).

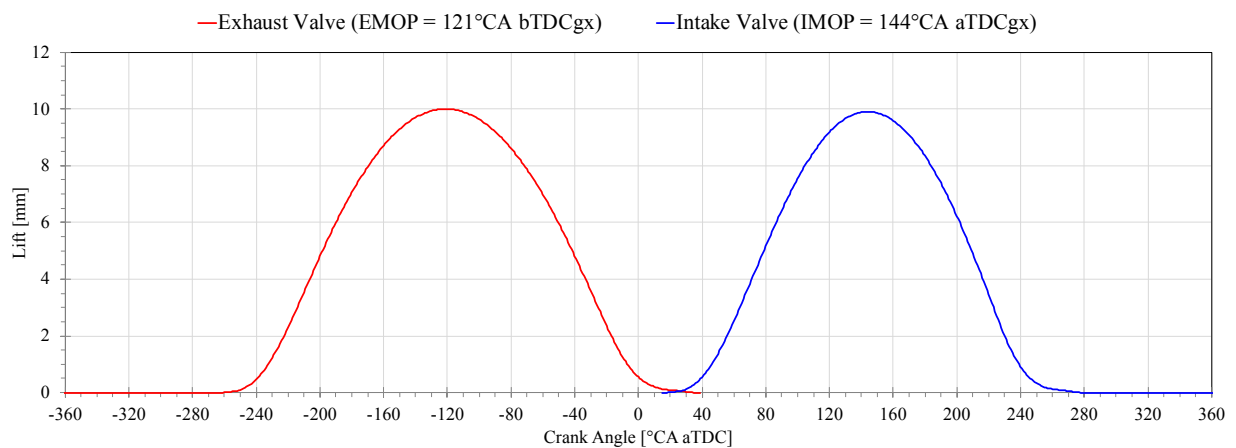


Figure 2.4 – RFF Valve Profiles with minimum overlap

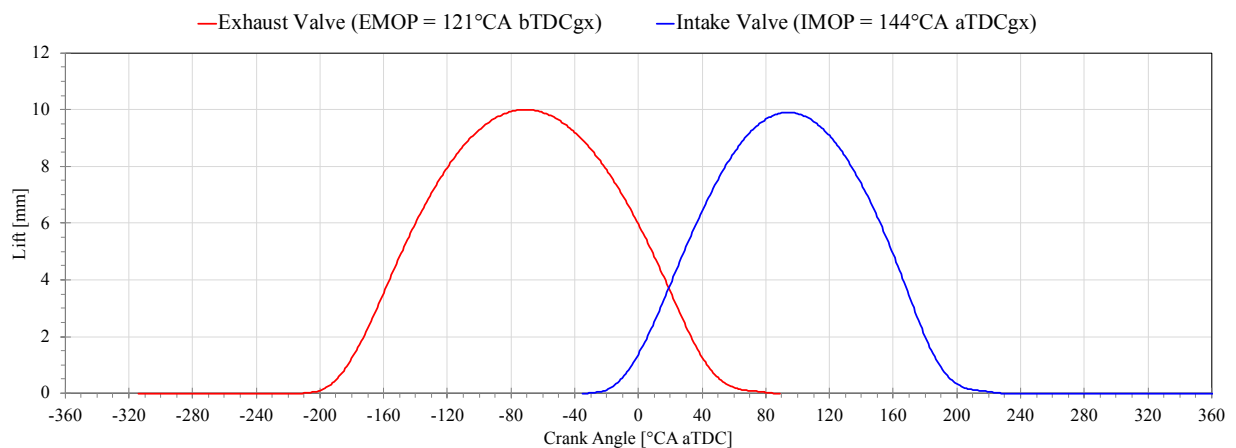


Figure 2.5 – RFF Valve Profiles with maximum overlap

During upgrade to CVVL on the intake side, exhaust remained the RFF type, but its EMOP was advanced by 2 °CA. Intake and exhaust ports design changed between RFF and CVVL phase and they are presented in Figure 2.6.



Figure 2.6 – Illustration of intake and exhaust ports design change between Build 1 and 2

Valve profiles of the CVVL head have also been significantly changed. Their duration was extended, whilst lift was lowered from 10 mm to 9.5 mm. Its base position is shown in Figure 2.7. System retained 50 °CA phasing capability of the RFF system that it replaced.

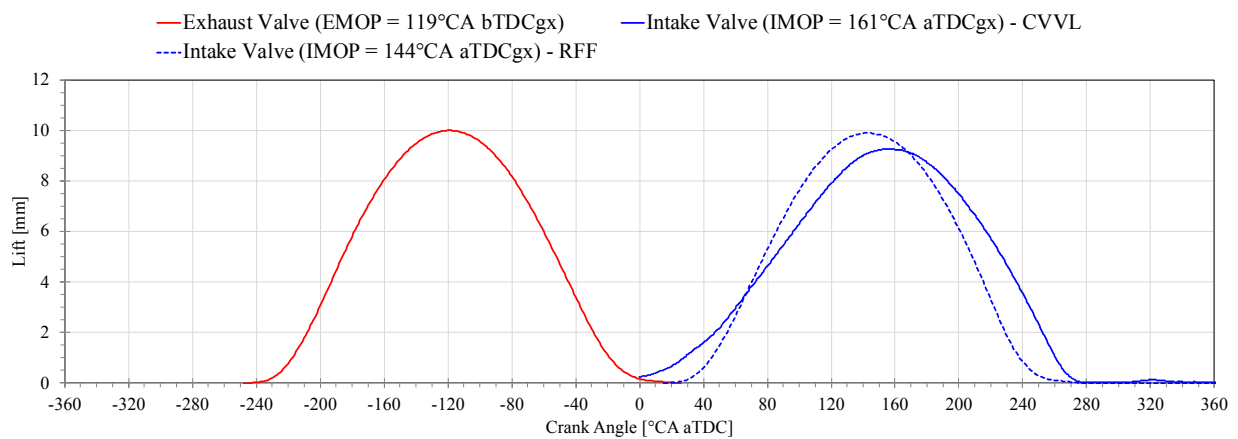


Figure 2.7 – CVVL Valve Profiles with minimum overlap

The CVVL system installed on the intake side was the electro-hydraulic type and was supplied by INA. It allowed for independent control of the valve lift between two intake valves (known

as staggered lift operation). The system was of the same type as Fiat MultiAir and Figure 2.8 illustrates modes in which it could be operated.

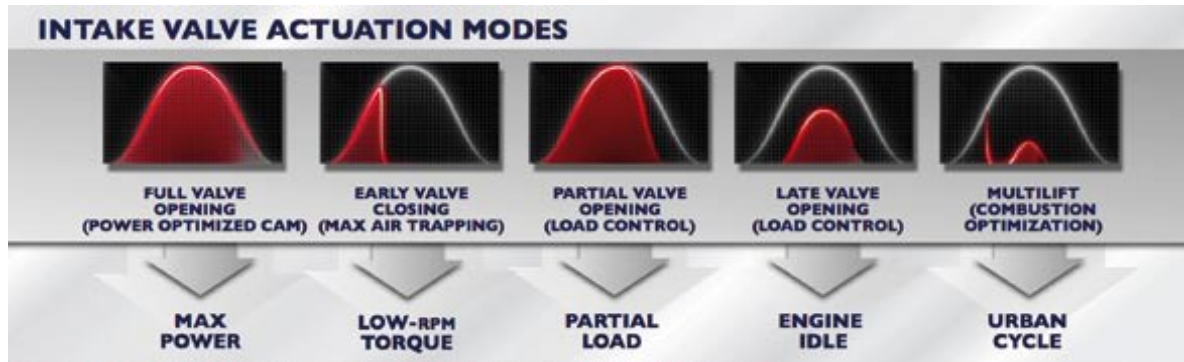


Figure 2.8 – MultiAir® system operation. (Bernard et al., 2009)

Valve lifts of both front and rear intake valves were monitored in real time during SCRE operation using Sensitec sensors that were fed to the indicating equipment. Signals were resolved to 0.5°CA . It allowed for live adjustment and precise control when targeting specific lifts. Signals were also crucial for calculations of heat release which are discussed later in this chapter.

2.2.2. Ignition Systems

Spark Plug used for all studies was a J-type ground electrode design, heat grade 8 NGK (H4P3-12405-ACA) with nominal gap size of 0.65 mm and tolerance of ± 0.05 mm. Gaps were tested prior to engine installation and adjusted if needed. Spark plug orientation was fixed for all tests and adjusted with 0.1 mm shims if needed; its ground electrode was on the exhaust valves side of the engine and parallel to the tumble motion (shown in Figure 2.9).

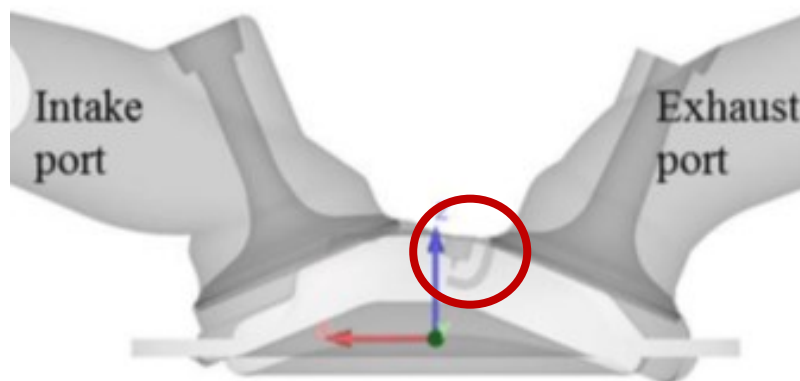


Figure 2.9 – Spark plug orientation in AJ-20 SCRE engine

Three ignition systems were used throughout the duration of the studies performed on SCRE.

The first system was the stock system provided by JLR, and it was a standard coil on plug (CoP) used on production engines (Part No LR035548). It was powered up and triggered by the harness connector from a MAHLE ECU (described later in this chapter). Its dwell time was set automatically depending on the running condition, as it was using live interpolation of INCA built-in map based on the engine speed and ECU voltage (shown in Figure 2.10).

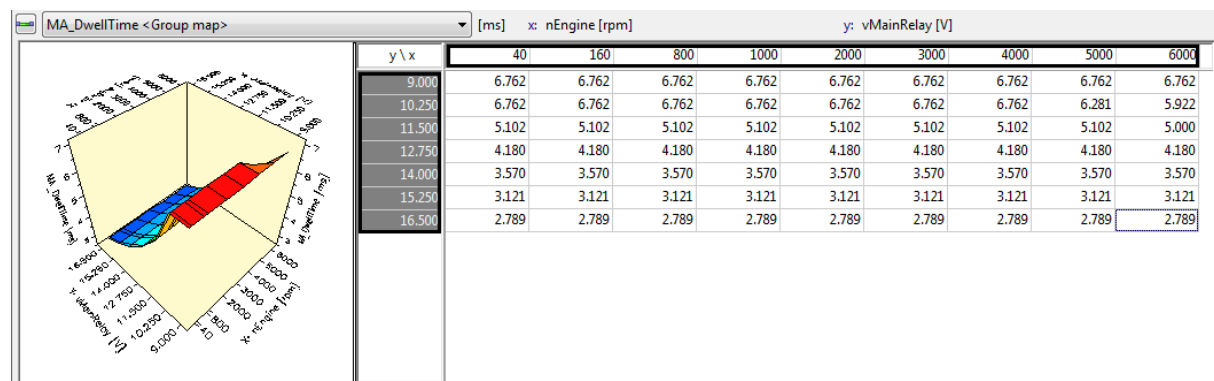


Figure 2.10 – Dwell time map used for stock coil ignition system

Second system was the Controlled Electronic Ignition (CEI) supplied by Bosch – their prototype high performance Transistor Coil Ignition (TCI) system. Compared to conventional ignition coil, it offered additional control variables for the parametrisation and claimed energy delivery to the spark plug within the 50-300 mJ range. It offered single, highly customisable operation mode that could effectively achieve elongated spark with current maintained at over 80 mA. It was provided with remote control which allowed for setting of the following parameters:

- Crossover [-0.1,0, 0.1, 0.2 ms]
- Power Level [Single Spark, 50%, 75%, 100%]
- Dwell Time [0.4, 0.6, 0.8, 1.0, 1.1, 1.15, 1.2, 1.3, 1.5, 1.5, 1.6 ms]
- Boost Time [0.3, 0.5, 0.8, 1.0, 1.5, 2.0, 2.5, 3.0, 4.0, 5.0 ms]

It was commissioned in collaboration with Bosch engineers and required modifications to the ECU harness. Spark event was timed by ECU signal (labelled as *Control Input* in Figure 2.11 showing schematic diagram of the system), therefore it did not interfere with other systems which relied on automated adjustment of ignition angle (described later in this chapter).

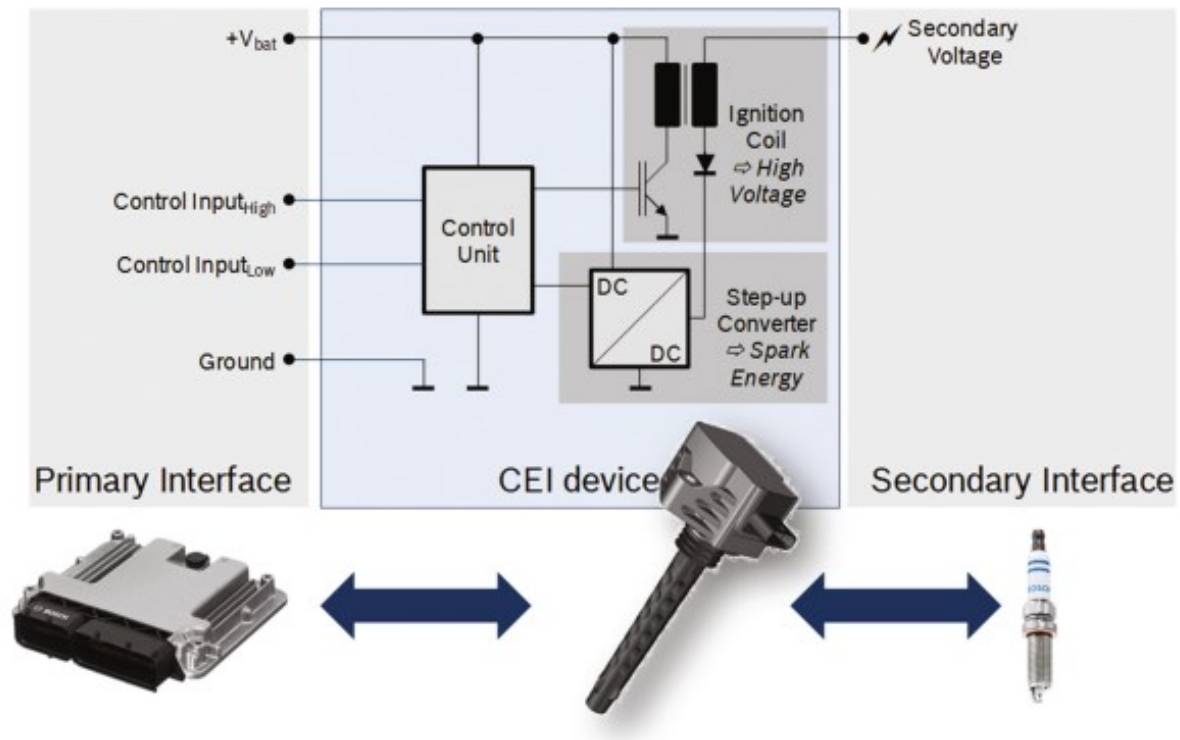


Figure 2.11 – Block diagram of Bosch CEI system set-up (Brandt et al., 2016)

The third system was the Dual Coil Ignition (DCI) system supplied by Borg Warner (BW). It enabled altering the primary coil energy and recharge event duration. Primary current could be set within the window of 9-12 A and recharge period could vary between 0-5 ms (where applicable).

It replaced standard Coil on Plug (CoP) with a Dual Coil Pack (DCP) design. Two coils within the DCP could be charged and discharged either simultaneously or alternately to control the spark discharge. By superposing of both high voltage outputs an uninterrupted spark could be achieved. DCI system could be operated in 4 modes:

- *Single strike* – one coil is charged until the designated primary current is reached and then discharged
- *Dual strike* – both coils are charged and then discharged simultaneously (50% offset is applied to the second coil, so it starts charging only half way through the first coil charging)
- *Multistrike* – primary current limit is set, as is the recharge time, which controls the duration of the recharging period. The charging and recharging sequence is closed loop

controlled, i.e. measured primary and secondary currents are used to start charge/discharge events when thresholds are reached.

- *Dual Coil Continuous* – coils are charged and discharged alternately for the duration of the set recharge period, with overlapping of discharge so that spark does not extinguish (unlike in multistrike mode)

Control of the DCI system was achieved through DCI Toolbox software supplied by Borg Warner. It communicated with ICM control unit via CAN interface. ICM Control Unit was wired to engine ECU which provided ignition trigger signals to ensure spark timing was preserved (similarly to Bosch CEI system set-up). Schematic of the BW DCI system is shown in Figure 2.12.

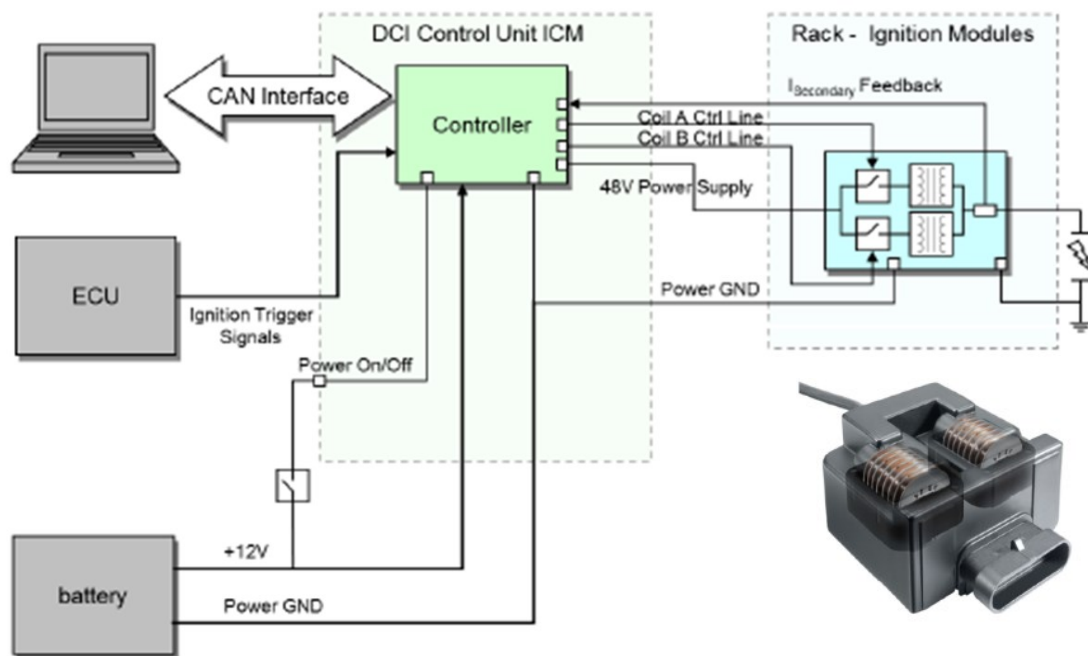


Figure 2.12 – Block diagram of Borg Warner DCI system set-up

It should be noted that both Bosch and Borg Warner systems were powered by an external power source (12V Power Supply Unit), hence engine performance was not affected by their power draw. Both systems were also interfaced with Mahle ECU which sent the ignition signals, however dwell time maps were disactivated in INCA (flat-mapped to 1 ms) and both systems had internal control over dwell times to achieve user-specified operation.

2.2.3. Fuel System

Fuel injector used for all studies was the centrally mounted Bosch DI (EB2) six-hole type H4P3-9F593-EA1. High pressure fuel (up to 200 bar) was delivered to the injector by the Bosch pump (0 261520 198) driven by the exhaust camshaft. Fuel delivery to the high-pressure pump is described later in this chapter, together with the fuel conditioning loop.

All tests were run on EURO 6 E10 95 RON Reference fuel (supplied by Haltermann Carless). Sample fuel specification sheet (representative for all batches used throughout all tests run for the purpose of this thesis) was attached in the Appendix section.

2.3. Measuring Equipment and Data Acquisition Methods

2.3.1. Temperature

Temperature readings at multiple points were provided by the use of K-Type Thermocouples (Labfacility MA-ISK-S15-150-P2-3.0-C7-SP and MA-ISK-S30-150-P2-3.0-C7-SP) and Platinum Resistance Thermometers (PRTs – Labfacility RAA-S4A-3.0-150-P1-3.0-C7-4P/PLUG). PRTs were used on the closed loop control circuits due to their superior accuracy and higher resistance to measurement drift over time.

Locations of the sensors are listed in the table 2.2 below:

Table 2.2 – Location of temperature sensing equipment

K-Type Thermocouples	Platinum Resistance Thermometers
Test Cell (ambient)	Air intake – upstream of LFA
Air intake filter	Fuel pump outlet
Supercharger inlet	Intake plenum
Supercharger outlet	Engine water cooling jacket – outlet
EGR cooler – water in	Oil gallery
EGR cooler – water out	
Exhaust gas – exhaust runner	
Oil sump	
Charge Air Cooler inlet	
Charge Air Cooler outlet	
Air intake runner	
EGR cooler – gas in	
EGR cooler – gas out	
Fuel pump inlet	
Kistler coolant – manifold inlet	
Kistler coolant – manifold outlet	
Engine water cooling jacket – inlet	

All thermocouples were subjected to annual calibration using the Druck DPI620G to ensure consistency of measurements throughout the duration of the project.

2.3.2. Pressure

Pressure readings were taken by either high precision, research grade absolute pressure transducers (GE Druck A5013-TB-A3-CA-H0-PB) that were used to monitor fluid pressures and act as a “slow” response, 10 Hz channels being fed to DSG data acquisition system or “fast” response channels that were crank angle resolved and formed part of the indicating equipment described in the next subsection.

Locations of “slow” response Druck transducers are listed in the table 2.3 below:

Table 2.3 – Location and type specification of GE Druck pressure transducers

Transducer Location	Transducer Type / Range
Barometric Pressure (ambient)	RPT 410V / 600 to 1100 mbar absolute
Air intake – upstream of LFA	PMP 5013 / 800 to 1200 mbar
Crankcase	PMP 5013 / 800 to 1200 mbar
Supercharger outlet	PMP 5013 / 5 bar absolute
Charge Air Cooler outlet	PMP 5013 / 5 bar absolute
Intake plenum	PMP 5013 / 5 bar absolute
Intake runner	PMP 5013 / 5 bar absolute
Exhaust runner	PMP 5013 / 5 bar absolute
Engine water cooling jacket – inlet	PMP 5013 / 5 bar absolute
Engine water cooling jacket – outlet	PMP 5013 / 5 bar absolute
Fuel circuit (upstream of HP pump)	PMP 5013 / 10 bar absolute
Oil gallery	PMP 5013 / 10 bar absolute
Oil sump	PMP 5013 / 5 bar absolute
Oil – Piston Cooling Jets	PMP 5013 / 10 bar absolute
Oil – engine head	PMP 5013 / 10 bar absolute

All Druck pressure transducers were subjected to annual calibration using the Druck DPI620G to ensure consistency of measurements throughout the duration of the project.

2.3.3. Indicating Equipment

Experimental set-up used AVL's IndiSet Advanced Gigabit 642 indicating equipment which was configured as a 16-channel system. It allowed for simultaneous sampling with 16 analogue differential inputs with individual 14Bit/ 800 kHz ADC per channel.

All analogue signals coming to IndiSet equipment were connected through BNC ports that could accommodate +/- 10V signals. They were resolved to crank angle degrees thanks to AVL's 365C Crank Angle Encoder mounted on the front of the engine crankshaft.

AVL's crank encoder (365C), with angle mark resolution of 0.5 °CA (up to 0.1 °CA by means of multiplication) provided more robust readings than hall effect sensor used as part of the DaTAQ Pro system and was used as a main engine speed reading device for any tests.

TDC was set using the AVL capacitive TDC Sensor 428. Dynamic determination of the TDC when motoring the engine used the “horizontal-cut-principle” shown in Figure 2.13. It ensured that angular TDC position was accurate to within less than +/- 0.1 °CA.

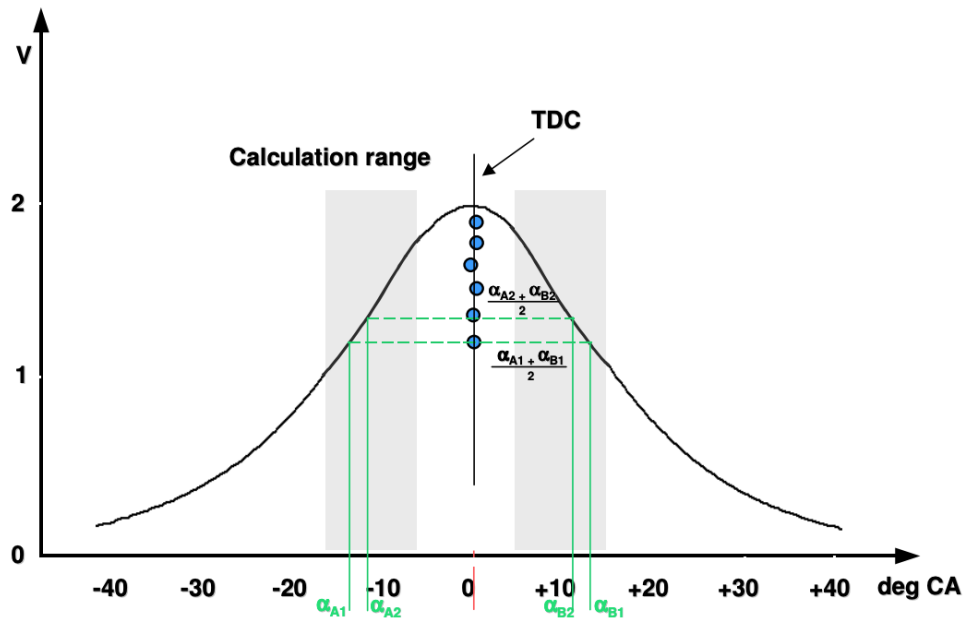


Figure 2.13 – Horizontal-Cut-Principle for TDC determination (AVL, 2010)

Hardware was operated through AVL's IndiCom software which performed combustion analysis and controlled data acquisition process. It was interfaced with DaTAQ Pro system to enable remote recording triggering.

The following pressures were measured using the corresponding Kistler sensors:

Table 2.4 – Specification and location of Kistler sensors

Sensor Type and Location	Kistler Part No	Maximum Error (mbar)
In-cylinder piezoelectric	6041B (water cooled)	± 9.50
Intake runner piezoresistive	4005BA5FV200S (water cooled)	± 0.91
Exhaust piezoresistive	4075A10V200S (water cooled)	± 1.22

All three sensors outlined above were water-cooled and rigorous control of running them at 50°C was achieved by using the dedicated Kistler's conditioning system for water-cooled pressure sensors (Type 2621F). All signals generated by the sensors were also passed through Kistler 4665 charge amplifying rack with units 5064B22 and 4665 used for piezoelectric and piezoresistive sensors respectively.

6041B piezoelectric sensor was mounted flush with the combustion chamber surface.

In-cylinder pressure sensors of piezoelectric type (6041B) require zero level correction (pegging) since they only effectively measure pressure difference during each engine cycle (Δp). Absolute pressure level is then needed at given crank angle and it is used as a reference point for subsequent visualisation and analysis.

Indicom software can either use *Thermodynamic Zero Level Correction* function or reference in-cylinder transducer to the intake or exhaust manifold pressures measured by piezoresistive transducer (accurate, absolute pressure reading).

Thermodynamic Zero Level Correction algorithm assumes adiabatic compression between crank angle positions X_1 and X_2 with a constant polytropic coefficient. The absolute, reference pressure level is then calculated from the difference in pressure between the two points on the compression curve using the predefined, constant polytropic coefficient. Calculation method is presented below:

X_1 first angle value (set to -100°CA)

X_2 second angle value (set to -65°CA)

$$p_1 = \frac{\Delta p}{\left(\frac{V_1}{V_2}\right)^\gamma - 1} \quad (2.1)$$

p_1 pressure at CA X_1

V_1 volume at CA X_1

V_2 volume at CA X_2

γ ratio of specific heats (C_p/C_v)

Δp pressure difference between X_2 and X_1

X_1 and X_2 are predefined points on the compression curve, however they can be set manually to different values if needed. Typically, *Thermodynamic Zero Level Correction* algorithm runs with X_1 and X_2 set to -100°CA and -65°CA respectively and $\gamma=1.35$ is recommended to be used for DI gasoline engines. Use of this algorithm is industry standard practise, especially on multi-cylinder development engines which are only equipped with in-cylinder Kistler sensors.

However due to the fact that SCRE was commissioned to run with heavily diluted mixtures that can significantly alter γ values, industry standard approach was deemed not accurate and not fit for purpose when operating under heavily diluted conditions and it did not meet objectives set for this study.

An investigation was launched to determine the best pegging method which would not assume a fixed polytropic compression coefficient, γ and use referencing of the signal generated by the one of the piezoresistive sensors.

Range of conditions were tested (Table 2.5) to understand the relation between intake, in-cylinder and exhaust pressure traces.

Table 2.5 – Engine running conditions used in the study determining best in-cylinder pressure trace pegging method

Engine Speed [RPM]	Plenum Pressure [kPa]	Valve Overlap [°CA]	Valve Lift Mode	Motoring / Firing
1500	53.5 – 101.3	0 - 0 25 - 25	Full Lift	Firing & Motoring
1500	101.3	0 - 0	EIVC	Firing & Motoring
2500	55.0 – 95.0	0 - 0	Full Lift	Firing & Motoring

It was found that both intake manifold pressure signal and in-cylinder pressure signals plateaued after IMOP irrespective of engine speed, load (reflected in plenum pressure) and valve overlap when operated in full valve lift mode. It can be seen in Figure 2.14, where intake runner pressure trace was plotted together with raw in-cylinder pressure. Raw in-cylinder signal was then pegged to the intake runner trace at the most stable point which was found to be halfway between IMOP and IVC. To ensure that the method was not affected by electrical

noise, any random spikes and the effect of digitization error was minimised, values of both signals were averaged over $\pm 10^\circ$ CA from the target point, and then pegged together. Standard options of the IndiCom software were however limited to setting a fixed pegging point at given crank angle. It was identified as a significant risk to data acquisition method, especially for tests which required phasing the intake camshaft, effectively changing location of the optimum plateau described earlier. Engine operator could easily forget setting the value for each test point. Re-setting the pegging angle would also have negative impact on time it would take to record data. Having Sensitec signals (described in section 2.2.1) available in IndiCom environment, it was decided that running edge detection macros in real time could automatically identify IMOP and IVC. They were then used to calculate a pegging angle. Macros used for that purpose are presented in Figure 2.15.

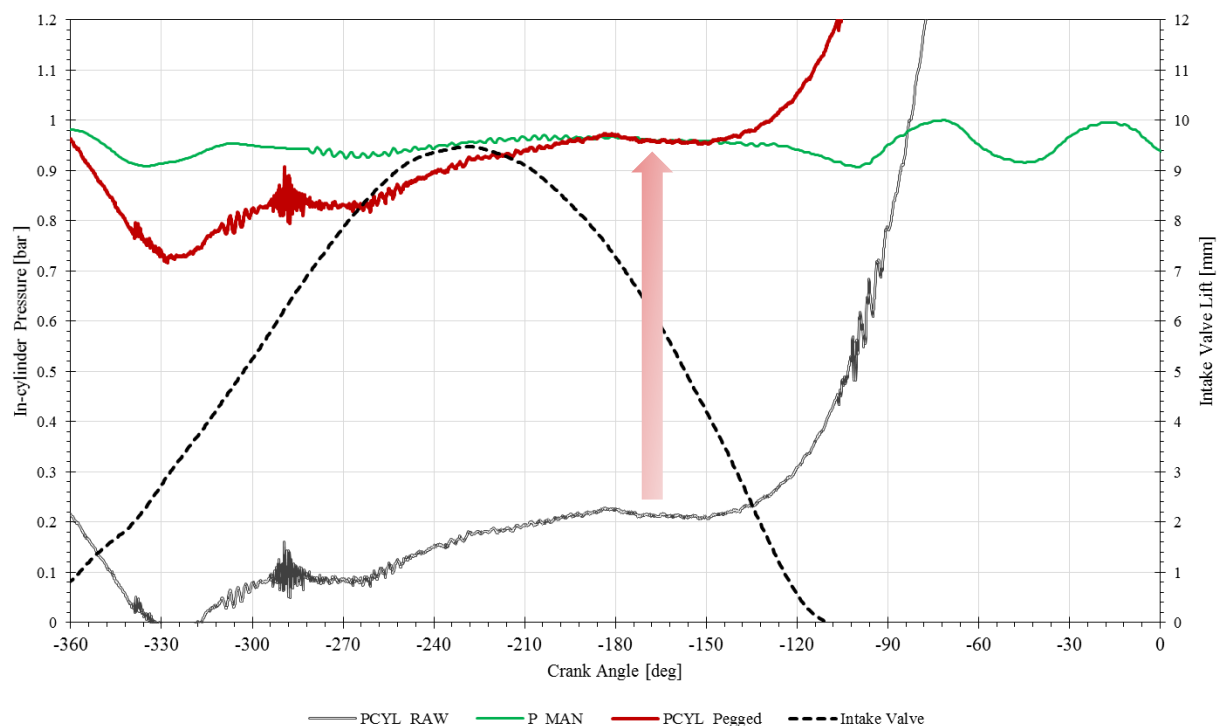


Figure 2.14 – Example of pegging in-cylinder pressure signal to intake runner trace halfway between IMOP and IVC at 1500 RPM, 95 kPa plenum pressure

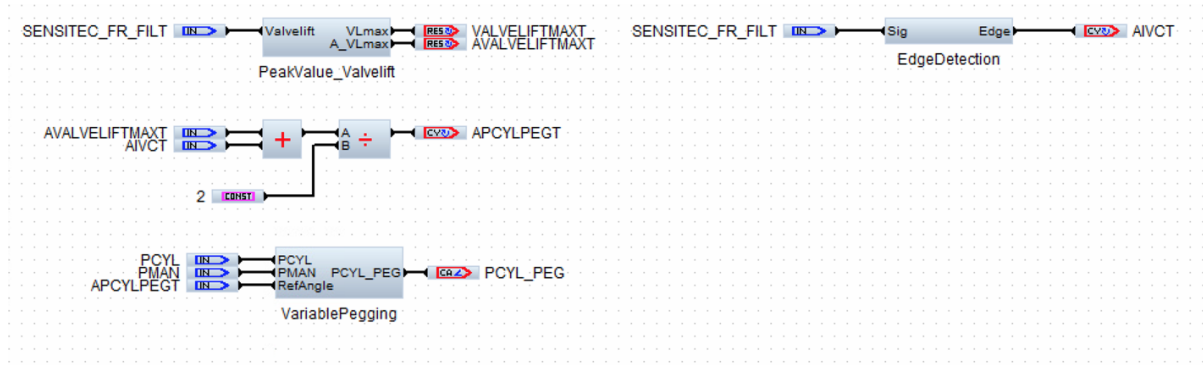


Figure 2.15 – IndiCom macros used for in-cylinder signal pegging

Once the method was selected and validated for full lift operation on the intake valves, an alternative approach was needed for instances where load control was achieved through valve throttling and running CVVL system in Early Intake Valve Closing (EIVC) mode. During such operation pressure inside the cylinder was significantly lower than in the intake plenum and the method described earlier would not work.

Via extensive comparison of in-cylinder, intake and exhaust pressure traces, it was found that in-cylinder pressure signal can be pegged to exhaust signal at its EMOP, using the same calculation window of $\pm 10^\circ\text{CA}$. Accuracy target of 0.02 bar was achieved. CFD simulations of engine breathing process presented in Figures 2.16 and 2.17 illustrate how pressures differ within the intake runner once the EIVC strategy is employed, however exhaust pressure remains unaffected and can be matched to in-cylinder values at EMOP. It should be noted that CFD work was outside the scope of this thesis and results based on experimental boundary conditions were provided by A. Kountouriotis. Further details regarding the CFD set-up and models validation can be found in his thesis (Kountouriotis, 2019).

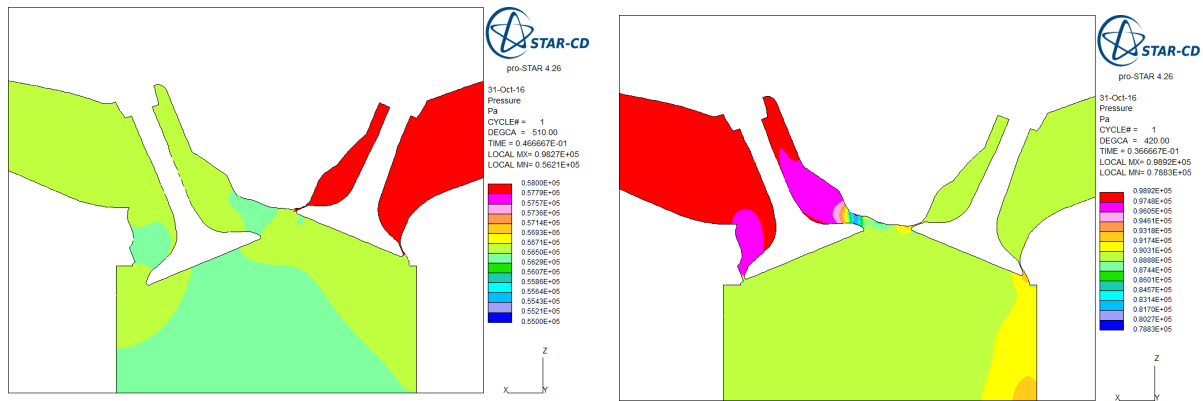


Figure 2.16 – Comparison of intake runner vs in-cylinder pressures achieved for full lift and EIVC valve opening strategies at 1500 RPM, 55 kPa plenum pressure (or equivalent for EIVC); both valves at their IMOP

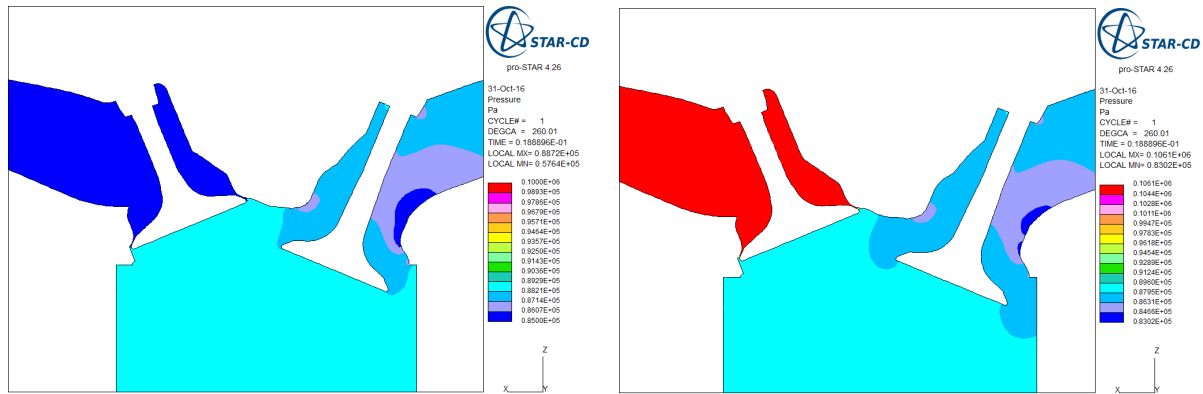


Figure 2.17 - Comparison of exhaust runner vs in-cylinder pressures achieved for full lift and EIVC valve opening strategies at 1500 RPM, 55 kPa plenum pressure (or equivalent for EIVC); both valves at their EMOP

This method however had one significant drawback compared to the one using intake runner pressure signal. Exhaust valve lifts were not measured using Sensitec sensors, so any tests run in low lift operating strategy on the intake valves required manually setting the pegging angle matched to exhaust camshaft phasing. Therefore, pegging to intake runner pressure signal was introduced as a base method which required changing only for test using valve throttling for load control.

Heat release calculations were performed on-the-fly by the AVL Indicom software. It ran *Thermodynamics 1* model which uses in-cylinder pressure readings as an input.

Real-time calculations are called *Fast Heat Release Calculations* and they traditionally use constant polytropic coefficients. Standard results which are output by the software include the following:

Result Curves

dQ Heat release (per deg. CA over crank angle)

Int Cumulated heat release (over crank angle)

Result Values

MBF5% Crank angle value for 5% mass burned fraction

MBF10% Crank angle value for 10% mass burned fraction

MBF50% Crank angle value for 50% mass burned fraction

MBF90% Crank angle value for 90% mass burned fraction

The polytropic coefficients for both compression and expansion are assumed to be constant, however they have different values for the compression and expansion phases and thereby recreate the actual fact that the burned gas is characterized by a lower polytropic coefficient than the fresh gas.

This method is however fouled when subject to running with diluted charge. As the dilution percentage increases, polytropic coefficient of the mixture that undergoes compression increases which is not accounted for by the code. After identifying this issue course of action has been determined to address it. Polytropic coefficients for both compression and expansion were calculated for each cycle using the in-cylinder logP vs logV pressure trace. Calculation windows were based on the work presented by Stone and Green-Armytage (1987) and Ball *et al.* (1998). It was computed from mid-point of IVC and ignition angle to ignition angle for compression polytropic coefficient and 10°CA after P_{MAX} to 60°CA aTDC for expansion. Ignition signal from ECU harness was fed to IndiSet to automate the process. Calculations were performed in real-time for every cycle by macros presented in Figure 2.18.

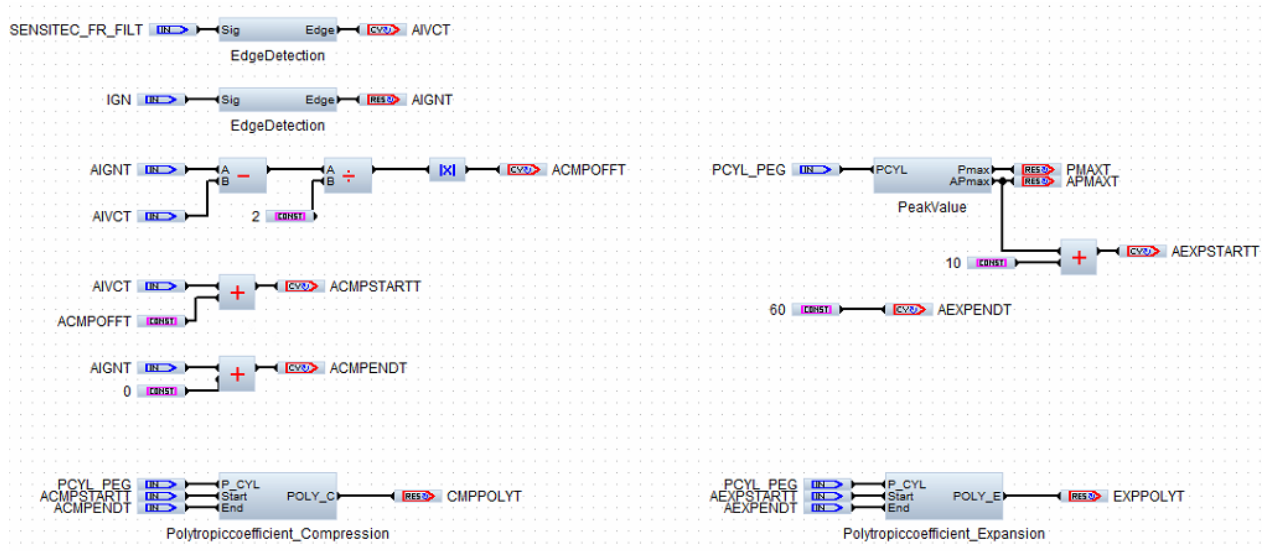


Figure 2.18 - IndiCom macros used for real-time calculations of compression and polytropic coefficients

A simplified process (built into AVL software) was used to calculate the heat release which computes the energy effectively delivered to the gas from the cylinder pressure. Surface losses (approx. 6-20%) are neglected, the displayed heat release is therefore accordingly lower than the actual energy released. The calculation is based on the first law of thermodynamics.

$$Q_i = \frac{C}{k-1} [k \cdot p_i \cdot (V_{i+n} - V_{i-n}) + V_i \cdot (p_{i+n} - p_{i-n})] \quad (2.2)$$

- n Interval (set to 0.1° CA)
- k Polytropic coefficient
- p Cylinder pressure
- V Volume
- C Constant (=100 due to conversion to SI units)

Due to vaporization effects the heat release curve will typically adopt negative values before combustion for direct injection gasoline and diesel engines. The start of combustion (SOC) is taken as the angle value dQ becomes positive again. The algorithm calculates backwards from the heat release peak down to the zero level.

Once heat release curves and points for 5%, 10%, 50% and 90% MFBs were computed, results were used to calculate respective burn durations. Macros designed for that are presented in Figure 2.19. It should be noted that heat release calculation (Equation, 2.2) used variable

polytropic coefficients. This method was introduced to IndiCom environment by AVL as a result of collaboration during the course of this study. It is not offered in the commercial version of the software under the standard library of macros and functions.

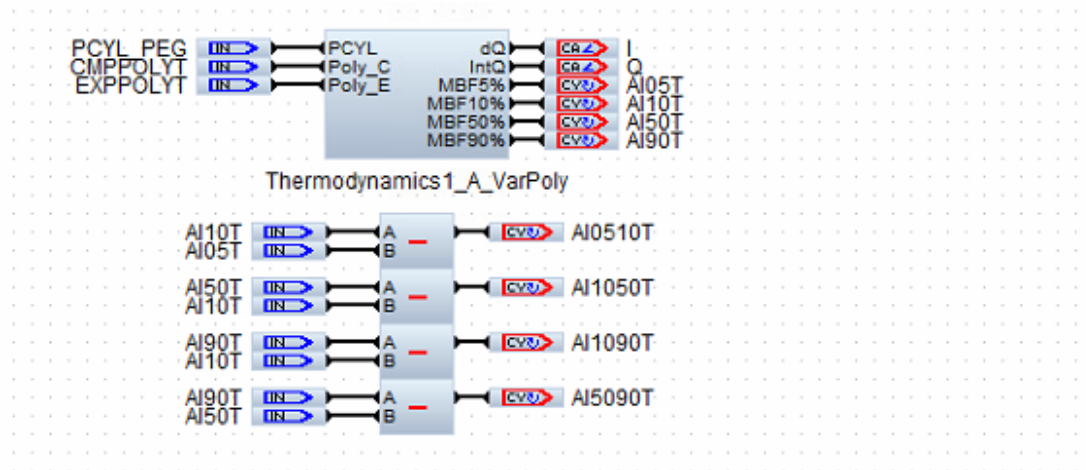


Figure 2.19 - IndiCom macros used for heat release analysis and calculation of mass fraction burned points and their corresponding durations

2.3.4. Fluids conditioning

Engine coolant conditioning system was designed in-house and it followed the schematic presented in Figure 2.20.

Coolant temperature was controlled through PID controller built-into the DSG DaTAQ Pro software and it could range between 10°C (temperature of cooling water supplied to the test cell by building services) and 105°C. It was running at 90°C for most of the tests, simulating fully warm engine condition. PID controls were calibrated at aforementioned 90°C and temperature was controlled to $\pm 1^\circ\text{C}$ of the target value.

Control of the heating was achieved through the use of 3kW immersion heater mounted in the 25 litre tank. Cooling was dealt with by Bowman heat exchanger. Flow through Bowman was controlled by the PID operated Kinetrol valve regulating flow of the plant water.

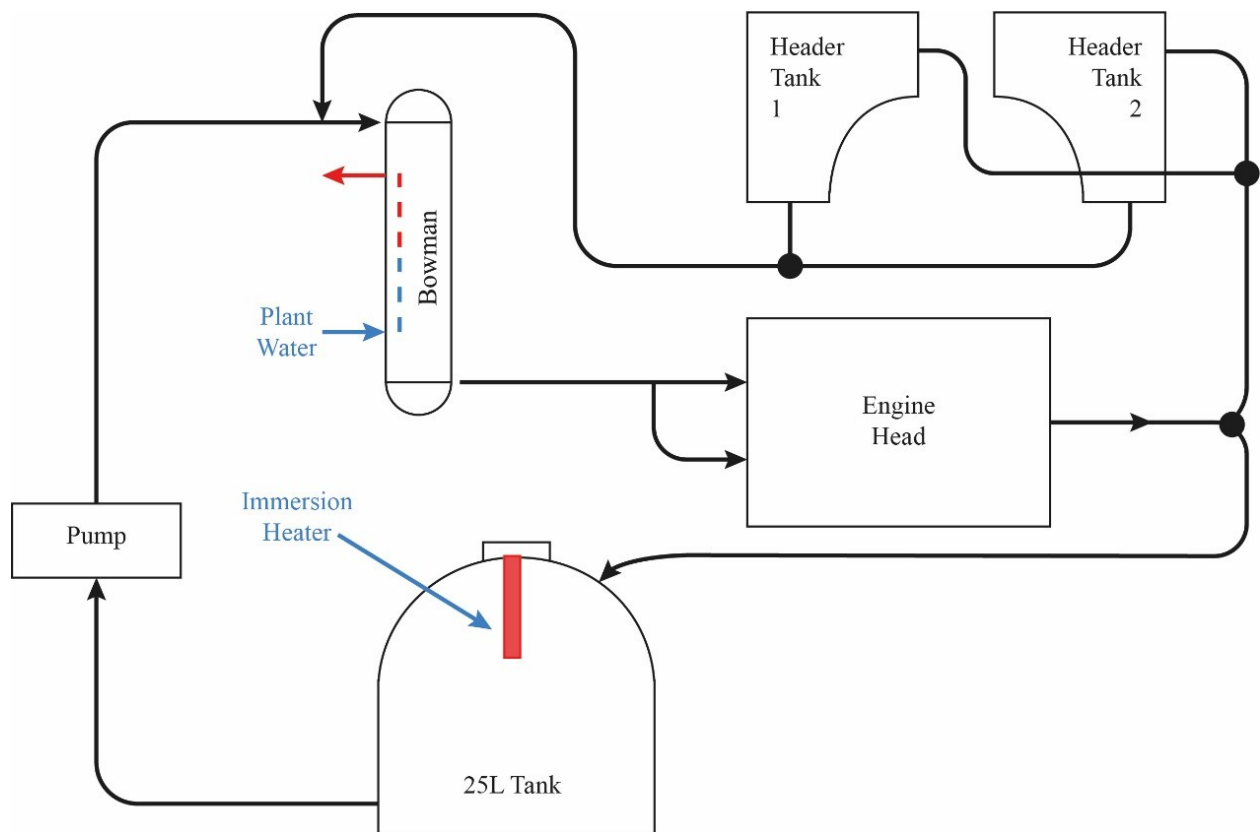


Figure 2.20 – Schematic of SCRE cooling system

The fuel conditioning and measurement system was based around a Siemens SITRANS M2100 Coriolis Flow Meter (1 7ME4100-1AD10-1AC8-Z). Coriolis instrument was selected due to its measurement principle and the fact that it compensates for the temperature differences (i.e. it is insensitive to any temperature drift over time), thus it was preferred over the popular gravimetric systems that are quite popular within the industry – likes of AVL 735S. It was also better suited to small flows required for operating single cylinder engine under low or medium speed/load conditions that were often lower than 1 kg/h. Accuracy of fuel measurement was set at 0.1% full scale mass flow (25 kg/h). It was installed upstream of the engine mounted high pressure fuel pump, and downstream of the fuel temperature conditioning loop. Conditioning circuit was comprised of the Fuelab low pressure pump (40401-1), Fuelab pressure regulator (52901-1), Bowman heat exchanger (15-9E) and Applied Thermal Control K3 chiller unit which supplied water/coolant solution to the heat exchanger. System was designed in-house and it proved to be effective in ensuring constant fuel temperature within

+/- 0.5 °C. Sample calculation of error analysis of GSFC deducted by flow measurements is shown in Appendices section.

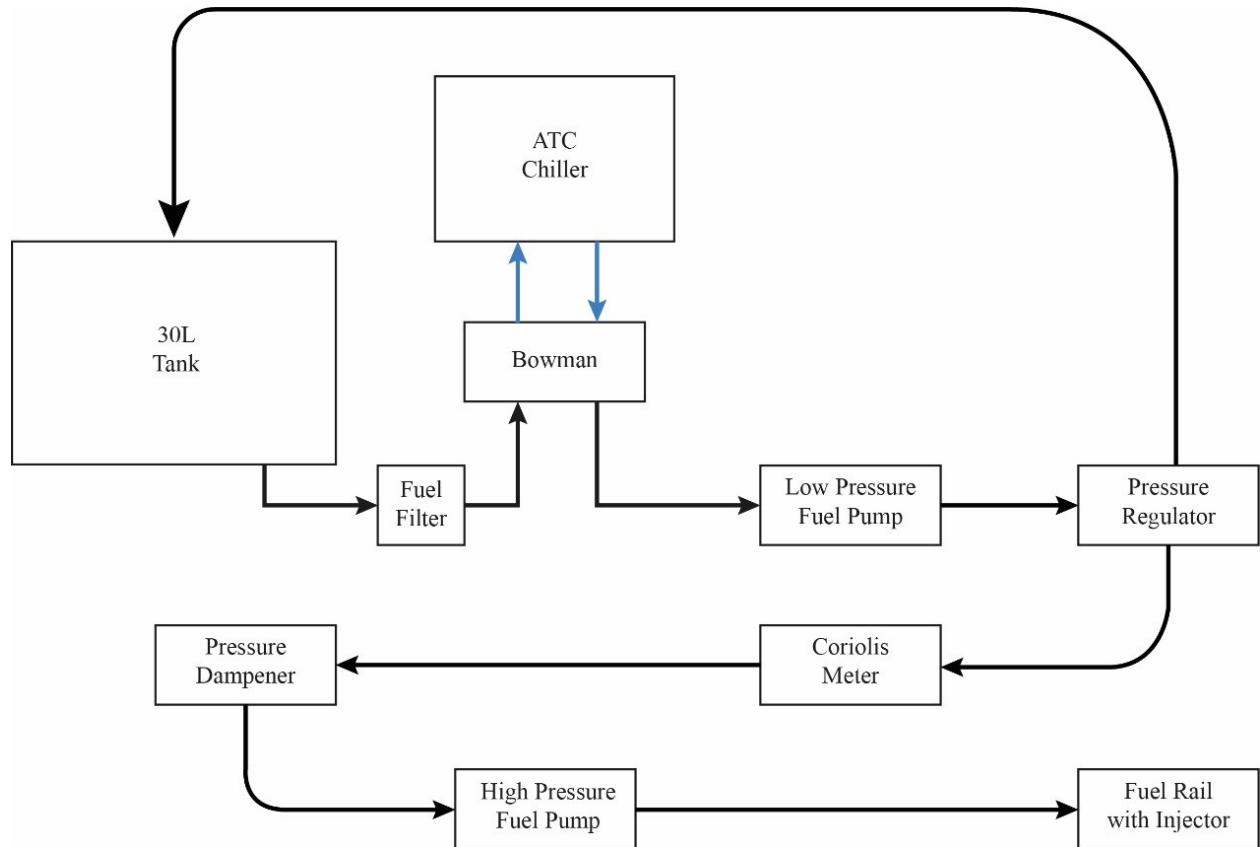


Figure 2.21 – Schematic of SCRE fuel conditioning circuit

Oil delivery circuit was powered by the Rocquet pump (1L22DE10R) coupled to the DC motor (Electro-Adda 1107-053626). Motor was interfaced with DaTAQ Pro control system via Siemens inverter. Pressure was set according to a set-point dialled in through DSG software. PID controller was reading oil gallery pressure and then adjusting speed of the pump to achieve given set-point. Oil gallery pressure was regulated to within 1 kPa, usually being set to 450 kPa absolute. Heating was achieved through two immersion heaters mounted inside the Ricardo Hydra bottom end, whilst cooling was based on the same method of controlling flow through Bowman described for coolant system. Similarly, accuracy was at +/- 1 °C at the nominal 90°C. Main oil system was also supplying engine head and Piston Cooling Jet (PCJ).

They had separate feeds equipped with needle valves for manual setting of desired pressure. System schematic is shown in Figure 2.22.

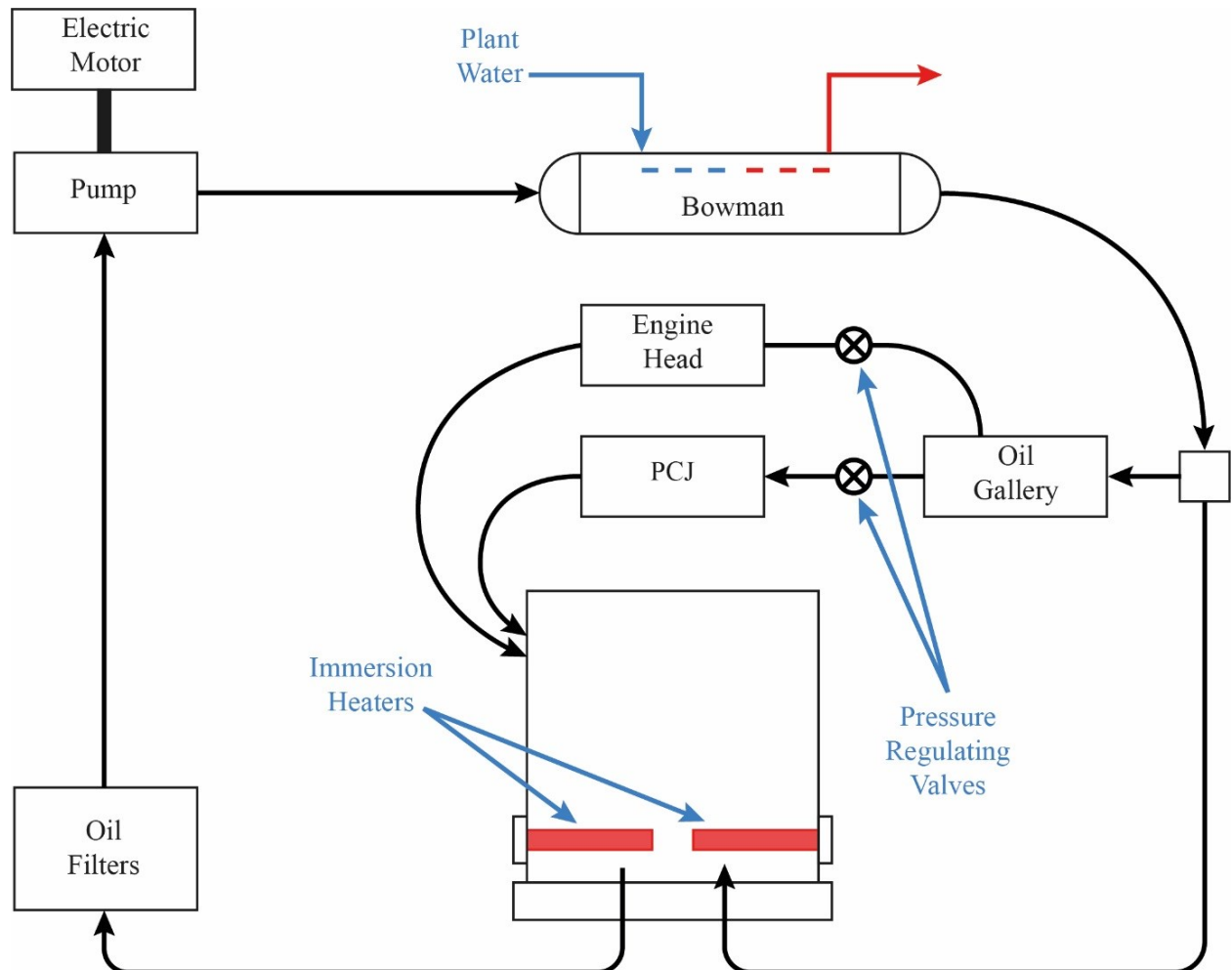


Figure 2.22 – Schematic of SCRE oil conditioning system

2.3.5. Air Path

Intake air path was designed so that engine could be run in either Naturally Aspirated (NA) or boosted mode. When running NA, air was being drawn through Meriam Laminar Flow Element (LFA, type 50MC2-2F) supplied by Labcell, then fed through Bosch Mass Air Flow (MAF) sensor which supplied feedback to engine Engine Control Unit (ECU) and Bowman heat exchanger (FG100-4075-2) for cooling if necessary.

When air needed to be heated, it was achieved through the use of two 375 W band heaters (RS 374-2965) mounted downstream of the Bowman heat exchanger.

Alternatively intake path could be set up as a boosted system in which upstream of the Bowman heat exchanger (and downstream of the MAF sensor) air was compressed with prototype small displacement Eaton V250 supercharger. Supercharger was driven by the 7.5 kW ABB DC motor (M3BP 132SMC-2 B3) coupled with ABB inverter drive operated through DaTAQ Pro control system. Air could be supplied at the maximum pressure ratio of 1.8:1.

Incoming air was then throttled (under NA operation) using the Bosch throttle (0 280 750 148) and passed through 29.5 litres intake plenum, which ensured that air flow was uniform and it took out any fluctuations in flow that might have been caused by the elements installed upstream (EGR feed and supercharger).

Exhaust system incorporated 29.5 litres plenum (same dimensions as on the intake side) and back-pressure valve supplied by Sierra CP Engineering. Its position could be controlled through DaTAQ Pro and adjusted on-the-fly if necessary. EGR take-off point was located downstream of the exhaust runner. EGR circuit consisted of the EGR cooler and EGR valve. Its feed back to the intake system was located downstream of the LFA meter and upstream of the SC which enabled running boosted regardless of the intake to exhaust pressure ratios.

2.3.6. Exhaust Emissions

Exhaust tailpipe emissions were measured using Horiba MEXA ONE Exhaust Gas Analysis System. It was configured to provide readings of the following species:

CO	Non-Dispersive Infra-Red (NDIR) Analyser – 50-5 000 ppm range
CO ₂	NDIR Analyser – 0.5–20% range
NO/NO _x	Chemilluminescent Analyser (CLA) – 10-10 000 ppm range
O ₂	Magnetopneumatic Analyser (MPA) – 1–25% range
THC	Heated Flame Ionisation Detector (FID) – 10–60 000 ppm range
CH ₄	Heated Flame Ionisation Detector (FID) – 50–25 000 ppm range

Analyser was equipped with two CO₂ NDIRs – one sampling CO₂ concentration in the exhaust and the other one measuring inlet air, therefore direct reading of EGR rate was being supplied to DaTAQ Pro system. DSG provided integration of the MEXA ONE analyser into the test bed system and all readings were read and logged into the output file through DaTAQ Pro.

Note that engine was not equipped with a catalytic converter, thus all exhaust species were measured as engine out emissions.

2.4. Engine Management System

Engine Control Unit (ECU) was provided by Mahle and it was an open loop prototype designed for JLR. Control over all engine operating parameters was executed by the use of ETAS INCA software. It was integrated into DSG DaTAQ Pro test bed control system and control over pre-defined parameters (i.e. ignition angle, cam phasing, fuel rail pressure etc.) could be executed through set-points within DaTAQ Pro.

2.5. Control systems developed for Imperial – JLR collaboration

Two main control systems were developed in collaboration with DSG engineers to simplify and automate running of the test sequences. It was found that during vast majority of the tests, engine was required to be running with a fixed 50% Mass Fraction Burned (MFB) set to a target value of 8°CA aTDC.

Closed loop “Combustion Control” code was implemented. It took readings of 50% MFB calculated from a heat release curve on the IndiCom, fed it to DaTAQ Pro system, which simultaneously read spark timing angle using the ECU controlling software INCA and adjusted it on-the-fly so that required 50% MFB was being achieved. Due to the fact that engine might be operating in the unstable region and combustion events are inherently subjected to some cycle-to-cycle variations, code used moving average window of 50 cycles. By doing so ignition angle was adjusted by small increments and any outlying cycles (eg. Partial burns, misfires or knocking cycles) were averaged with standard cycles, damping their effect. Code proved to be very successful and it significantly reduced test sequences run times, as it relieved engine tester from constantly adjusting ignition offset.

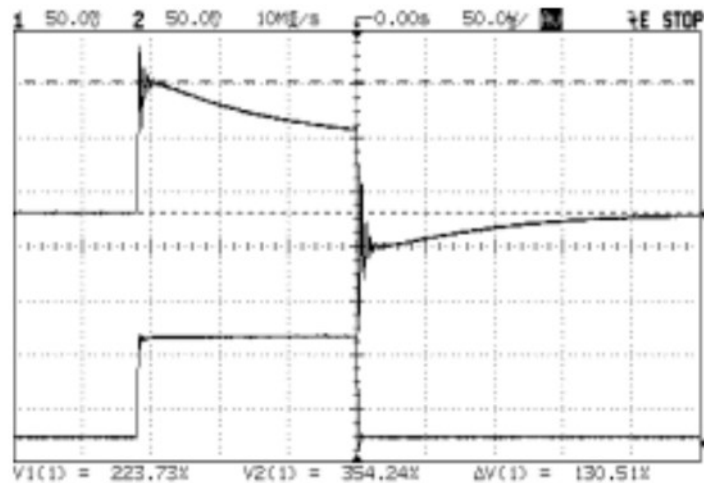
Other code developed specifically for the Imperial – JLR collaboration was the closed loop control of the EGR rate. In principle it worked similarly to “Combustion Control” code, but instead of looking up 50% MFB, it read EGR rate as supplied by MEXA, and then opened or closed EGR valve (controlled through INCA) to achieve set-point which was dialled in to the DaTAQ Pro system. Described control system was very useful when running EGR sweeps and let the engine tester to focus on adjusting other operating parameters (i.e. cam positions, injection timing, spark timing etc.) by maintaining constant EGR rate.

2.6. Spark Measurements and Analysis

One of the objectives set for the studies was to analyse how high energy ignition systems contribute towards extending the tolerable dilution limit. To do so, energy delivered to the spark had to be quantified.

The spark voltage was measured using the Tektronix Probe P6015A HV connected to a modified HT lead. It had capabilities of recording 40 kV peak values, typical rise time of only 4.0 ns and bandwidth of 75 MHz. Signal was attenuated by a factor of 10^3 .

Current was measured using Pearson 3972 current monitor. It was chosen based on its high frequency (20 MHz) and relatively low droop rate of 1.0 %/millisecond. Droop rate is the downward slope of the top of the output voltage pulse resulting from a flattop current input pulse. Because the current monitor cannot pass DC, whenever the output voltage is non-zero and the current is constant, as during the flat top of the pulse, the voltage decays toward zero exponentially. This is illustrated in the upper trace in Figure 2.23, whilst the lower trace represents the applied current pulse. Low droop rate is especially important when measuring elongated spark signals of modern, high energy systems. Sensitivity of the current probe was $0.1 \text{ V/A} \pm 1\%$.



amplification factor which could be manually set by the user. TL 072 Op Amp was used in non-inverting configuration. Its low-noise, high input impedance and low power consumption enabled running of the single 9V battery instead of dual power supply. Amplification used throughout the study was set to 45.

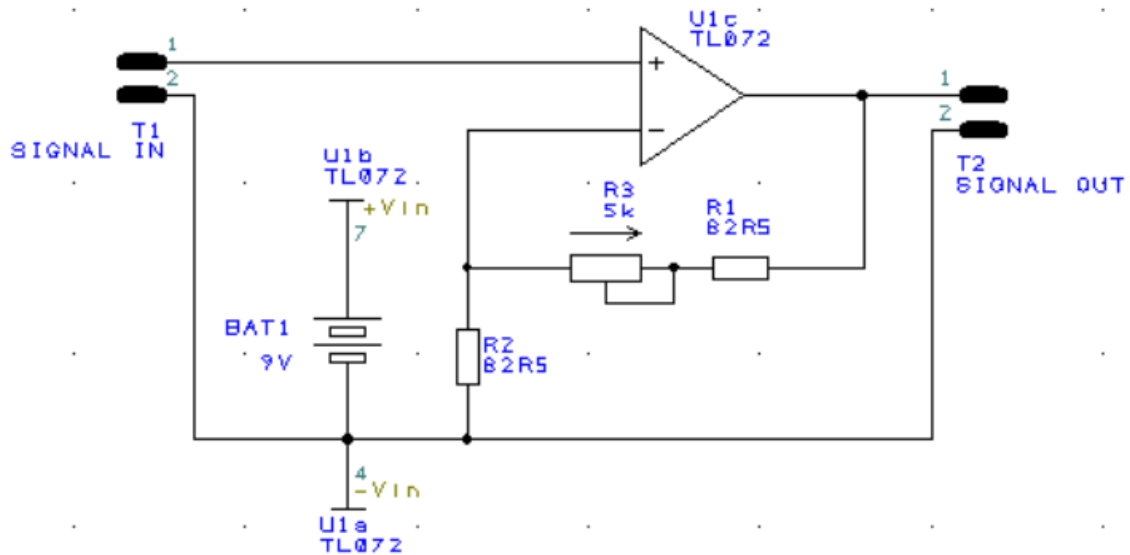


Figure 2.24 – Circuit schematic of low noise signal amplifier built for the secondary current data acquisition system

LabView was used to create data acquisition system based on the PXI-6115 module. Virtual Instrument (VI) was commissioned to acquire three signals in parallel – current, voltage and encoder pulses. Block diagram representing VI created for the purpose of this study is shown in Figure 2.25.

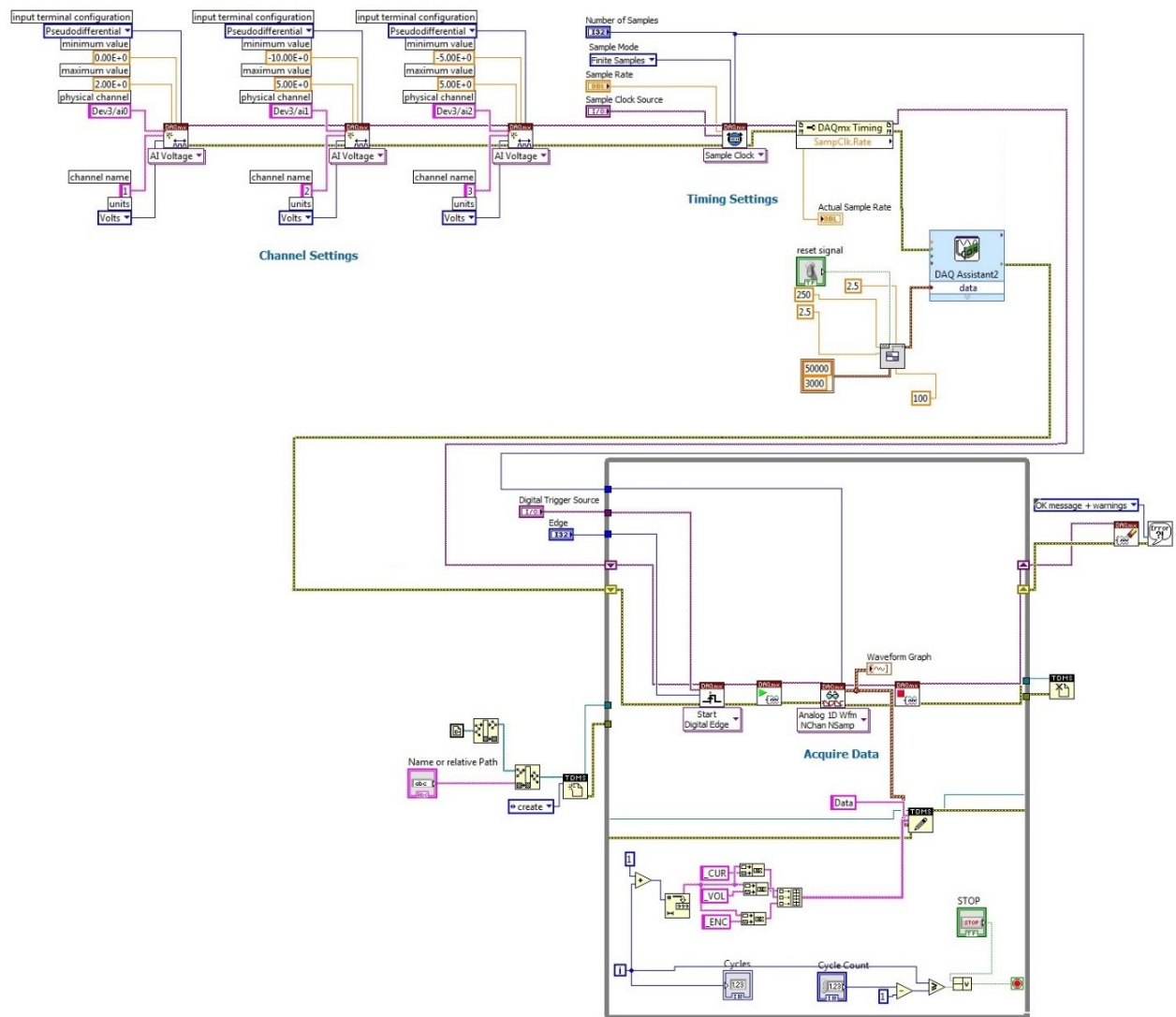


Figure 2.25 – Block diagram representing LabView VI built for secondary voltage and current data acquisition

Most challenging part of design was ensuring that data collected via LabView system could be easily compared with corresponding in-cylinder pressure data collected with IndiCom on the cycle-by-cycle basis. It enabled performing further analysis of any given cycle and comparing calculated heat release metrics with ignition energy and duration. To achieve that, data acquisition for two systems was initialised by a number of triggers. Idicom was put into “Event Measurement” mode, and LabView VI generated a TTL signal fed to IndiCom. It was a single signal initialising data acquisition. At the same time IndiCom software was configured to output a triggering signal at 1.5°CA prior to ignition for every engine cycle. Such function required setting in IndiCom initialisation file, hence software had to be restarted for every

single test point and accompanied with changes to its initialisation file. Such method significantly extended time needed for taking the measurement, however it was necessary to ensure efficient data collection with LabView and simplified data analysis and post-processing. By triggering LabView to acquire data at a specific point within the cycle, it only recorded values during a specified period (corresponding to ignition event), thus reducing size of the output files which played important role given that 3 channels were collected for 300 cycles at 10 MHz each. At 1500 RPM engine cycles takes 80 ms. Data collection for ignition signal was only needed for up to 6 ms (when measuring elongated, 5 ms spark durations), being only 7.5% of the cycle total time. Even after significant effort was made to reduce size of the files to minimum, a typical binary output file, TDMS format, had a size of 350 MB (short durations) to 425 MB (5 ms signal duration).

Once the signals were acquired and saved in the binary TDMS file, data processing was handled by script written in MATLAB. It proved to be very efficient in performing calculations, however loading data into the MATLAB environment was quite time-consuming, and was identified to be the major issue for data post-processing.

MATLAB script was written as a flexible code that could be used for analysis of LabView data acquired in a number of ways and represented flexibility built into the data acquisition VI. It had few variables which required initialising – frequency of data acquisition, gain setting for the current probe amplifier, number of cycles measured. After the code loaded data in the form of matrix, it applied amplifier gain settings to ensure that current data was expressed in mA. It then checked for any negative values of current or voltage and eliminated them from dataset. Based on the frequency of measurement and number of rows, the code generated time domain needed for energy calculation. Energy was then calculated using MATLAB trapezoidal integration function applied to the following equation:

$$E = \int V \cdot I dt \quad (2.3)$$

Signal duration was calculated using the current signal. Based on the time scale, the code generated number of partitions that had 0.1 ms size. Current signal was passed through low-pass Butterworth filter to eliminate any spikes that might produce false readings of active current signal. Filtered current data was “inserted into the partitions” by means of trailing average for previously specified partition sizes.

As a result, new variable representative of current trace, free from electrical noise and spikes, having number of datapoints matched to its duration in ms was created. This variable was then passed through a loop which assigned every point with the value of either 1 or 0 depending on the threshold value (set at 10 mA). As a result, a TTL-shape signal was created which could be quantified in two ways. First one was summing all the 1s and dividing by 10 to get duration of the signal in ms, whilst second method was implemented by running two loops on the filtered signal. First loop found location of the first value over the aforementioned threshold, whilst second loop found location of the last value over the threshold. Subtracting the two and dividing result by 10 gave duration of the signal in ms. Two methods were equivalent for most of the signals, however for BW running in Multistrike mode two durations were significantly different. First represented only the duration when the current was active (extinguished spark between multiple events had no current flow) and second method output overall signal duration irrespective of what was happening to current trace in-between. Data processing method is illustrated in Figure 2.26 using the example of BW Multistrike operation.

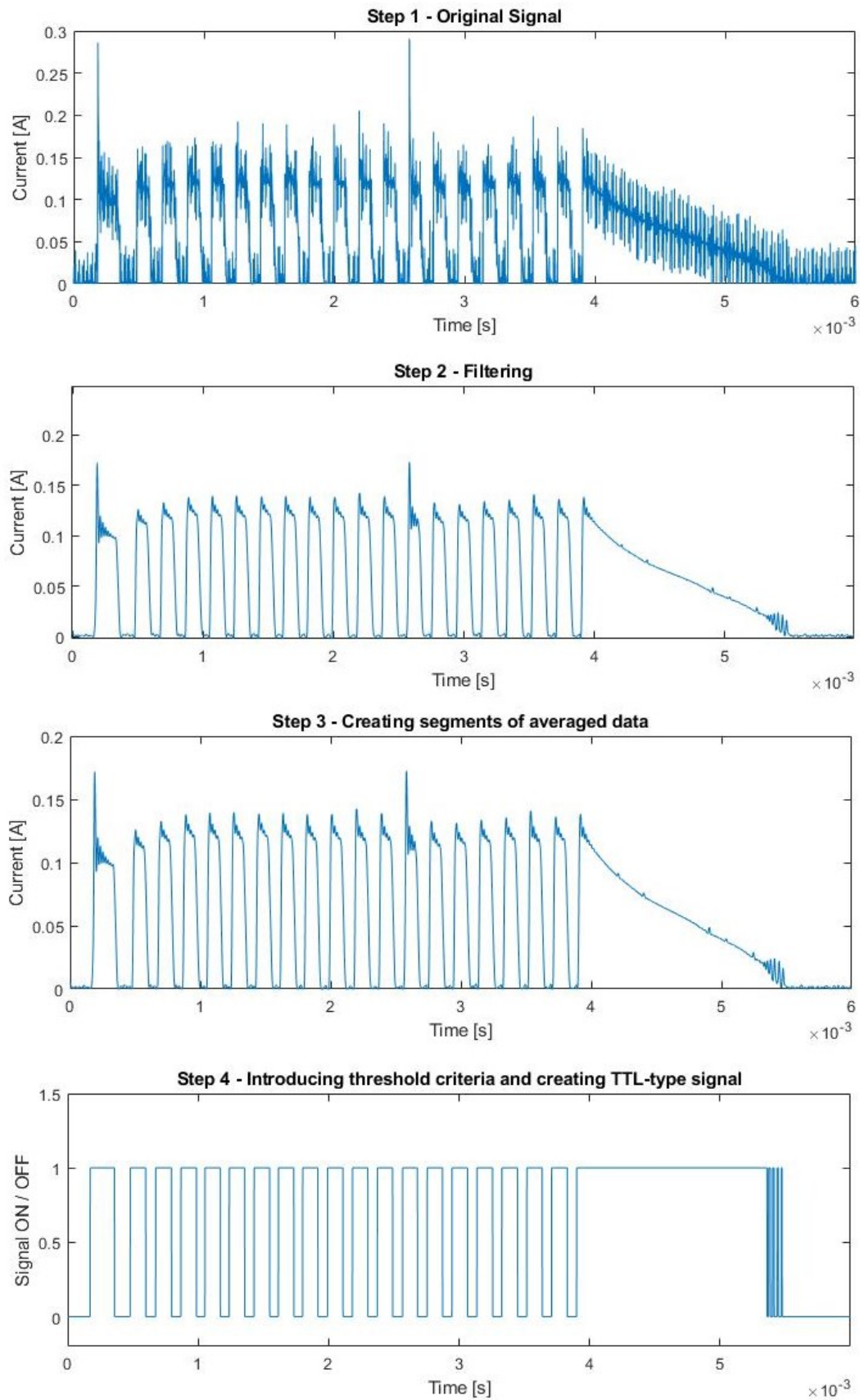


Figure 2.26 - Representation of MATLAB script calculating signal duration for BW Multispark operation

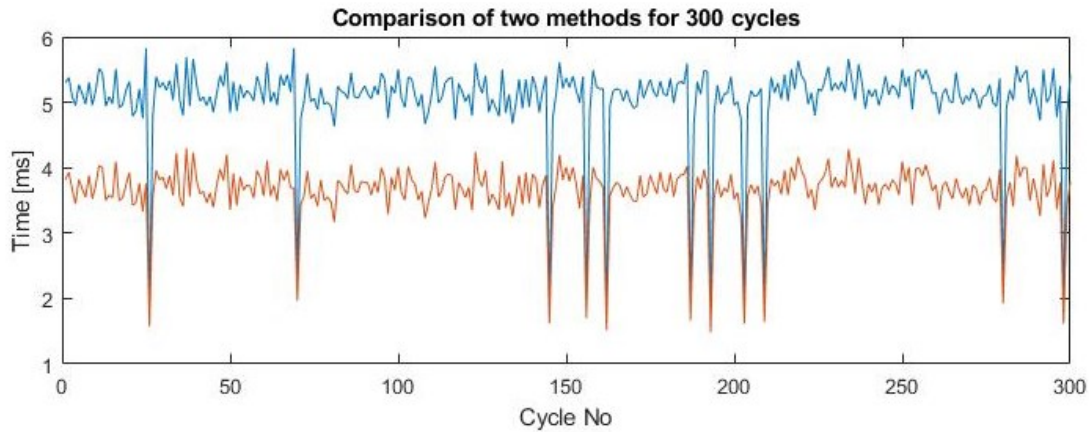


Figure 2.27 – Representation of cycle durations (300 cycles) calculated by two methods for BW Multispark operation.

2.7. Preliminary Studies

2.7.1. Matching to Multi-Cylinder Engine

After successfully commissioning Build No 2 of the engine which was based around installation of CVVL on the intake side (as per Table 2.1), focus was put on ensuring that data collected using SCRE was closely matched to its multi-cylinder counterpart AJ-20 engine. Being able to replicate conditions present in multi-cylinder engine was important to enable feeding high quality data into engine development programmes and making sure that experiments performed in this study result in real-world optimisation of multi-cylinder engines.

To replicate multi-cylinder engine conditions using SCRE, comparison of the gas dynamics was needed. Firstly, engine was baselined “as is” without any changes to its breathing system and it was found that the two were not well matched when trying to replicate running conditions of one on the other. Pumping loop and exhaust pressure traces were identified as the main problem areas. Knowing that multi-cylinder engine was equipped with turbocharger which significantly alters exhaust pressure wave action, it was proposed to introduce orifice plates into exhaust system that could somehow replicate it. Six orifice plates were manufactured, and

they effectively reduced exhaust area from 100% (40mm diameter) to 25% area (20mm diameter), 10% area (12.6 mm diameter), 5% area (9mm diameter), 3% area (7mm diameter) and 1.5% area (5 mm diameter), as shown in Figure 2.28. Orifice plates were installed 20 cm downstream of the exhaust port.



Figure 2.28 – Orifice Plates used for matching exercise; openings ranging from 1.5% to 100% of the exhaust pipe area

Study was designed at 1500 RPM, testing a range of plenum pressures targeting BMEP of 2.52 bar and 7.00 bar which were the highest residency points used in multi-cylinder engine testing.

It was found that introducing orifice plates significantly altered pumping loops (presented in Figure 2.29 for 1500 RPM, 3.0 bar NMEP, 3.6 bar GMEP) and back-pressure in the exhaust (as expected). Based on the findings of this study, it was concluded that best matching was achieved with the use of 20 mm orifice plate (reducing area to 25%).

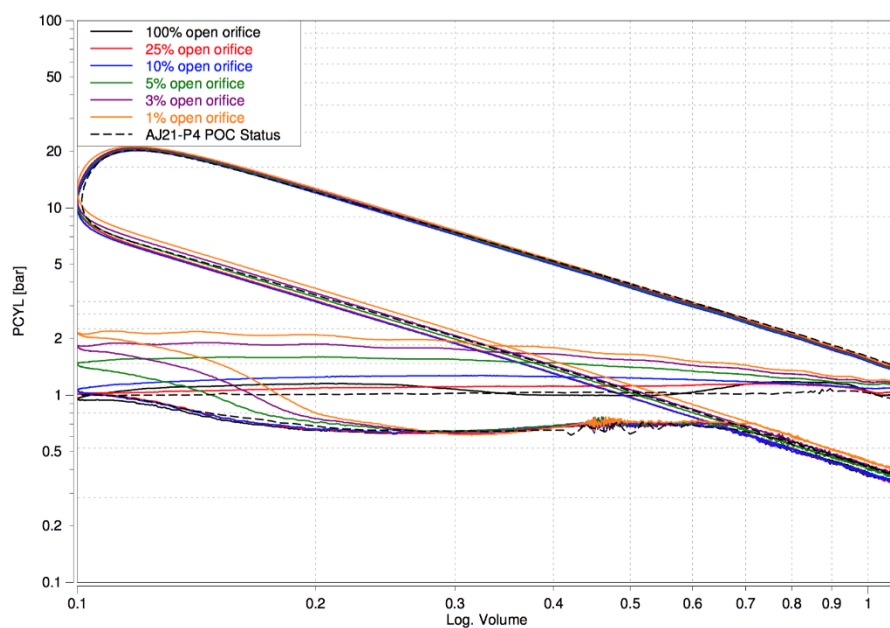


Figure 2.29 – Comparison of $\log P$ vs $\log V$ plots for various orifice plates at 1500 RPM, 3.0 bar NMEP (3.6 bar GMEP)

Initial study was followed by validation exercise during which a wide range of operating parameters was changed (EGR rate, cam timing, injection timing, ignition advance) to ensure consistent duplication of results, however it was found that pumping loops cannot be replicated across all conditions (plots are included in the Appendices section for completion).

It was concluded that due to differences in pumping loops between SCRE and its multi-cylinder counterpart, all subsequent studies should target Gross Mean Effective Pressure (GMEP) instead of BMEP or NMEP. GMEP represents only compression and expansion parts of the 4 stroke engine cycle without taking Pumping Mean Effective Pressure (PMEP) into account. By doing so, more direct comparisons can be made between single and multi-cylinder engines. Validation exercise confirmed that this approach was best and most practical solution that would not require physical changes to exhaust system for every tested condition. Validation plots can be found in Appendices. Despite of the fact that GMEP was chosen for targeting engine load, engine stability metric used throughout this thesis is Coefficient of Variance (CoV) of NMEP. It was done to ensure consistency with JLR's metric derived from Noise, Vibration and Harshness (NVH). CoV of NMEP and CoV of GMEP are qualitatively the same, however they differ by about 16% for engine conditions presented in all the subsequent studies.

2.7.2. Start of Injection Sweeps

Fuel injection timing plays a very important role in gasoline engine operation. It directly affects cycle efficiency and exhaust species formation. Injecting too early or too late within the cycle might lead to piston wetting or ineffective preparation of air-fuel mixture prior to combustion. To establish the optimum strategy, given that engine was limited to single injection per cycle, a series of sweeps across the highest residency points were performed with primary focus on optimising the efficiency, whilst keeping the pollutants emission to minimum.

Start of Injection (SOI) was swept across a wide range of 340 °CA bTDC_f to 160°CA bTDC_f for three operating conditions:

- 1500 RPM; ~ 3.6 bar GMEP (const. intake plenum pressure of 54 kPa); 0-0 valve overlap
- 1500 RPM; ~ 3.6 bar GMEP (const. intake plenum pressure of 48 kPa); 25-25 valve overlap

- 2000 RPM, ~ 10.0 bar GMEP (const. intake plenum pressure of 122 kPa); 0-0 valve overlap

All the above were run at stoichiometric ($\lambda=1.0$) condition with oil and coolant temperatures of 90°C and fuel injection pressure of 200 bar.

The first two points were chosen to represent low speed and load condition, highlighting any potential differences that might result from introducing valve overlap. Third point was introduced to analyse effect of SOI under medium speed, boosted condition.

Results of the study are presented in Figure A.3 and A.4 attached in the Appendices section, where first one is focused on combustion metrics, whilst the second one addresses emissions formation. Based on the obtained results, it was concluded that the most beneficial region exists in the 310 to 300 °CA bTDC_f region (highlighted in both plots). Having no means of measuring particulate matter (PM), unburned hydrocarbons (THC) were selected as the main indicator of potential undesirable SOI effects. Heywood (1988) linked increased THC formation with potential fuel impingement on the piston surface which was one of the main reasons why this study was launched. It can be seen that advancing injection past 310°C**A** bTDC_f THC production rose sharply for both 1500 RPM conditions. It was attributed to injecting too close to TDC_{ge} which resulted in significant piston wetting. On the other hand, delaying injection and injecting later than about 270°C**A** bTDC_f exhibited sharp rise in CO and drop of CO₂ for 2000 RPM scenario. Such behaviour was linked to delayed injection having negative effect on homogeneity of mixture formation, hence not allowing effective combustion. It was also manifested in increased fuel consumption (measured as NSFC) and lower combustion stability (measured as CoV of NMEP) shown in Figure A.3.

It was concluded that 300°C**A** bTDC_f should be used for any studies performed on SCRE representing AJ-20 combustion system. It offered minimised THC formation, whilst combustion metrics showed high combustion stability and close to optimum fuel consumption across all tested points.

2.8. Test Points

Test points used for any work carried out using the SCRE were picked after consultations with project sponsoring company – Jaguar Land Rover. Their research department identified main points of interest, called “high residence points” for fuel economy and emissions certifications.

They were most useful from the perspective of feeding research findings into their multi-cylinder testing further down the engine development process. It was of primary importance that any findings presented in this thesis could be leveraged in the development process for next generation of gasoline engine. At the same time, points were chosen to represent wide range of engine operation, ensuring that objectives set for this work could be completed.

Table 2.6 represents three points which were chosen for main work packages performed in this work.

Table 2.6 – Engine operating conditions used in the studies

Engine Speed	RPM	1500	1500	2000
Engine Load (GMEP)	bar	3.6	7.9	9.8
Lambda (λ)	-	1.0 – 1.8		
EGR	%	0 – 20		
Intake phasing	°CA	Parked at 0 °CA		
Exhaust phasing	°CA	Parked at 0 °CA		
IVC	°CA bTDC _f	90		
MFB50	°CA aTDC _f	8		
Intake air temperature (measured at intake plenum)	°C	45		
Oil temperature (gallery)	°C	90		
Coolant temperature (engine-out)	°C	90		
Start of Fuel Injection	°CA bTDC _f	300		
Fuel Pressure	bar	200		
Fuel injector	-	Central Bosch DI (EB2)		
Fuel	-	95 RON Euro 6 (E10)		

2.9. Closing Remarks

Successful commissioning of the SCRE test facility was completed at Imperial College in collaboration with JLR who sponsored the project. The engine was equipped with research grade instrumentation and design of ancillary systems enabled easy installations of new prototype systems for proof of concept testing.

Given the objectives of this thesis, the focus was put not only on flexibility but also on ensuring that engine could be used purposely on the studies built around charge dilution. Industry standard methods used for indicated measurements were reviewed and modified accordingly. As a result, in-cylinder pressure trace pegging methods were introduced for various operation modes of advanced valvetrain system, together with modified heat release algorithms that used variable polytropic coefficients calculated for each cycle.

Furthermore, a new very high sampling rate (10 MHz) data acquisition system was designed and delivered specifically for measurements of secondary current and voltage spark traces. It was accompanied with data post-processing tools written in MATLAB.

Data acquisition was automated to high degree and robust design of all the sub-systems enabled running SCRE in the configuration described above for up to 10 hours a day which is an industry standard for thermal engines.

Chapter 3

Valve Deactivation Strategy

3.1. Preliminary Remarks

Given the previously discussed ambiguity as to how valve deactivation affects in-cylinder temperature and associated heat transfer effects, comprehensive study addressing this issue was launched. Its main motivation was to decouple the benefits and shortfalls of running engine in the single valve lift mode and provide fundamental understanding of the in-cylinder flow motion and emissions formation with the use of diluted mixtures (excess air and EGR). Lack of clear understanding as to at which engine loads, the effects of valve deactivation stop being beneficial and what causes the preference of using the strategy at lower engine loads influenced selection of test conditions. The study was performed across three engine running conditions outlined in Chapter 2, page 103:

- 3.6 bar GMEP at 1500 RPM
- 7.9 bar GMEP at 1500 RPM
- 9.8 bar GMEP at 2000 RPM

Two points being run at 1500 RPM were chosen to highlight any potential changes that are load dependent, whilst 2000 RPM highest load point served as an indicator of running higher

speed, heavily boosted point under diluted conditions. All three conditions were high residency points used in engine certification process, hence the study and its results were highly relevant in aiding JLR's engine development process.

Experimental work was carried out using the set-up described in detail in Chapter 2, whilst significant part of the study was focused on the complementary numerical simulations using the commercial STAR-CD/es-ice code. As part of JLR's Centre of Excellence for Spark-Ignition Engines at Imperial College, simulations using experimental data inputs as boundary conditions were set-up and run by Alexandros Kountouriotis. Exact details on models validation and set-up of the CFD simulations can be found in his PhD thesis, Kountouriotis (2019).

Computational grid consisted of 2.2 million cells at BDC including the intake and exhaust ports with a cell size of 0.4-0.8 mm. Base timestep used in the simulations was set to 0.05°CA , and it was reduced by half during the opening and closing of the valves. Sensitivity studies concluded that k-epsilon RNG model was the most appropriate for turbulence modelling, whilst heat transfer was estimated by Angelberger model for the piston, engine walls and cylinder head. Walls of the intake and exhaust ports were modelled as adiabatic entities. Simulations were run to provide insight into the gas exchange, in-cylinder flow activity and in-cylinder residual concentration and distribution.

3.2. Results

3.2.1. 1500 RPM, 3.6 bar GMEP

The first thing that should be noted when comparing Dual Valve Actuation (DVA), standard operating scenario with both intake valves opening simultaneously, and Single Valve Actuation (SVA), running with one of the intake valves completely shut, is the fact that plots representing SVA contain more test points and form bigger area when plotted against EGR Rate on the x-axis and Lambda on the y-axis. It is caused by the increased tolerance to dilution exhibited by the engine running with one of the intake valves being deactivated. Coefficient of Variance (CoV) of Net Mean Effective Pressure (NMEP) over 300 cycles was chosen as the combustion stability metric and can be seen in Figure 3.1 below. Thick black line was added to this plot and it represents 3% CoV of GMEP which was imposed as Noise, Vibration and Harshness (NVH) stability limit being used by JLR in engine development. It is clearly demonstrated that the dilution tolerance which satisfies the aforementioned limit was increased from lambda 1.15 to 1.50 and 1.00 to 1.16 for EGR rates of 0% and 17.7% respectively (where 17.7% EGR Rate was the maximum EGR dilution achievable for both DVA and SVA cases within the stability limit).

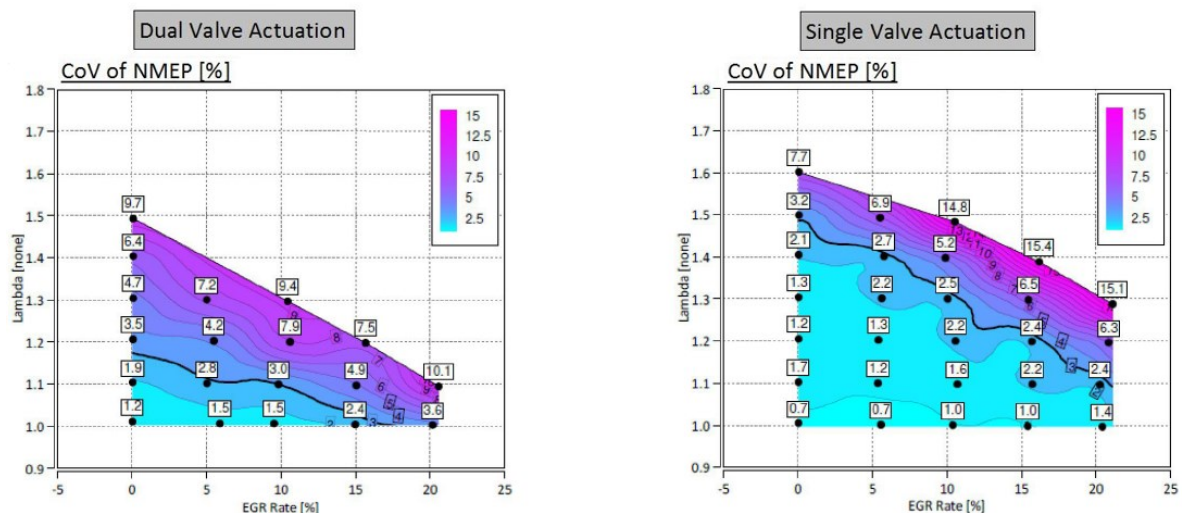


Figure 3.1 - Combustion stability comparison of DVA vs SVA at 1500 RPM, 3.6 bar GMEP

Other significant factor that differentiates DVA from SVA is noticeably lower plenum pressure required for SVA to achieve the same GMEP at any given dilution point (3.5% to 7.0 % difference across the entire map).

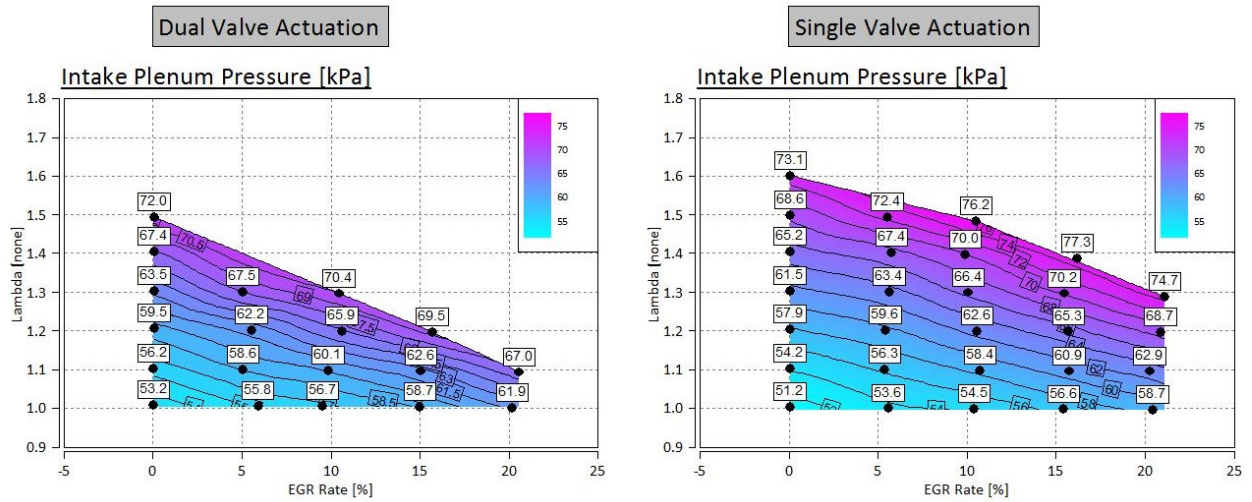


Figure 3.2 – Intake Plenum Pressure comparison of DVA vs SVA at 1500 RPM, 3.6 bar GMEP

Given that both DVA and SVA cases were run targeting 50% Mass Fraction Burned (MFB) at $8^{\circ}\text{CA}_{\text{aTDC}_f}$, it was important to analyse ignition advance and combustion durations. It was shown that ignition angles of DVA were significantly advanced compared to SVA (Figure 3.3) which indicates slower burn duration for equivalent dilution levels. When comparing the ignition to 10% MFB with the point within the cycle where the ignition took place, it can be seen that the majority of the difference was coming from this initial stage of combustion. It is believed that such behaviour is due to initial flame transportation aided by the swirl motion. Interestingly, once the comparison is being made between points that have different dilution levels but fall on the same 3% CoV of NMEP stability limit line, the difference between DVA and SVA becomes negligible (less than 1°CA , shown in Figure 3.5). Most of the points adhering to 3% CoV of NMEP limit, regardless of the valve operation mode, have ignition to 10% MFB duration within the $25\text{-}28^{\circ}\text{CA}$ window.

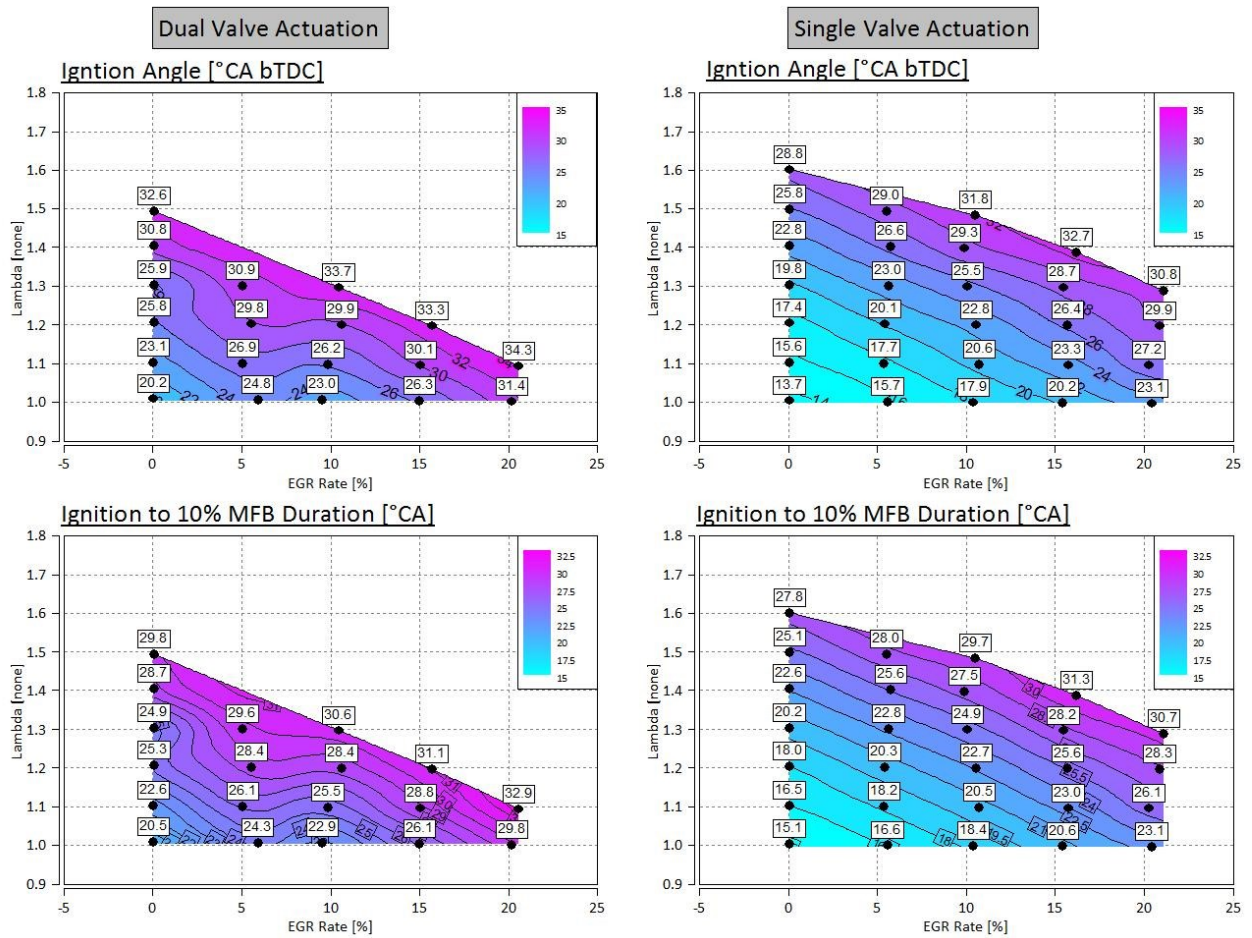


Figure 3.3 – Ignition advance and initial burn duration comparison of DVA vs SVA at 1500 RP, 3.6 bar GMEP

Once 10 to 90% MFB is plotted (Figure 3.4), it can be seen that, as stated previously based on the observation of ignition to 10% MFB, the difference in later stages of the combustion process between DVA and SVA is much less pronounced for the same dilution levels, and again negligible if one compares DVA and SVA based on the maximum of 3% CoV of NMEP criteria (Figure 3.5).

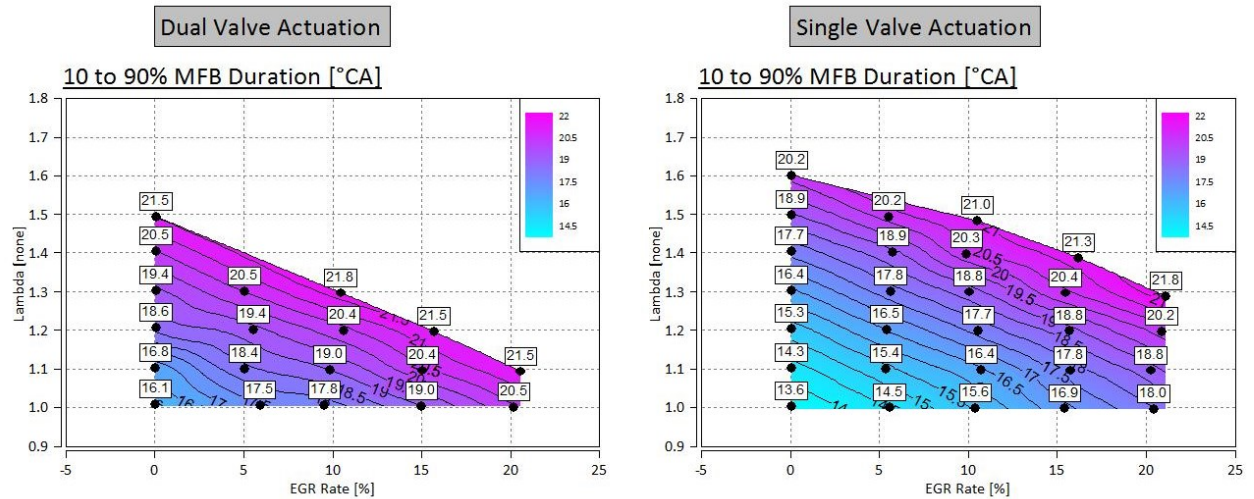


Figure 3.4 – 10 to 90 % MFB duration comparison of DVA vs SVA at 1500 RPM, 3.6 bar GMEP

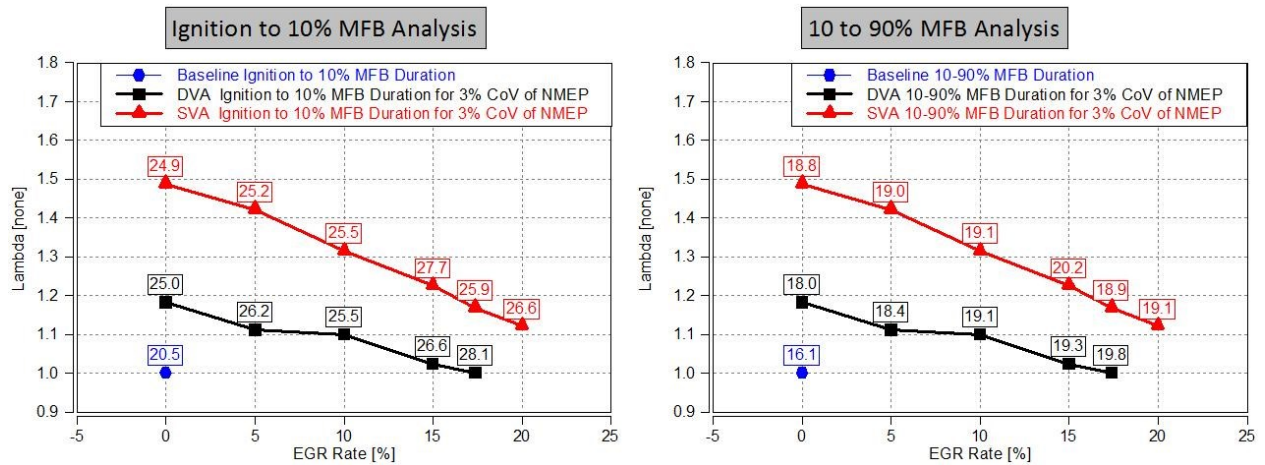


Figure 3.5 – DVA vs SVA burn duration analysis at 1500 RPM, 3.6 bar GMEP

Fuel efficiency measured as Gross Specific Fuel Consumption (GSFC) was almost identical for DVA and SVA at stoichiometric condition (241.7 g/kWhr vs 241.1 g/kWhr) and definitely within the measurement error tolerance (<0.25% difference). GSFC benefit of running SVA over DVA can be noted once dilution (both running lean and introducing EGR or combination of both) was introduced. In order to better illustrate fuel saving that could be achieved whilst adopting single valve operation, Figure 3.7 showing GSFC reduction (restricted to 3% CoV of NMEP) normalised to 1.0 at stoichiometric conditions was produced. It is evident that maximum reduction of about 5% can be achieved under DVA conditions for 0 to 10% EGR and about 3% for EGR rates of 15%, whilst SVA consistently outperformed DVA and reached

maximum of 7.5% at 0% EGR, and 6-7% for higher EGR rates up to 20% when compared against baseline stoichiometric condition. Points which achieved highest reduction for each strategy were highlighted with green in Figure 3.7.

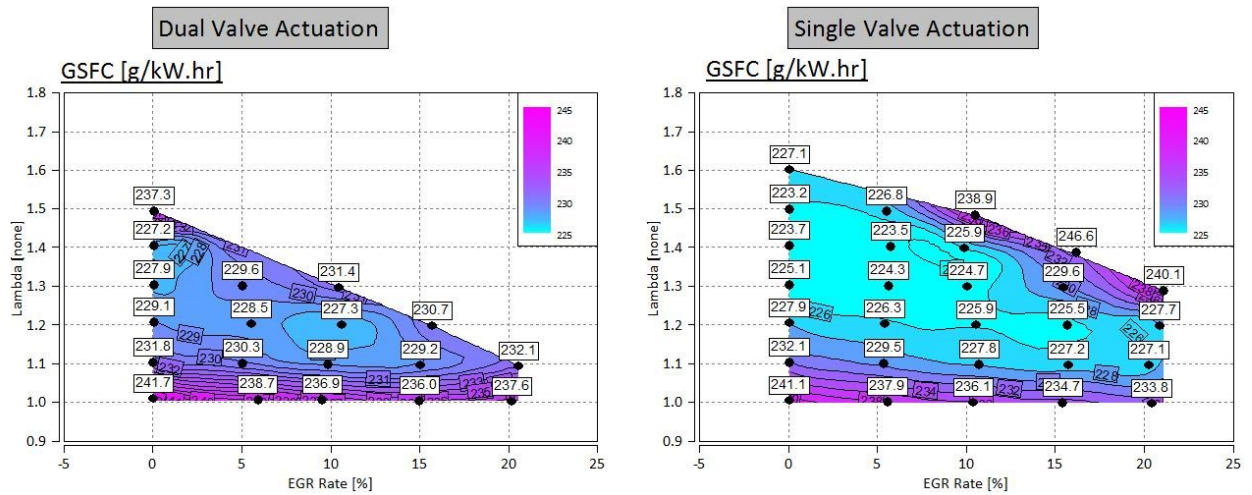


Figure 3.6 – GSFC comparison of DVA vs SVA at 1500 RPM; 3.6 bar GMEP

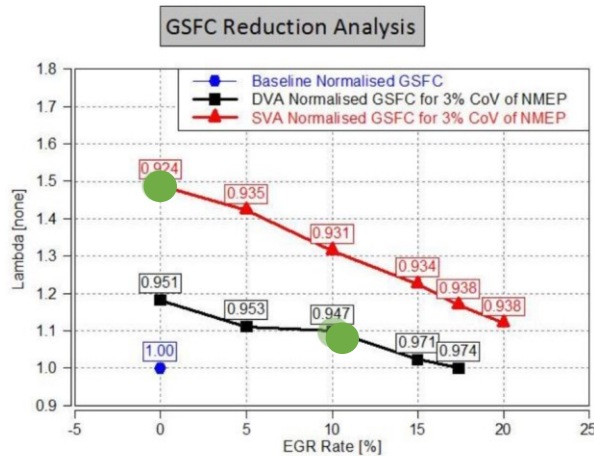


Figure 3.7 – GSFC reduction analysis for a fixed combustion stability limit of 3% CoV of NMEP at 1500 RPM, 3.6 bar GMEP

Experimental findings outlined above required Computational Fluid Dynamics (CFD) simulations to be run in order to generate insights into the in-cylinder flow patterns, heat transfer phenomena and to estimate in-cylinder trapped residuals.

Given the experimental results proved significantly higher dilution levels can be tolerated when adopting SVA strategy, in-cylinder flow field characteristics were studied in detail. First set of simulations was produced for “baseline” lambda 1.0, motored condition without introducing any dilution, focusing on the part of the cycle between Intake Valve Opening (IVO), which occurs at gas-exchange TDC (TDC_{ge}) and firing TDC (TDC_f), shown in Figure 3.8.

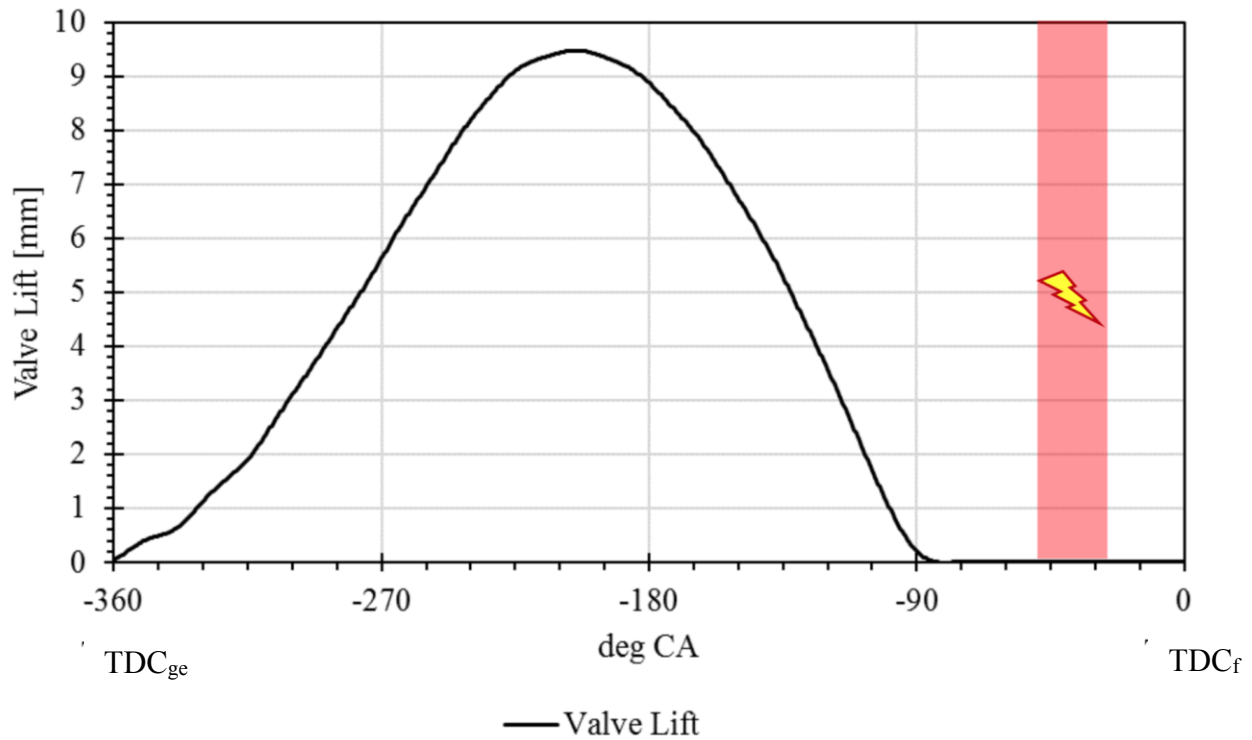


Figure 3.8 – Representation of the valve lift duration and ignition region during intake and compression strokes used in numerical simulations

It was found that increased SVA activity can be attributed to the averaged in-cylinder velocity, tumble, swirl and turbulent kinetic energy (TKE) values.

Averaged in-cylinder velocity was presented in Figure 3.9. It can be seen that over significant part of the cycle (between TDC_{ge} and TDC_f) SVA achieves much higher velocity, and from 270° CA b TDC_f (mid-stroke) onwards, it was consistently more than twice as high as the corresponding DVA condition (represented by SVA/DVA dashed line in Figure 3.9). Such significant difference was attributed to the restriction in the intake throat area of the SVA which resulted in higher valve jet velocity. Maximum velocities were also reached at different points within the cycle. DVA achieved maximum of 51 m/s at 45° CA a TDC_{ge} , whilst SVA reached

68 m/s 40°CA later, roughly mid-stroke in the intake stroke. Past their maximum velocity values both DVA and SVA averaged velocity profiles decreased in similar manner, keeping aforementioned constant ratio of 2.0-2.5 in favour of SVA until Intake Valve Closure (IVC) which occurs at 90° CA bTDC_f. Ignition points for both DVA and SVA at $\lambda=1.0$ were marked, and their corresponding values for DVA and SVA were 5.7 m/s and 12.8 m/s respectively, showing that at the point of ignition SVA attained more than double the averaged velocity of the DVA, even though it occurred closer to TDC. At the TDC_f, about 90° CA after IVC, ratio of SVA/DVA reached its maximum of 3.3:1.

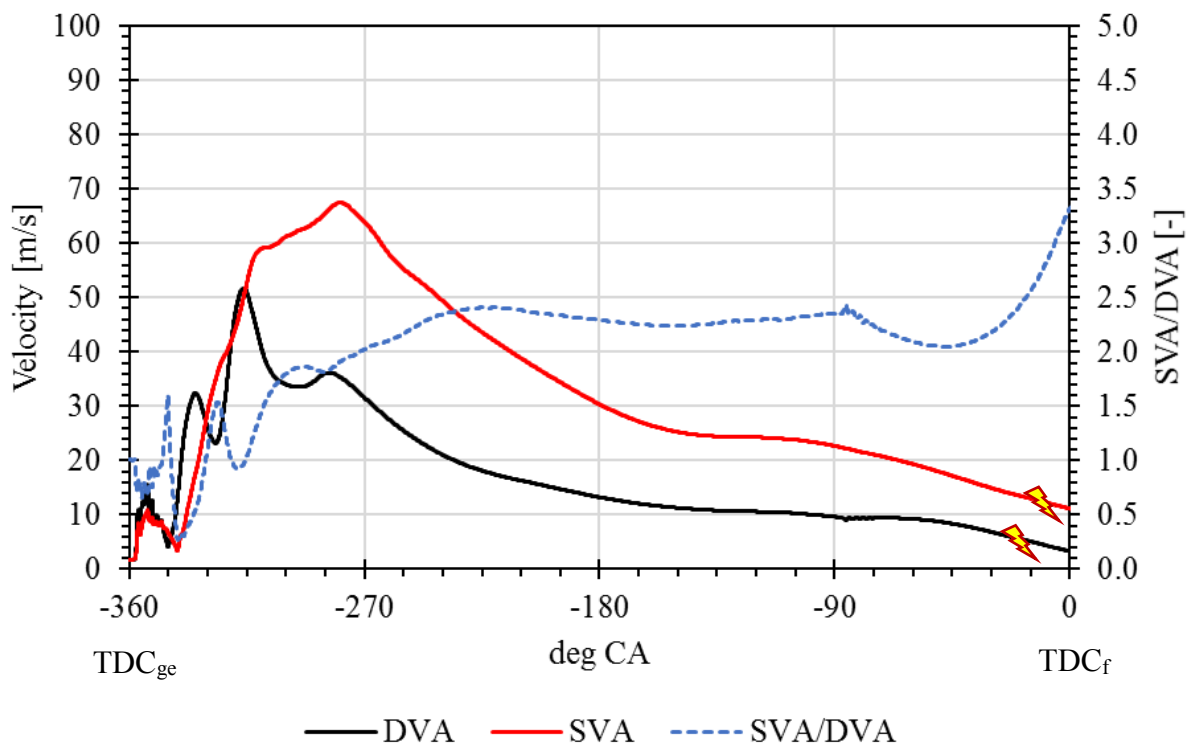


Figure 3.9 – Averaged in-cylinder velocity for DVA and SVA at 1500 RPM, 3.6 bar GMEP

JLR's AJ-20 intake ports were designed to provide high tumble motion within the cylinder for standard dual valve operation. Seeing how significant impact single valve deactivation had on averaged velocities, both tumble and swirl motions were examined for DVA and SVA.

Figure 3.10 shows that symmetric intake ports under DVA conditions did not produce any swirl motion (as expected), however tumble motion was developed, and it reached its maximum of 29 around mid-stroke during intake part of the cycle. After reaching its maximum, it consistently reduced to 1 at TDC_f. It is believed that such behaviour was caused by the piston effectively “squeezing” tumble motion. SVA operation produced significantly different tumble

profile which was also not the primary motion (as opposed to DVA). It is evident that swirl became the “governing” motion for SVA and it reached maximum of 50 which was almost double of the maximum value of 27 achieved for the tumble. Swirl profile of SVA resembled tumble profile shape of DVA as it also reached its maximum value mid-stroke and then consistently reduced, however it maintained a high level of 10 at TDC_f (10 times higher than the corresponding value of tumble for DVA). It is believed that swirl motion could sustain higher levels because the piston motion did not act in the same plane, and therefore had reduced impact on its magnitude during the compression stroke (less effective “squeezing” effect). The tumble motion observed for SVA reached similar level to DVA at its maximum (27 vs 29), however it occurred about 40° CA later within the cycle and was followed by reverse-tumble motion i.e. tumble motion crossed the 0 plane and started rotating in the opposite direction just after passing the BDC. As proposed by Kountouriotis (2019), it was assumed that two vortices (swirl and tumble) were two independent rigid motions, where the stronger swirl was transporting the tumble and as a result, rotated it during the cycle.

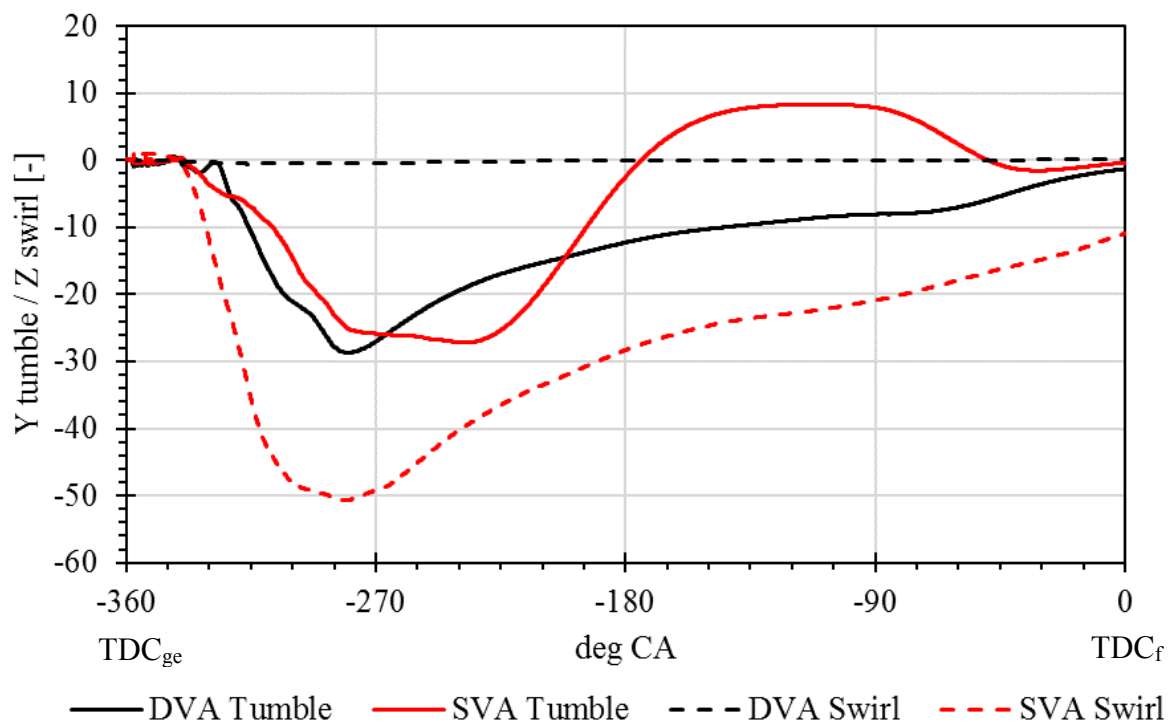


Figure 3.10 – Tumble and swirl motion representation for DVA and SVA at 1500 RPM, 3.6 bar GMEP

Knowing the implications of introducing valve deactivation on in-cylinder velocities, TKE between TDC_{ge} and TDC_f was also examined. As outlined by Scarcelli et al. (2013) increase

in turbulent kinetic energy achieved through implementing design modifications to combustion chamber can be effective in enhancing flame propagation in the early stage, hence improving combustion duration and its stability. TKE shown in Figure 3.11 exhibited behaviour similar to averaged velocities presented in Figure 3.9. SVA consistently outperformed DVA, and it reached peak TKE of about $700 \text{ m}^2/\text{s}^2$ at 55° CA after TDC_{ge} which was maintained for about 30° CA , after which it rapidly decreased until it reached $70 \text{ m}^2/\text{s}^2$ at BDC, after which rate of decrease slowed down until it reached its minimum of $15 \text{ m}^2/\text{s}^2$ at TDC_f . Window showing mid-stroke compression to TDC_f was zoomed-in to prove that under any firing conditions SVA, regardless of running less spark advance, always benefited from higher TKE at ignition. Most retarded sparking events ($\lambda=1.0$, 0% EGR) for both DVA and SVA were plotted, whilst red, dashed line represented possible spark advance for SVA. There is no intersection point with DVA's TKE, thus regardless of spark advance SVA always benefits from higher TKE at ignition. For the $\lambda=1.0$ baseline condition SVA reaches TKE of $22.8 \text{ m}^2/\text{s}^2$ vs $17.5 \text{ m}^2/\text{s}^2$ (DVA) which is 30% difference. Introducing dilution would only increase that discrepancy.

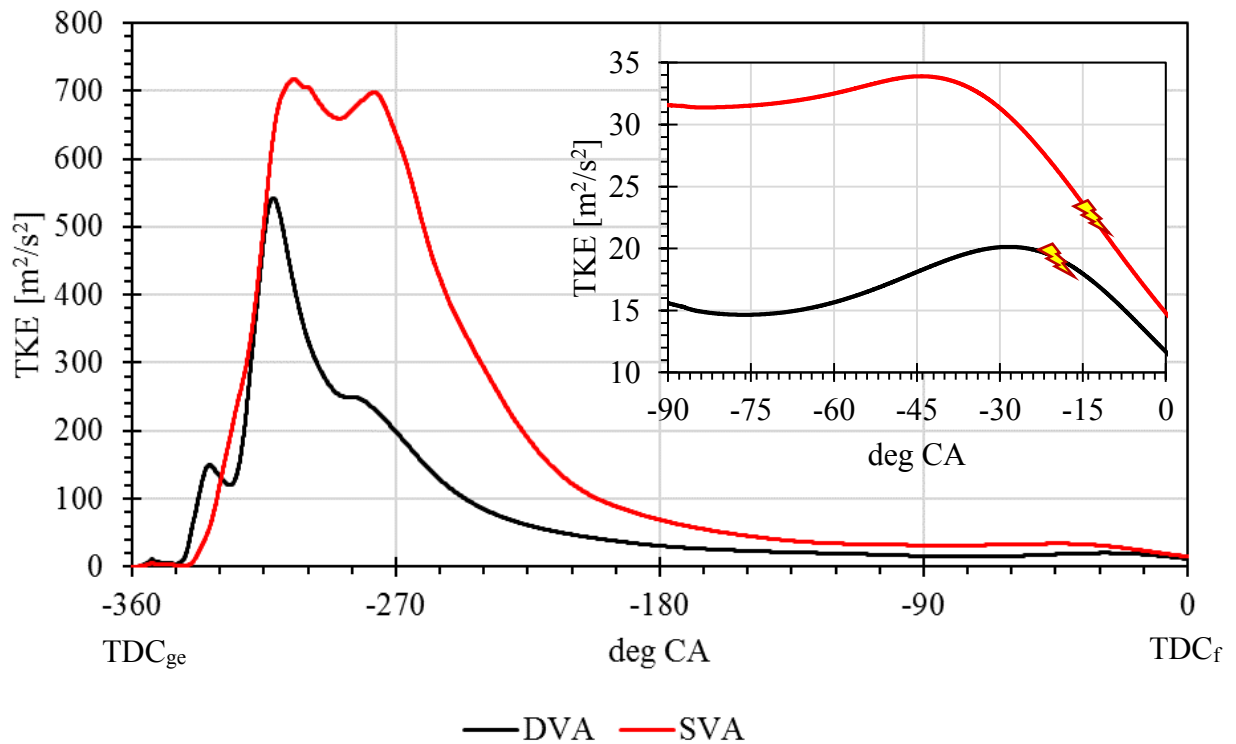


Figure 3.11 – TKE comparison for DVA and SVA at 1500 RPM, 3.6 bar GMEP with a zoom on mid-stroke compression to TDC_f region

Higher TKE observed for SVA provides an explanation for significantly shorter burn durations seen at SVA and thus faster flame speeds, given that turbulent burning velocity is proportional to the turbulence intensity and laminar burning velocity; Gülder (1991).

$$S_T \propto \sqrt{\frac{2k}{3}} S_L, \text{ where} \quad (3.1)$$

S_T – turbulent burning velocity

S_L – laminar burning velocity

$\sqrt{\frac{2k}{3}} = u'$ - root-mean-square velocity fluctuation / turbulence intensity

Laminar burning velocity is further discussed later in this chapter, together with temperature differences between DVA and SVA operating strategies.

To provide better visualisation of in-cylinder TKE differences between DVA and SVA close to ignition (20°CA before TDC_f) and give an insight into its spatial distribution, contour plots were produced. Figure 3.12 shows tumble (Y-axis = 0 mm) plane and swirl (Z-axis = +5mm) plane (exact details of which can be found in Kountouriotis, 2019).

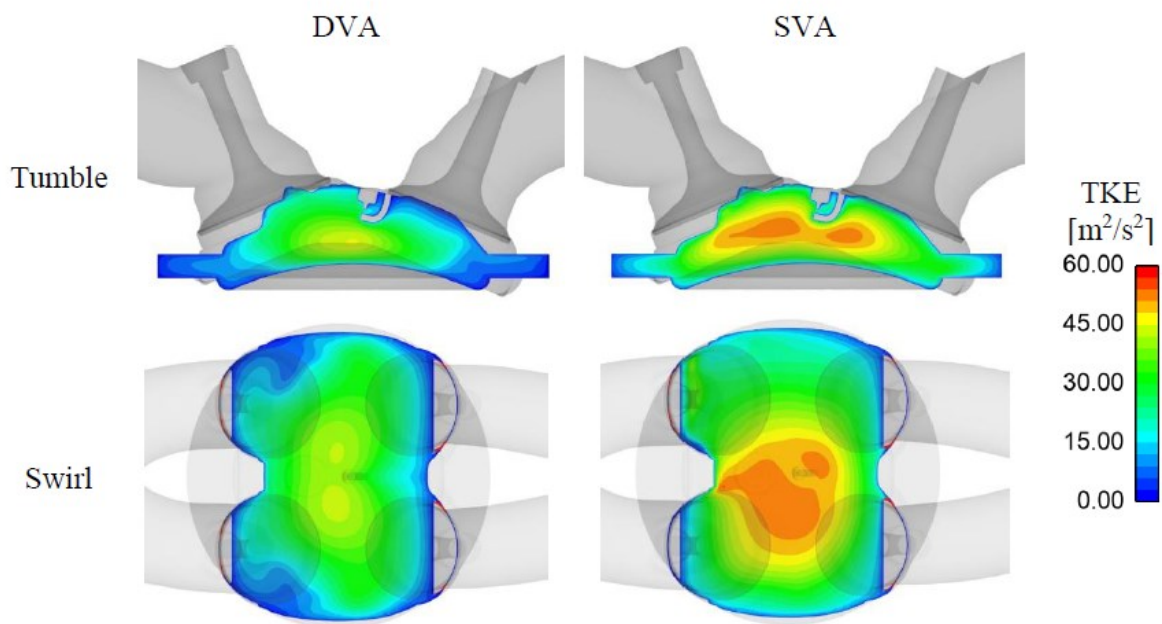


Figure 3.12 – Spatial distribution of TKE for DVA and SVA at $\lambda=1.0$, 20°CA bTDCf at 1500 RPM, 3.6 bar GMEP (Kountouriotis, 2019)

Contour plots in Figure 3.12 show that levels of TKE in the spark plug vicinity were much higher for SVA, thus flame propagation occurred much faster. In addition to that, highest TKE region for DVA can be spotted below the spark plug which is inferior to SVA location for establishing the flame kernel.

More beneficial spatial distribution, and overall higher levels of TKE for SVA over DVA and much faster burning observed for valve deactivation strategy suggest that GSFC should be improved, however as it has been already shown in Figure 3.6, fuel consumption for both DVA and SVA is almost identical, especially at low levels of dilution. To further understand why that is, and to find underlying cause, heat transfer was investigated.

Experimental data from SCRE provided values of compression polytropic coefficients (γ) calculated for each cycle. Values were then averaged over 300 consecutive cycles and presented in Figure 3.13. Compression polytropic coefficients of all SVA points are lower than their DVA counterparts (at any given lambda and EGR combination).

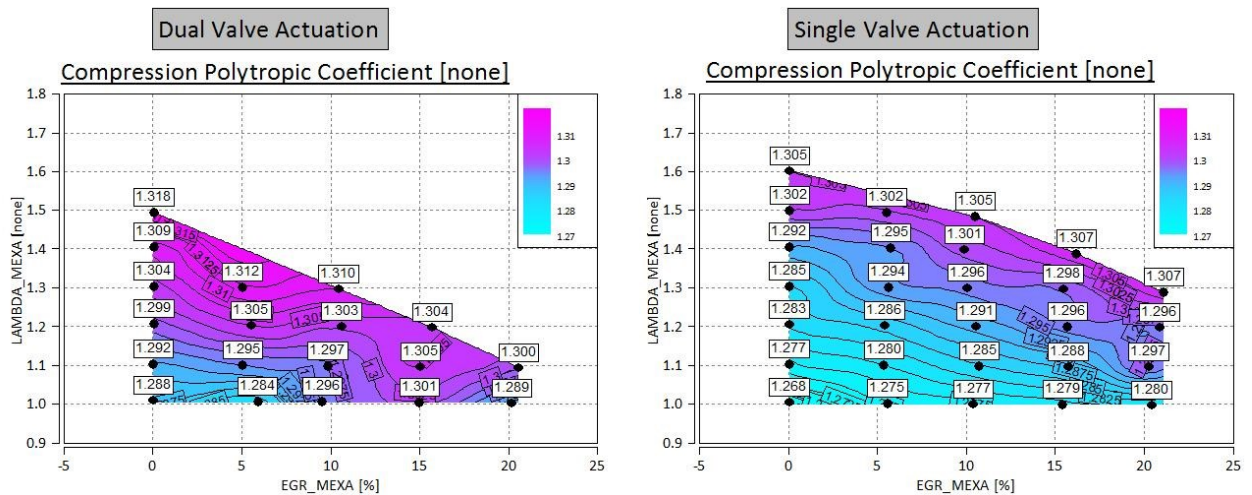


Figure 3.13 – Compression Polytropic Coefficients calculated over 300 cycles for DVA and SVA at 1500 RPM, 3.6 bar GMEP

Knowing that polytropic coefficient can only be influenced by two factors – trapped mass and heat transfer, both were scrutinised.

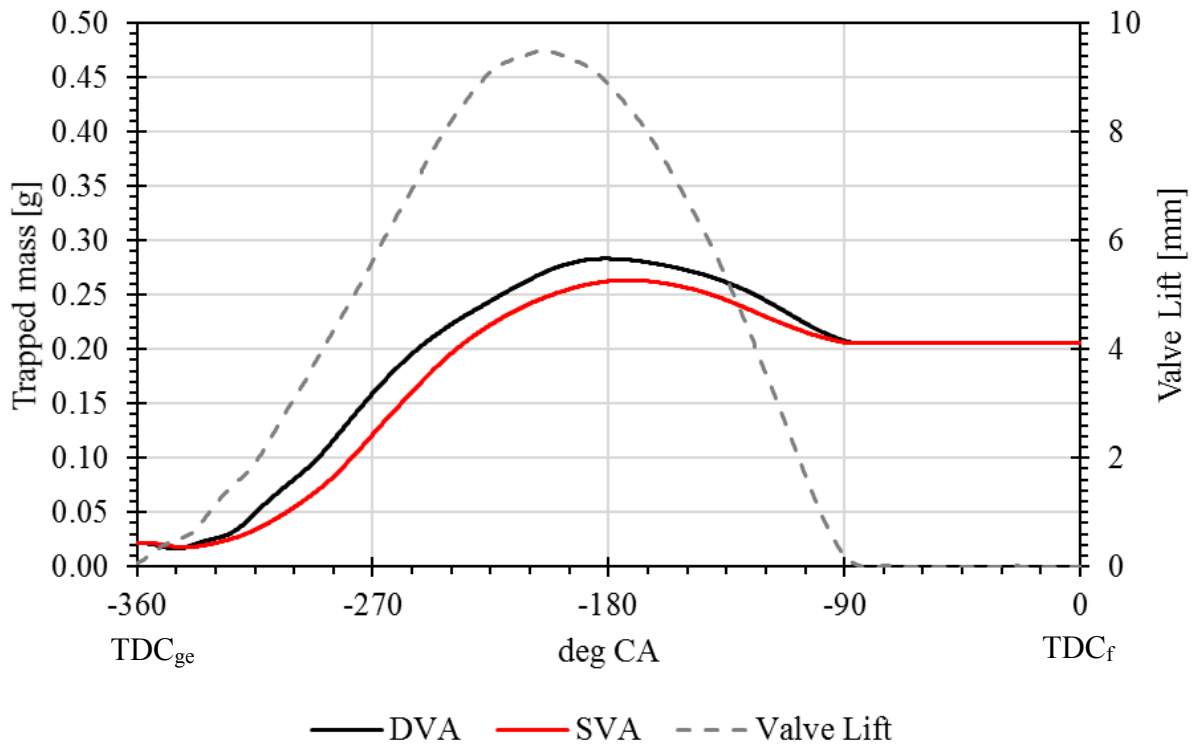


Figure 3.14 – Trapped mass for DVA and SVA at 1500 RPM, 3.6 bar GMEP

Both DVA and SVA strategies trapped exactly the same amount of mass at IVC, however DVA admitted more air into the cylinder once the valves opened and reached maximum of 0.283 g vs 0.263 g for SVA (8% difference). Following the maximum, more air was also expelled back to the intake runner through 2 intake valves during their closing phase.

In-cylinder temperature was another parameter that was looked at to fully understand lack of fuel consumption benefit of faster burning SVA operation compared to slower burning DVA. It was found that at TDC_{ge} temperatures for DVA and SVA were closely matched at 907 K and 902 K respectively. Once the intake valves started to open DVA exhibited sharper drop for the first 75° CA, after which the rate of temperature decrease slowed significantly and achieved minimum at BDC. During the initial 15° CA of the intake stroke, DVA's temperature dropped below SVA and maintained that trend until the TDC_f. The aforementioned sharper drop in temperature can be attributed to higher amount of fresh air entering the combustion chamber and thus cooling the in-cylinder mixture more effectively (Figure 3.14 and 3.15).

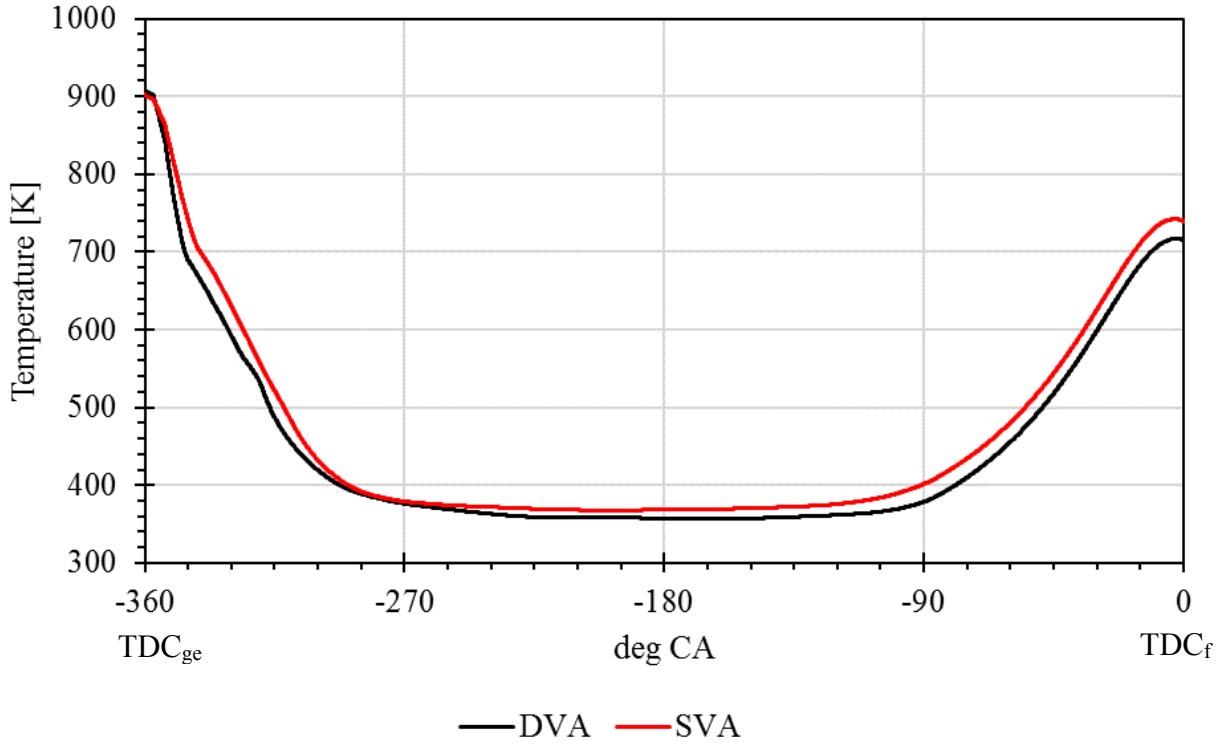


Figure 3.15 – Averaged in-cylinder temperature comparison between DVA and SVA at 1500 RPM, 3.6 bar GMEP

As it has been already discussed in Literature Review, according to Heywood (1988) and other researchers valve timings have significant impact on in-cylinder trapped residuals (RGF). CFD simulations of gas exchange process for Jaguar Land Rover AJ-20 engine found volumetric RGF to be higher for SVA – 8.1% compared to DVA – 7.0%, resulting in higher gas temperature. It is believed that this is linked to the fresh charge trapping characteristic plotted in Figure 3.14. More air was admitted and then expelled through the use of DVA, resulting in more effective “washing” of the combustion chamber, thus lowering the residual content.

Higher temperature of SVA was also linked to increased laminar flame speed and burning velocity. Theoretical and empirical correlations of the laminar flame speed with temperature, pressure, equivalence ratio ($1/\lambda$) and residuals content were presented by Metghalchi and Keck (1982) and Marshall *et al.* (2011). Also, a comprehensive review of laminar burning velocities under various conditions, for a number of fuels was presented by Aleiferis *et al.* (2013). The following, simple power correlation can be made:

$$S_u \propto S_{u0} \left(\frac{T_u}{T_0} \right)^\alpha \left(\frac{p}{p_0} \right)^\beta, \text{ where} \quad (3.2)$$

p_0, T_0 – reference temperature and pressure

S_{u0}, α, β – constants for a given fuel, equivalence ratio and diluent gas fraction

It should be noted that much greater sensitivity to temperature rather than pressure was reported by Aleiferis *et al.* (2013) which is in-line with the argument proposed earlier for faster laminar burning velocity of SVA. Bradley *et al.* (2009) also found that ethanol had a greater temperature dependence than hydrocarbons which is worth noting given that fuel used in this study had 10% ethanol content. Positive contribution of increased temperature is however countered by the higher residual content which has the opposite effect on the laminar burning velocity. Assuming that residuals can be estimated as 15% CO₂ and 85% N₂, increasing them from 7% (DVA) to 8.1% (SVA) of total volume, had a marginal effect compared to temperature. Such conclusion was reached based on a review of laminar burning velocity vs residuals mole fraction trend presented by Marshall *et al.* (2011).

Finally, CFD derived polytropic coefficients were calculated using the same calculation windows that were used during experiments (Table 3.1). Detailed description on how the start and end points of calculation are being determined can be found in Chapter 2.

Table 3.1 – Compression Polytropic Coefficients comparison between experimental and numerical values for DVA and SVA at 1500 RPM, 3.6 bar GMEP

		Start of Calculation [°CA bTDCf]		End of Calculation [°CA bTDCf]		DVA	SVA	Ratio SVA/DVA
		DVA	SVA	DVA	SVA			
$\lambda=1.0$	Experiment	57.9	53.0	20.2	13.7	1.288	1.268	0.9845
	CFD					1.335	1.317	0.9865
$\lambda=1.5$	Experiment	63.7	60.5	32.6	25.8	1.318	1.302	0.9879
	CFD					1.355	1.338	0.9875

Polytropic coefficients calculated within CFD were found to be about 3-4% higher than their experimental counterparts. Such discrepancy was most likely caused by the lack of blow-by modelling within CFD models, however ratios of SVA/DVA were very well matched (within

0.2%) for both methods. It gives an extra level of confidence in heat transfer modelling and quality of numerical simulations.

Heat transfer across the intake and compression strokes was calculated using the Angelberger *et al.* (1997) function. Again, exact details of this simulation set-up can be found in Kountouriotis (2019).

Heat transfer presented in Figure 3.16 adopts convention of heat transfer being transferred to the gas, hence most significant portion of the plot with negative values represents heat transfer to the walls, cylinder head and piston (3 modelled entities) from the gas. SVA strategy was subjected to significantly higher heat transfer (about 500W close to TDC_f , expressed as Absolute Heat Transfer Delta) which had detrimental effect on cycle thermal efficiency and offset any duration benefits gained through higher TKE discussed earlier. Delta was equal to 23.6% increase for SVA vs DVA. Finally, it can be seen in Figure 3.16 that heat transfer from the gas to the engine walls started much earlier in the cycle for SVA compared to DVA (crossing point of “0 Heat Transfer” line) and it can be attributed to gas temperature differences discussed in previous paragraph. Note that decrease in heat transfer close to TDC_f (approximately 5°C bTDC_f) was a result of flow field effects and slow piston movement, where density and temperature remained approximately constant close to TDC (motored scenario was simulated).

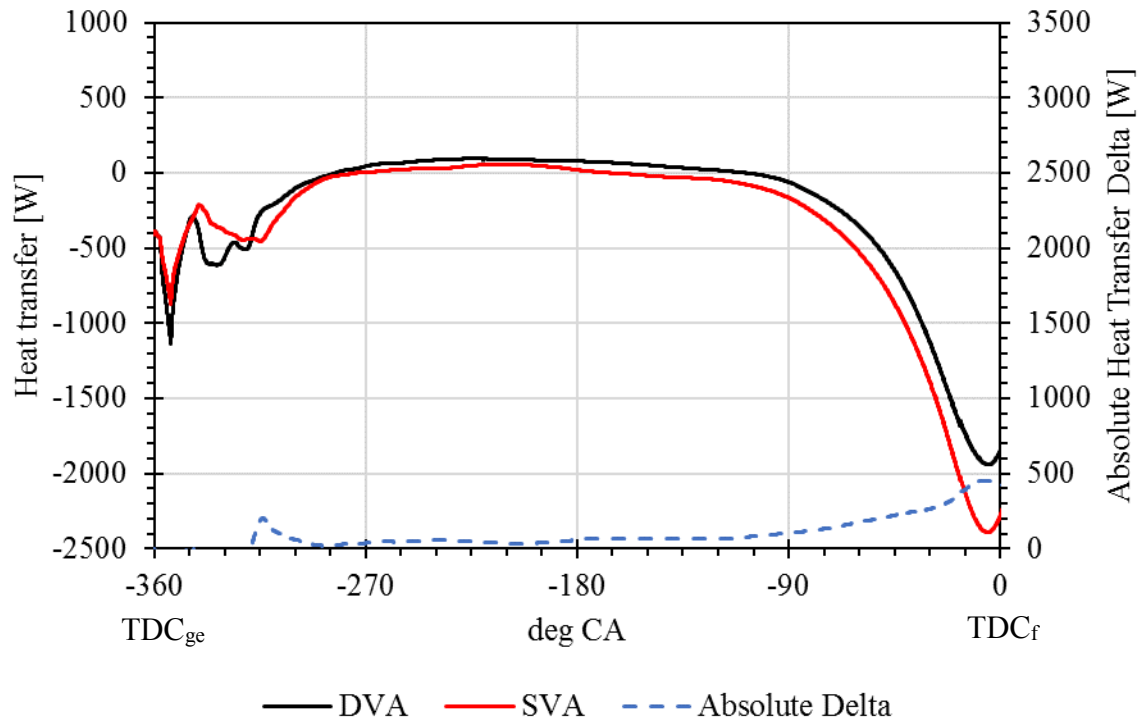


Figure 3.16 – In-cylinder heat transfer comparison between DVA and SVA at 1500 RPM, 3.6 bar GMEP

3.2.2. 1500 RPM; 7.9 bar GMEP

Increasing engine load from 3.6 bar GMEP to 7.9 bar GMEP enabled detailed study of load dependence at 1500 RPM. Presented in Figure 3.17 is the intake plenum pressure. It shows that increasing nominal load required running engine above the atmospheric pressure (using the DC motor driven supercharger to generate boost) for vast majority of tested points. Trend observed for lower load of decreased plenum pressure requirements for SVA holds.

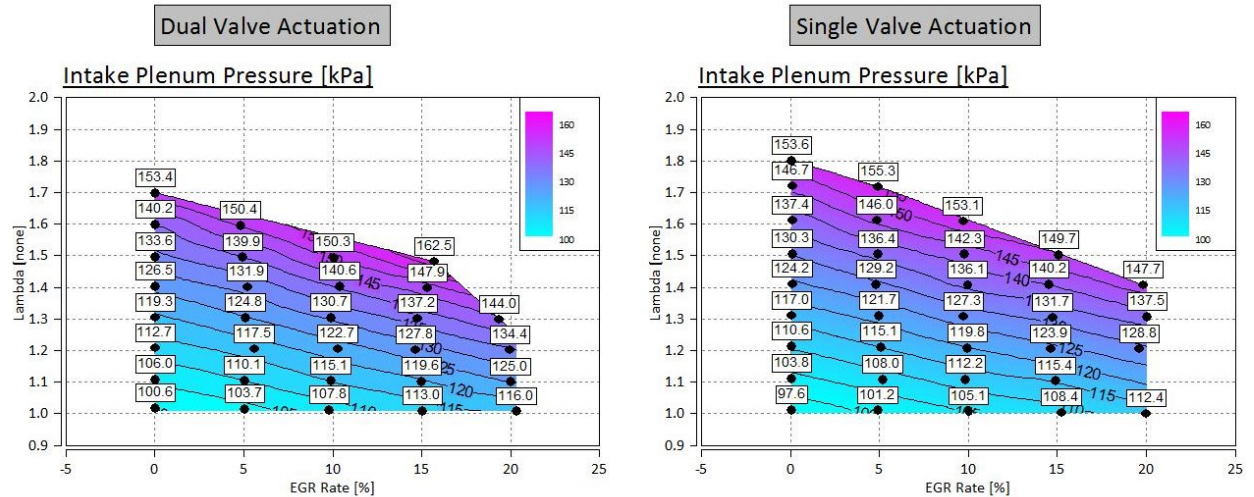


Figure 3.17 - Intake Plenum Pressure comparison of DVA vs SVA at 1500 RPM, 7.9 bar GMEP

Adopting SVA strategy enabled significant increase in dilution tolerance. Adhering to NVH derived combustion stability limit of 3% CoV of NMEP, dilution was increased by 0.35 lambda at 0% EGR (from 1.40 to 1.75) and 0.25 lambda at 20% EGR (from 1.10 to 1.35). Shift is illustrated by thick black line in Figure 3.18. It can also be seen that wide area, represented by light blue, was kept below 1.5% CoV of NMEP which can be considered as a very steady and low variability region, presenting high potential for engine calibration in real-life applications.

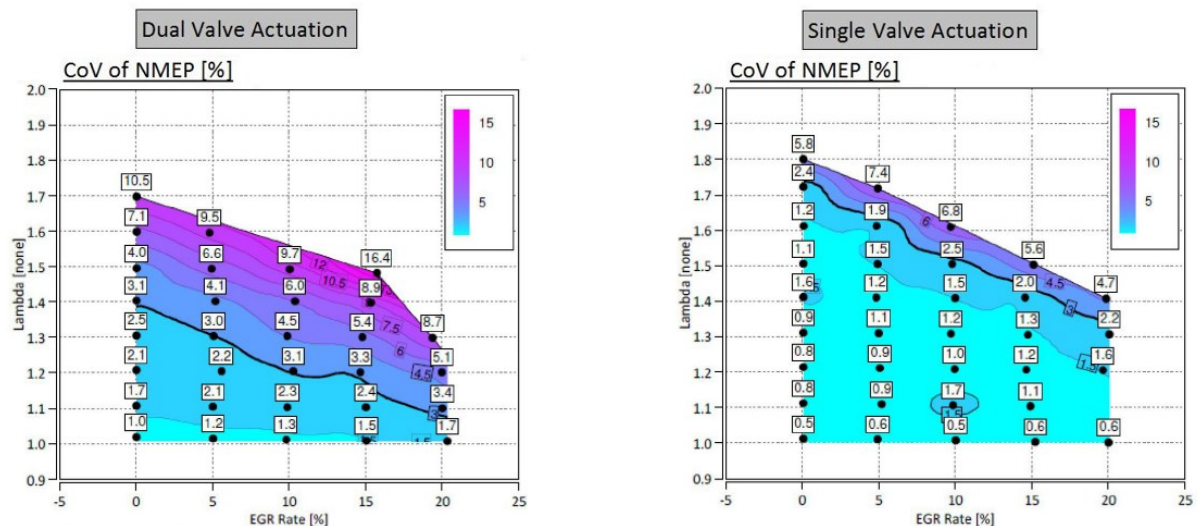


Figure 3.18 - Combustion stability comparison of DVA vs SVA at 1500 RPM, 7.9 bar GMEP

Combustion duration analysis exhibited the same trends that were seen at lower load i.e. significant advance of DVA ignition angles compared to SVA for equivalent dilution levels (6-10°CA), whilst points of similar combustion stability are well matched. It indicates that combustion duration is directly linked with its stability at given speed/load point regardless of dilution level. Ignition to 10% MFB was again faster for SVA, and there was a trend showing that the more dilution, the larger the difference between the DVA and SVA. At 0% EGR level, the gap between SVA and DVA more than doubled between $\lambda=1.0$ and $\lambda=1.7$ (5.1°CA and 11.0°CA respectively), and same behaviour was seen for higher EGR levels (5.3°CA and 12.0°CA at 5% EGR; 5.8°CA and 10.0°CA at 10% EGR; 6.5°CA and 12.0°CA at 15% EGR; 7.6°CA and 9.8°CA at 20% EGR).

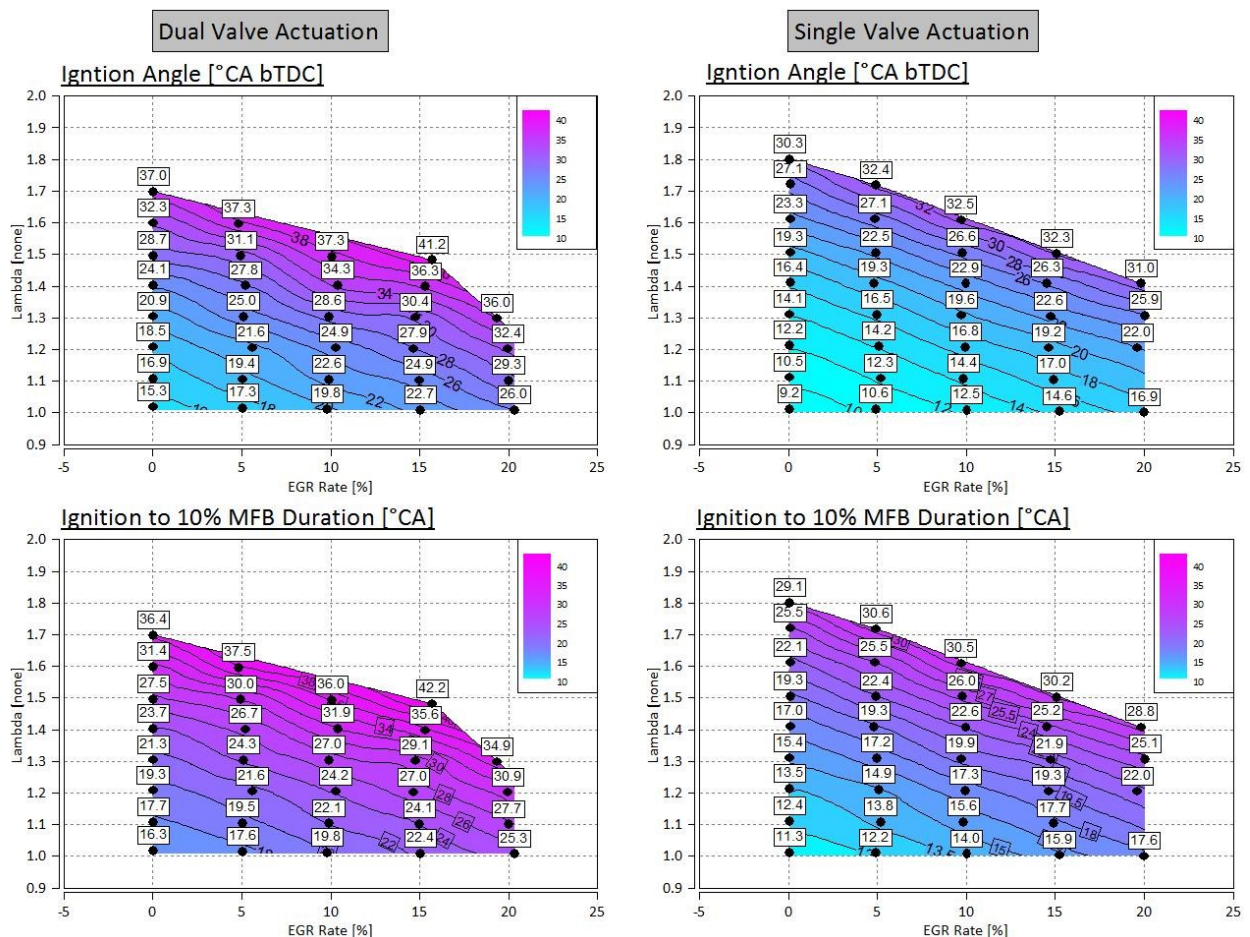


Figure 3.19 - Ignition advance and initial burn duration comparison of DVA vs SVA at 1500 RP, 7.9 bar GMEP

10 to 90% MFB exhibited fairly constant offset between DVA and SVA of 0.5 - 3.0 °CA at the same dilution levels (contrary to the 0-10% MFB trend described above) which proves that most of the difference in the combustion process should be attributed to its early stages.

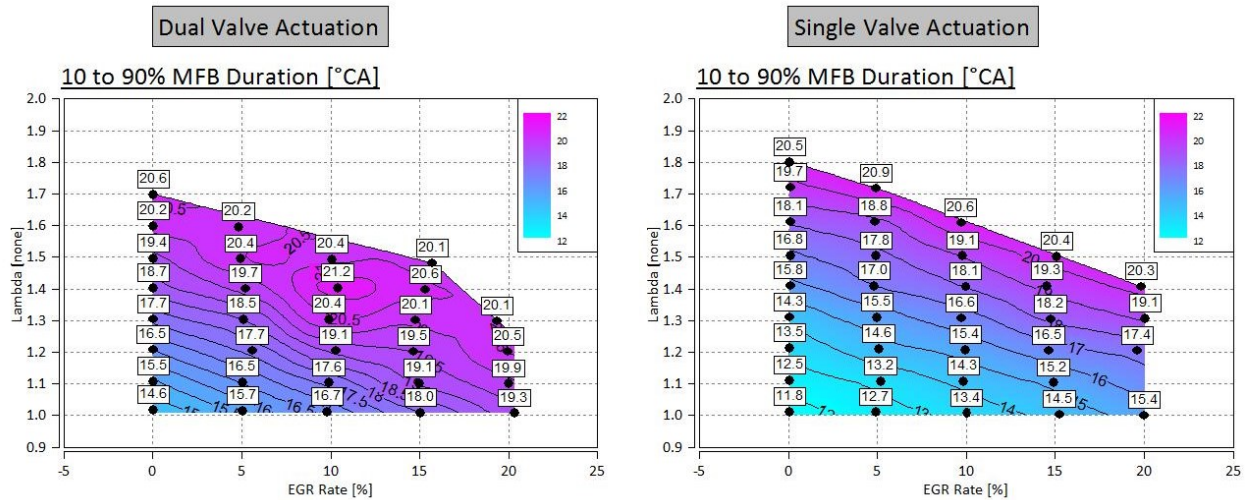


Figure 3.20 - 10 to 90 % MFB duration comparison of DVA vs SVA at 1500 RPM, 7.9 bar GMEP

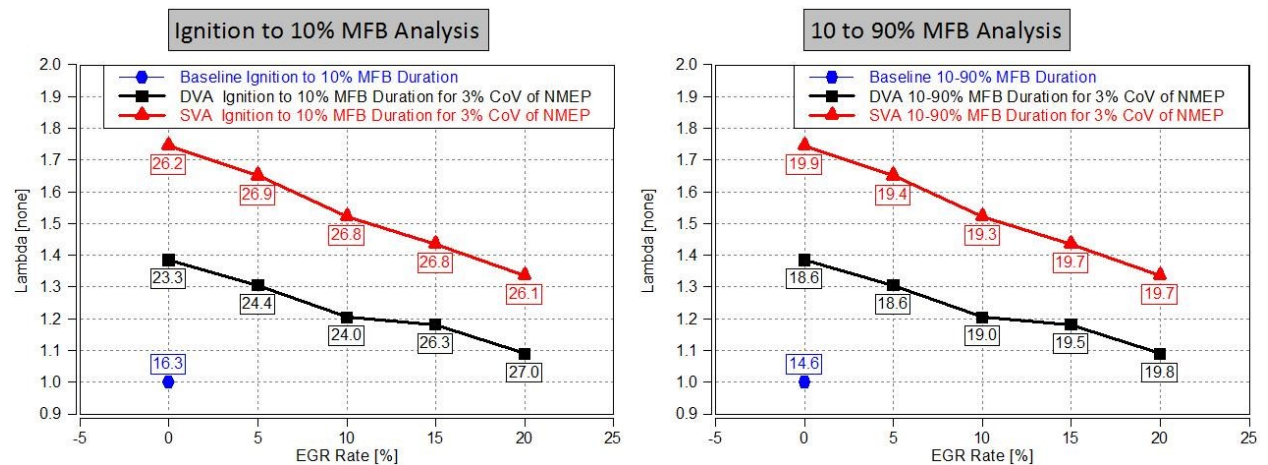


Figure 3.21 - DVA vs SVA burn duration analysis at 1500 RPM, 7.9 bar GMEP

In-cylinder flow properties analysis was performed using the same methodology as for the 3.6 bar GMEP condition (comparison of $\lambda=1.0$ and 0% EGR points for DVA vs SVA). During the course of investigation, velocity and TKE profiles were produced for 3.6 bar GMEP case at $\lambda=1.5$, showing no qualitative differences once compared to baseline $\lambda=1.0$ scenario. Exact details and data plotted for this sensitivity study can be found in Kountouriotis (2019).

Velocity profiles of 7.9 bar GMEP represented almost double the intake plenum pressure when compared to 3.6 bar GMEP (100.6 kPa vs. 53.2 kPa and 97.6 kPa vs. 51.2 kPa for DVA and

SVA respectively), however there was little quantitative difference between two loads. The most pronounced is lower value of first velocity peak at about $35^\circ\text{CA aTDC}_{\text{ge}}$ for DVA that is about 20% lower at 40 m/s when compared to 3.6 bar GMEP condition. SVA's peak value remained unchanged at about 70 m/s, and the ratio of SVA/DVA followed almost identical trend past BDC with the only difference being slightly higher absolute values, constantly recorded at about 0.2 delta.

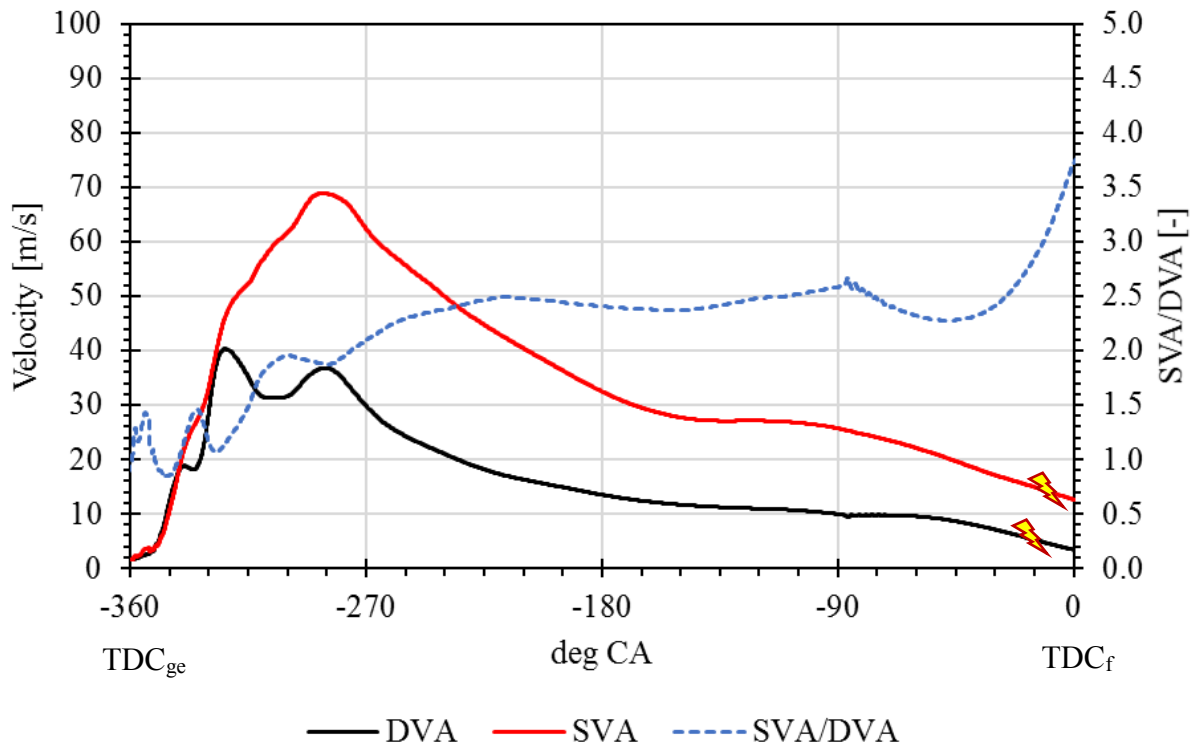


Figure 3.22 - Averaged in-cylinder velocity for DVA and SVA at 1500 RPM, 7.9 bar GMEP

Tumble and swirl representation during intake and compression (Figure 3.23) showed no noticeable difference when compared to 3.6 bar GMEP condition both qualitatively and quantitatively. Swirl remained the governing motion for SVA scenario and reverse-tumble occurred past BDC.

The in-cylinder averaged TKE was consistently higher for SVA, however its peak values were slightly lower for SVA ($690 \text{ m}^2/\text{s}^2$ vs $700 \text{ m}^2/\text{s}^2$ reached at 3.6 bar GMEP), and significantly lower for DVA ($340 \text{ m}^2/\text{s}^2$ vs $530 \text{ m}^2/\text{s}^2$) once compared to lower load case. Within the last half

of the compression stroke, area where ignition takes place, TKE exhibited the same behaviour already discussed for 3.6 bar GMEP running condition, however its values were slightly higher (shift of about 10%). Higher TKE observed for SVA, and the delta between DVA and SVA growing with increased ignition advance was in good agreement with burn duration differences between DVA and SVA progressing with increased dilution.

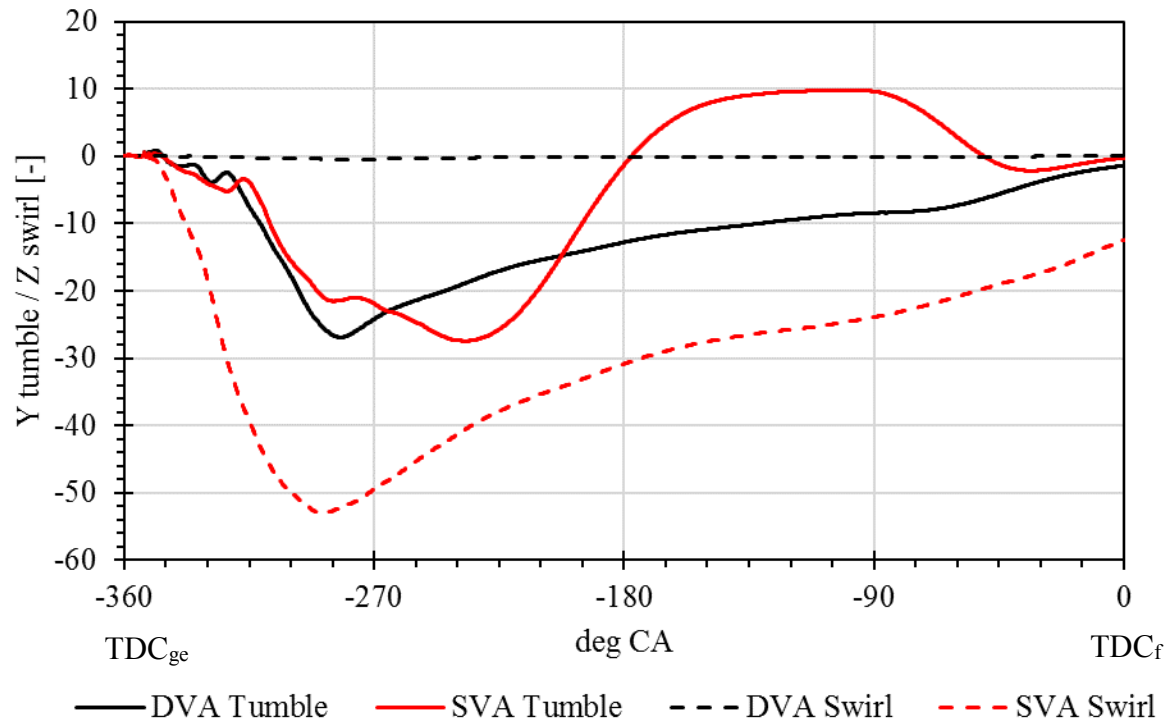


Figure 3.23 - Tumble and swirl motion representation for DVA and SVA at 1500 RPM, 7.9 bar GMEP

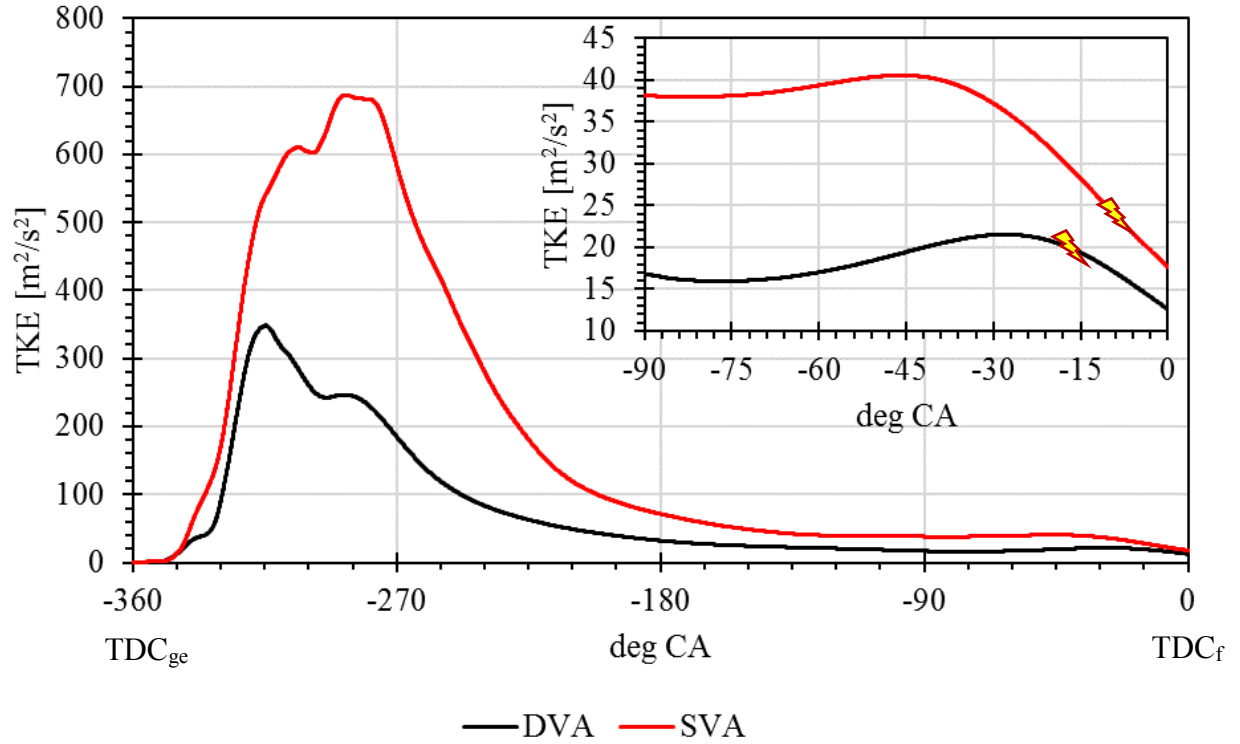


Figure 3.24 - TKE comparison for DVA and SVA at 1500 RPM, 7.9 bar GMEP with a zoom on mid-stroke compression to TDC region

Having analysed burn durations and establishing that deactivating one valve at 7.9 bar GMEP enabled significantly shortening duration due to increased TKE in the ignition part of the cycle, fuel consumption (GSFC) was investigated. Same dilution levels showed no benefit of adopting SVA over DVA, whilst few points actually indicated worse performance of SVA. There was no clear trend when comparing two datasets and only points close to the limit achieved for DVA exhibited differences larger than 1%, however these points had CoV of NMEP within 9-16% (for DVA scenario), and due to presence of misfires and partial burning cycles characterising high CoV of NMEP, the differences should be dismissed. Practical engine calibration would not be possible within this region.

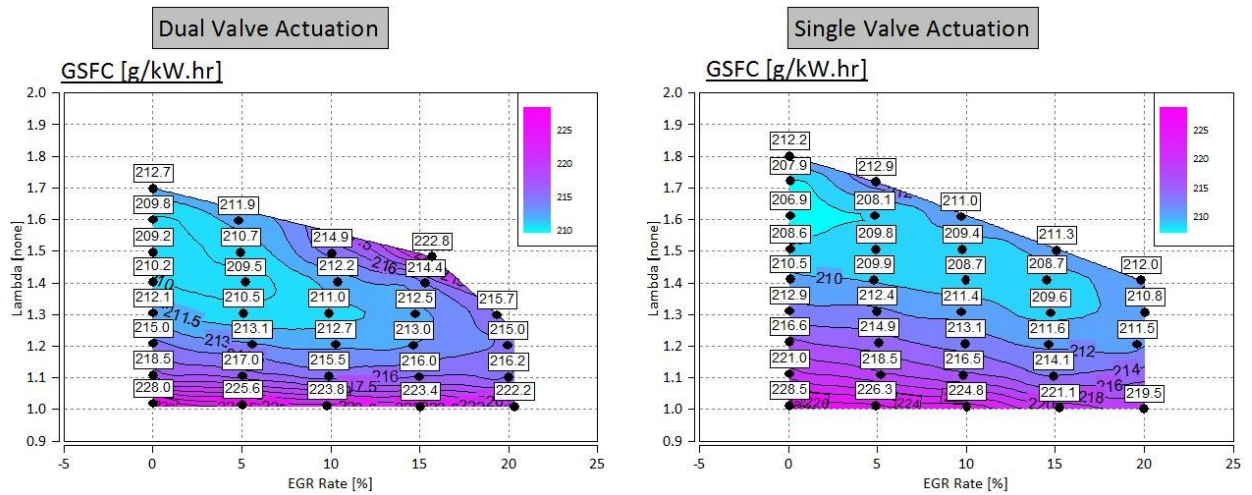


Figure 3.25 – GSFC comparison of DVA vs SVA at 1500 RPM; 7.9 bar GMEP

Maximum fuel consumption reduction of 3.1%, realised by running SVA, was gained at 20% EGR, when lambda was increased from 1.1 to 1.35 compared to DVA. The highest absolute GSFC benefit that was confined within the 3% CoV of NMEP criteria was achieved when running lean (with 0% EGR) at 1.75 lambda using SVA, reducing fuel consumption by 8.6% compared to stoichiometric DVA condition. Along the 0% EGR line DVA could achieve reduction of 7.9% at lambda 1.4 (vs baseline stoichiometric condition), hence the relative benefit of SVA stood at only 0.8%.

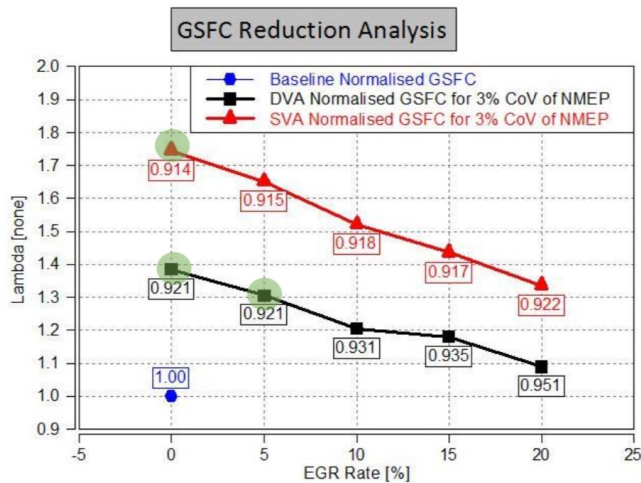


Figure 3.26 - 1500 RPM, 7.9 bar GMEP GSFC reduction analysis for a fixed combustion stability limit of 3% CoV of NMEP

Limited GSFC benefits of running lean, using more thermodynamically attractive mixtures, without the penalty of slower burn durations when adopting SVA over DVA were again attributed to detrimental effects of heat transfer. Qualitatively profiles presented in Figure 3.27 were well matched to these presented in Figure 3.16 for 3.6 bar GMEP running condition. Quantitatively, delta showing much higher heat transfer from gas to the engine walls, head and piston and for SVA became higher during compression stroke. Delta was calculated to be approximately 50% in this case which is more than double of 23.6% observed at lower load for the same speed of 1500 RPM.

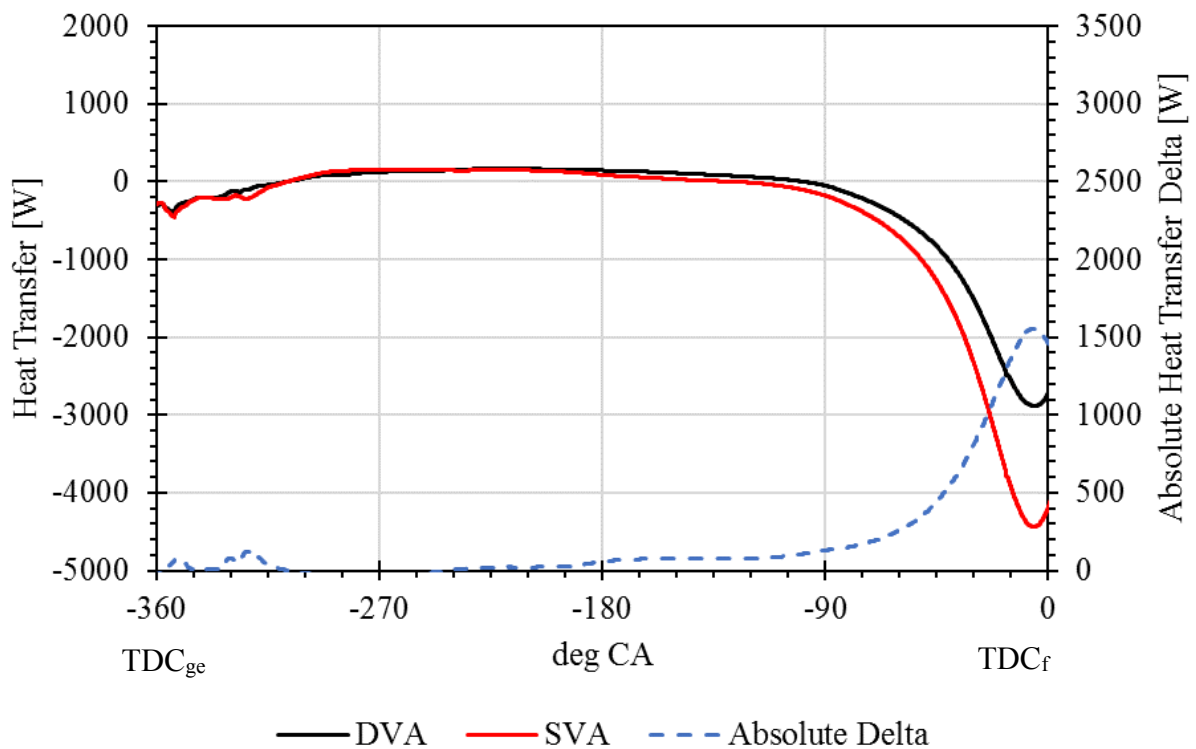


Figure 3.27 - In-cylinder heat transfer comparison between DVA and SVA at 1500 RPM, 7.9 bar GMEP

3.2.3. 2000 RPM; 9.8 bar GMEP

Third point used in this study (another high residence point used for engine certification) enabled investigation of in-cylinder motion, combustion performance and heat transfer effects under heavily boosted conditions (up to about 1 bar gauge pressure, as shown in Figure 3.28) and studying engine speed effects on the aforementioned quantities.

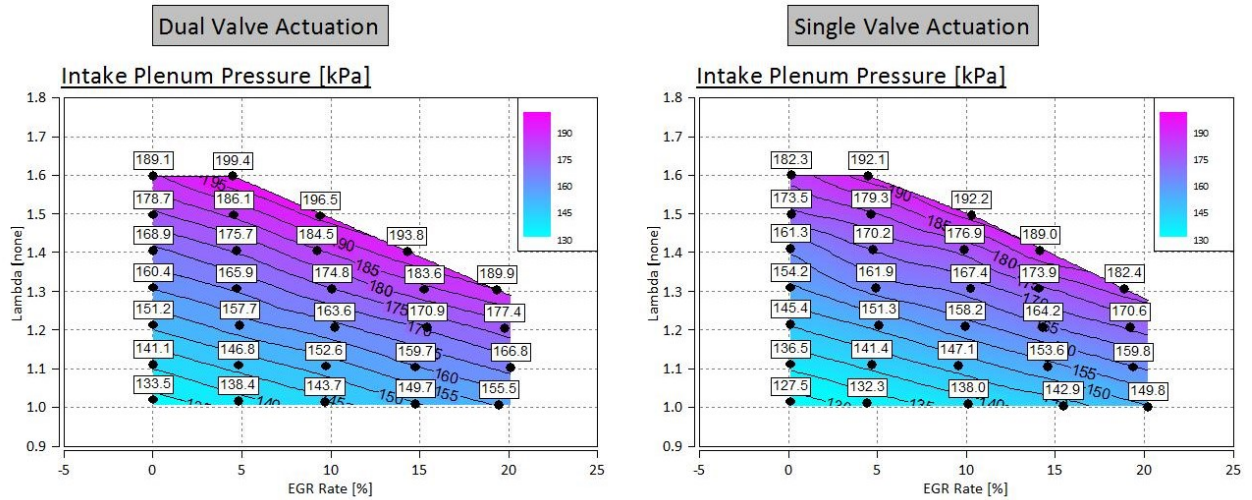


Figure 3.28 - Intake Plenum Pressure comparison of DVA vs SVA at 2000 RPM, 9.8 bar GMEP

Similarly to two previously discussed 1500 RPM points, dilution tolerance was increased once the SCRE was run using SVA strategy, however to much lesser extent. Thick black line showing 3% CoV of NMEP limit in Figure 3.29 was raised by less than 0.1 lambda for all EGR levels. It should be noted that stability profiles were very different for DVA and SVA. DVA exhibited more gradual loss of stability along the y-axis that represents lambda increase, whilst SVA had extremely large area where CoV of NMEP is kept below 1.5%, and then within just 0.1 lambda delta, CoV rapidly increased by the factor of approximately 6 to 9 (up to 15% EGR).

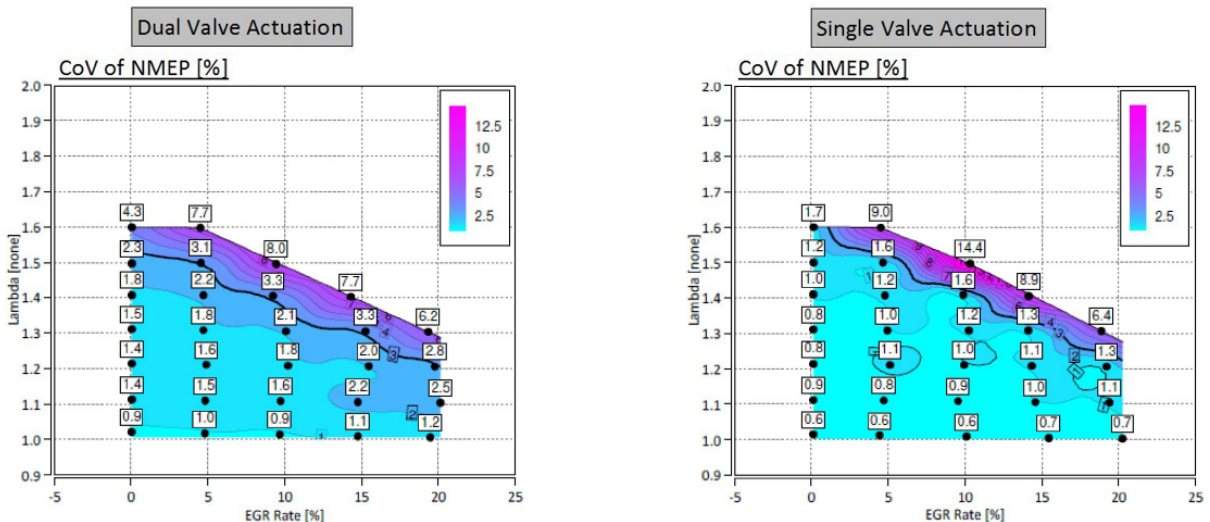


Figure 3.29 - Combustion stability comparison of DVA vs SVA at 2000 RPM, 9.8 bar GMEP

Combustion duration parameters at 2000 RPM, 9.8 bar GMEP followed trends observed for 1500 RPM conditions. Ignition was significantly advanced for DVA, compared to SVA for equivalent dilution levels. Initial phase of combustion (ignition to 10% MFB) was much faster for SVA with a trend of delta between SVA and DVA increasing when dilution was being increased i.e. they were most closely matched at stoichiometric condition with delta of 5.1 °CA, whilst points on the upper edge of the contour map (highest lambda for given EGR dilution) recorded deltas of 12.3 to 16.1 °CA (again increasing with EGR dilution).

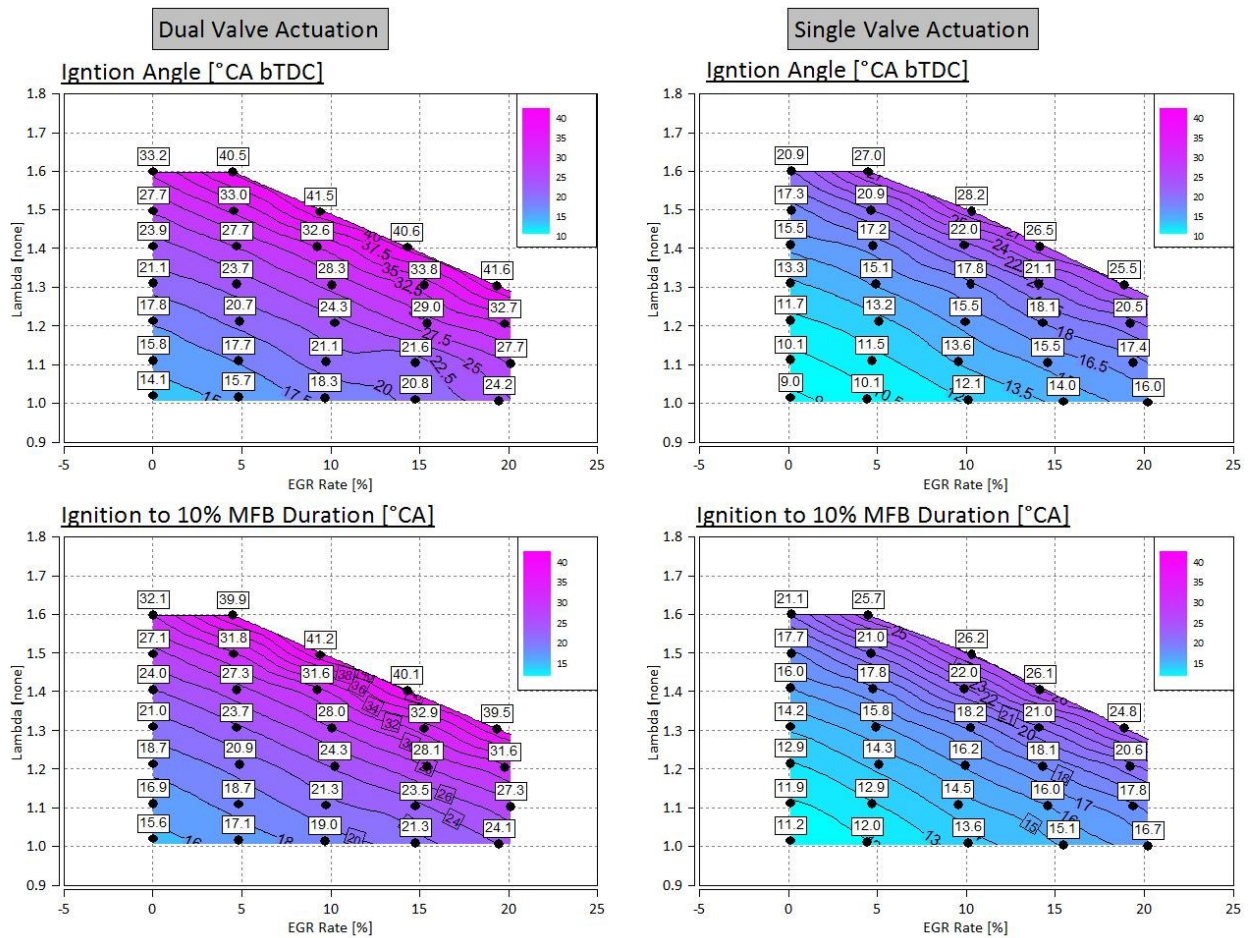


Figure 3.30 - Ignition advance and initial burn duration comparison of DVA vs SVA at 2000 RPM, 9.8 bar GMEP

Later stage of combustion, represented by 10 to 90% MFB, had constant offset (within 3°CA) between DVA and SVA at any given dilution level with SVA consistently outperforming DVA. Much larger discrepancies in burn durations, not seen before at 1500 RPM, were present when comparing DVA and SVA for points that fall on 3% CoV of NMEP line. That differences could

be attributed to much smaller lambda delta between two lines (three points that exhibit biggest differences of approximately 10 °CA actually had almost identical lambda values). Similarly 10 to 90% MFB durations showed much larger deltas than in the previously analysed cases, however to much smaller extent (about 1.5 °CA).

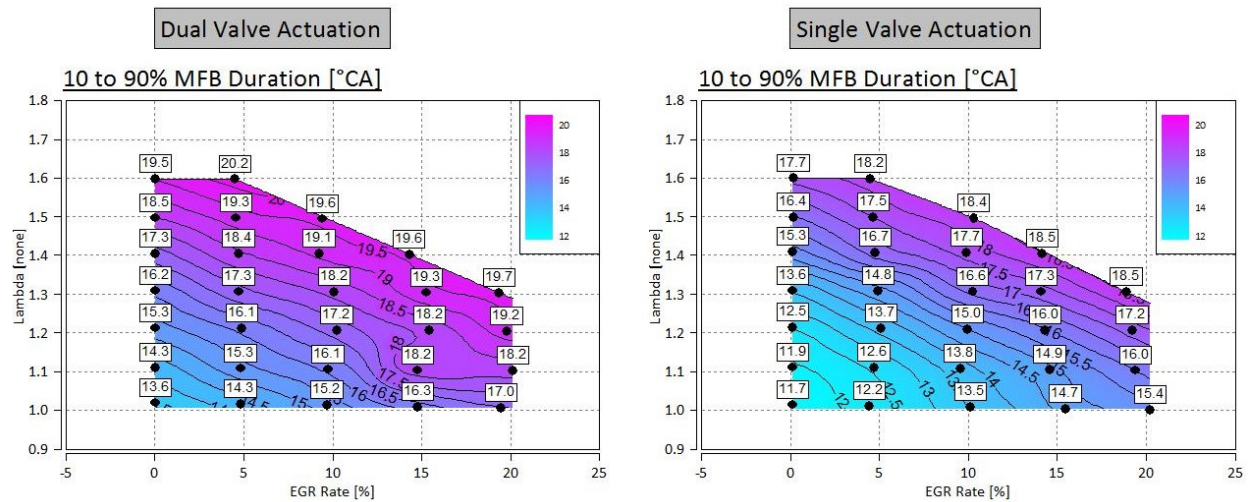


Figure 3.31 - 10 to 90 % MFB duration comparison of DVA vs SVA at 2000 RPM, 9.8 bar GMEP

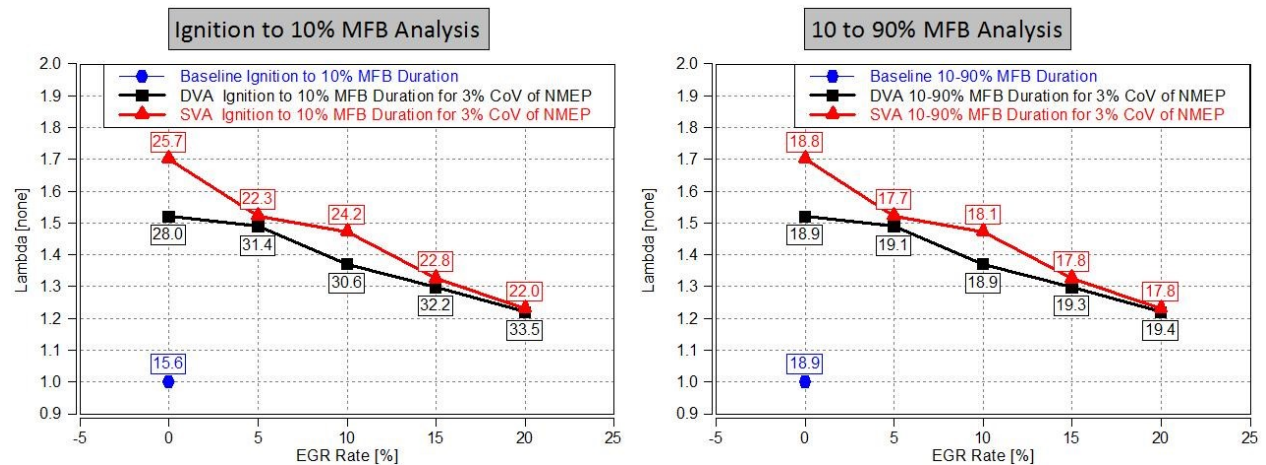


Figure 3.32 - DVA vs SVA burn duration analysis at 2000 RPM, 9.8 bar GMEP

Increasing the engine speed had an effect on the averaged in-cylinder flow characteristics. Whilst the averaged velocity profile shapes during intake and compression strokes showed no qualitative difference in their shapes (especially compared to 7.9 bar GMEP), the magnitude was higher throughout. DVA and SVA carried velocities of 20 m/s and 48 m/s through BDC which was 43% and 50% higher compared to (14 m/s and 32 m/s) 1500 RPM, 7.9 bar GMEP.

At TDC_f these values dropped to approximately 18 m/s and 6 m/s, maintaining SVA/DVA ratio closely matched to these shown in Figures 3.9 and 3.22 for 1500 RPM cases. It is believed that increased engine speed (by 33%), hence faster piston motion was responsible for aforementioned increase in velocity magnitude.

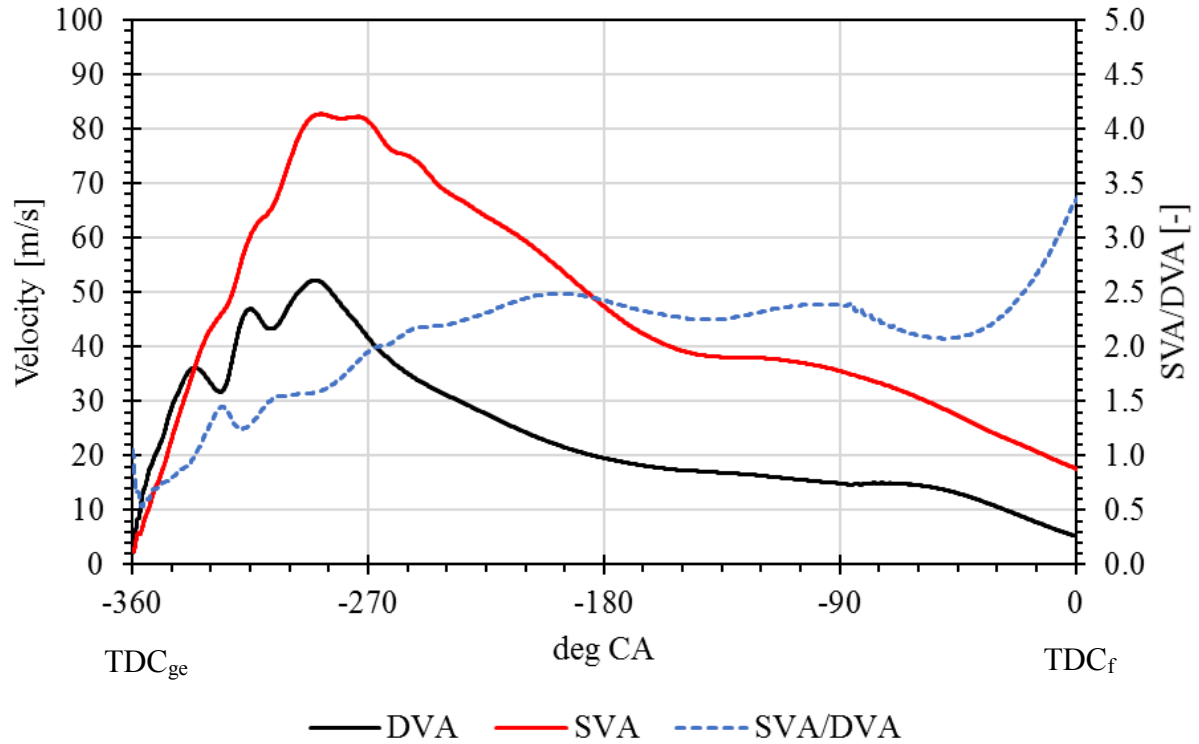


Figure 3.33 - Averaged in-cylinder velocity for DVA and SVA at 2000 RPM, 9.8 bar GMEP

Tumble and swirl motion followed exactly the same scenario. There was no change in shape, however increased piston speed resulted in higher peak values.

Due to increased in-cylinder velocities, the TKE also gained higher magnitude. Figure 3.35 showed that maximum TKE values of about $48 \text{ m}^2/\text{s}^2$ and $87 \text{ m}^2/\text{s}^2$ within the compression stroke for DVA and SVA could be achieved, which was approximately double the values recorded at 1500 RPM, 7.9 bar NMEP. These quantities provided good evidence for increased speed of combustion. Both ignition to 10% MFB and 10 to 90% MFB were closely matched for all conditions when expressed in $^\circ\text{CA}$, which translates to 33% shorter duration for 2000 RPM (vs 1500 RPM) when converted to time domain (important to note when using $^\circ\text{CA}$ and time related quantities in the discussion and analysis).

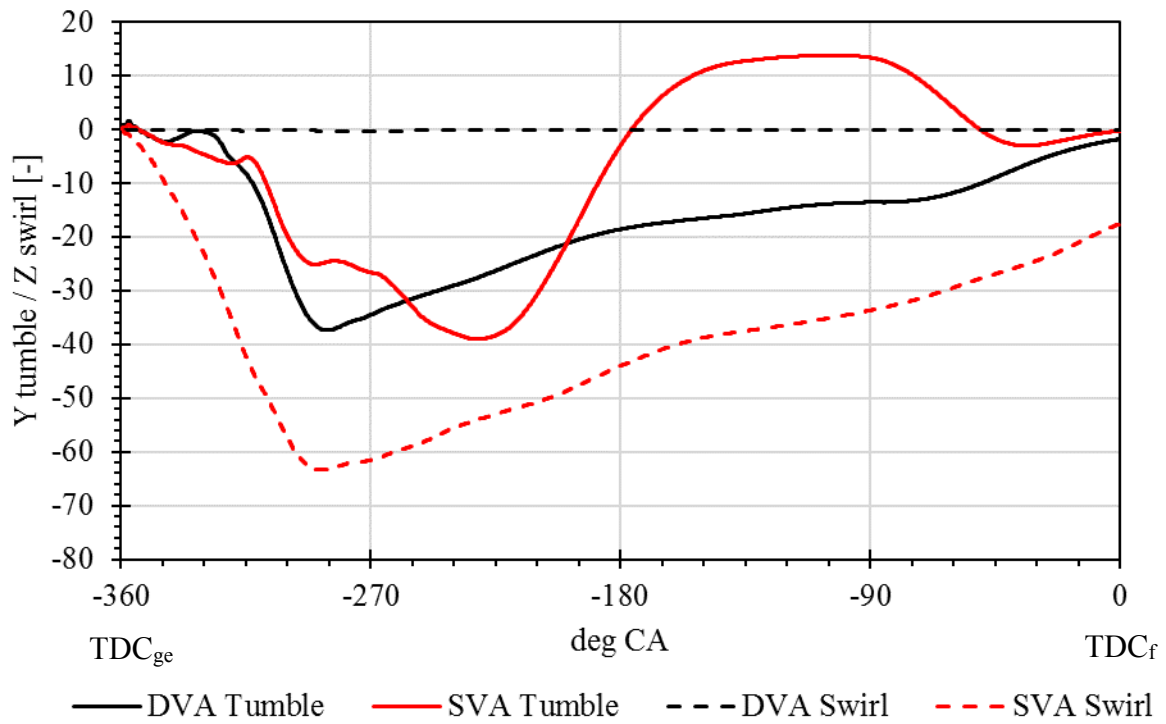


Figure 3.34 - Tumble and swirl motion representation for DVA and SVA at 2000 RPM, 9.8 bar GMEP

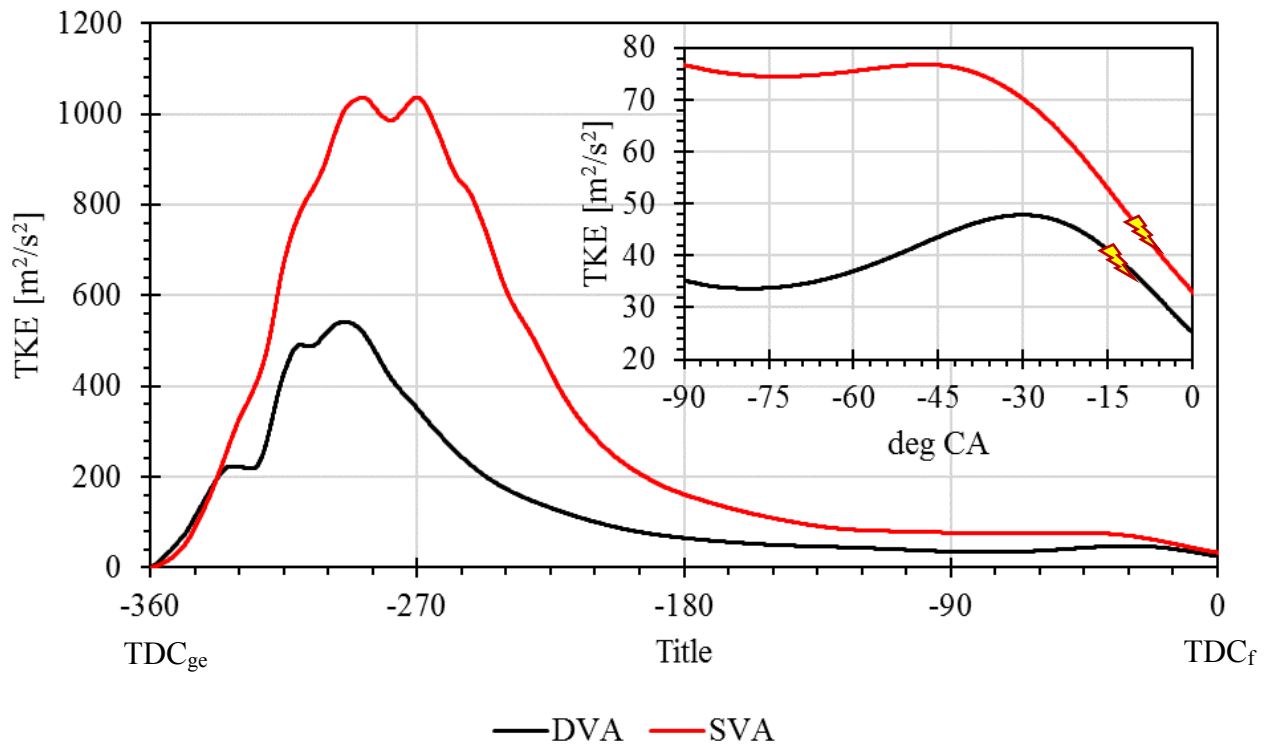


Figure 3.35 - TKE comparison for DVA and SVA at 2000 RPM, 9.8 bar GMEP with a zoom on mid-stroke compression to TDC region

Fuel consumption analysis showed that there was no benefit in running SVA for equivalent dilution levels. Values were very closely matched (within 1%), however quite often SVA performed marginally worse than DVA.

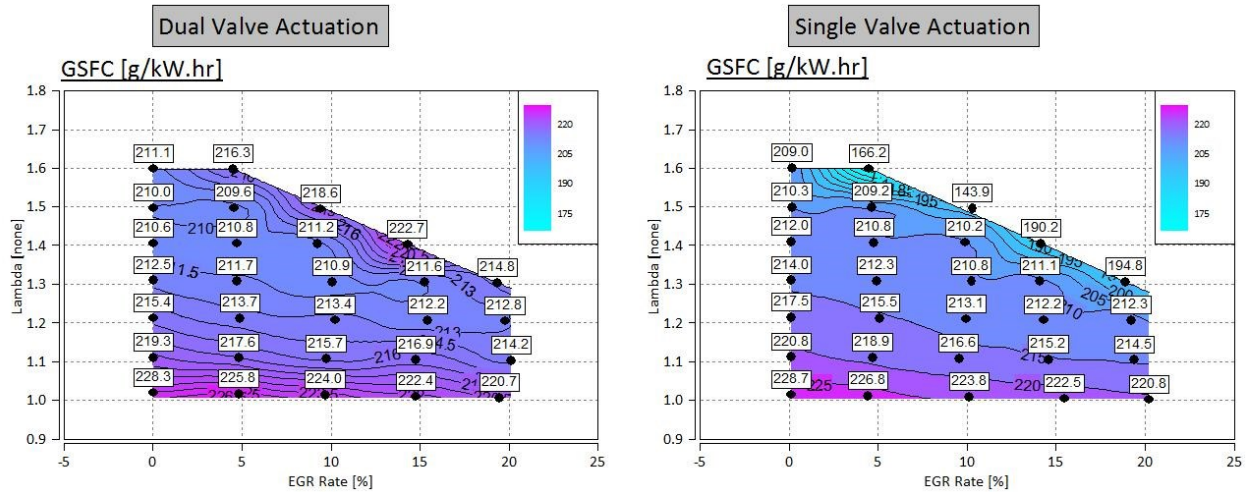


Figure 3.36 - GSFC comparison of DVA vs SVA at 2000 RPM; 9.8 bar GMEP

Once GSFC comparison was made under the 3% CoV of NMEP criteria, SVA performed better than DVA, however due to much smaller delta in lambda, differences were negligible for most cases. Maximum benefit of only 1.3% (at 10% EGR) was gained when one of the valves was deactivated.

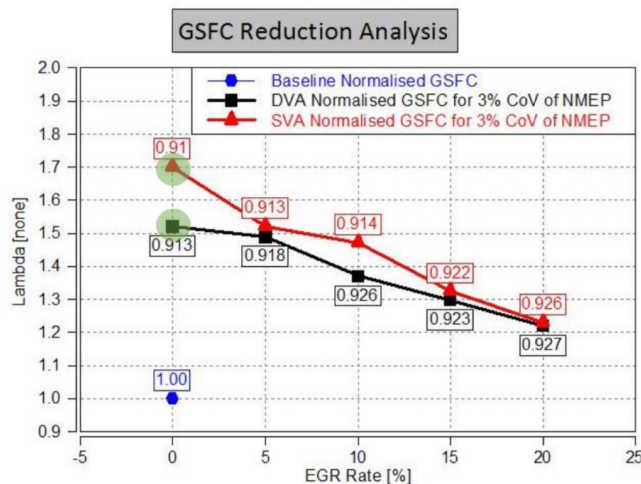


Figure 3.37 - 2000 RPM, 9.8 bar GMEP GSFC reduction analysis for a fixed combustion stability limit of 3% CoV of NMEP

Highest speed and load point analysed in this study achieved the lowest fuel consumption benefit, regardless of the significantly increased in-cylinder motion and thus faster burn. It can be attributed to heat transfer from the gas to cylinder walls, head and piston. Figure 3.38

illustrates that heat transfer delta between DVA and SVA was almost 3000 W at the end of compression stroke (56% increase). As for the previously described 1500 RPM points, higher heat transfer offset favourable burn metrics of SVA. It can be seen that the bigger the heat transfer differences, the lower the fuel consumption benefit.

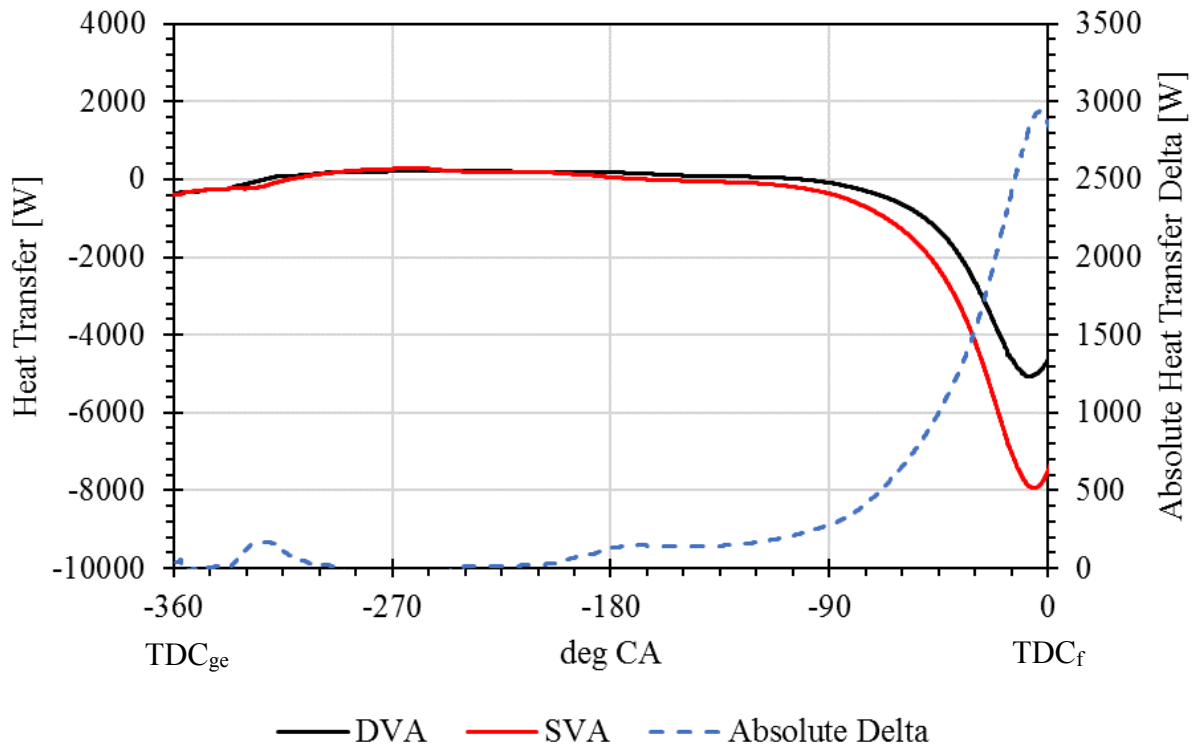


Figure 3.38 - In-cylinder heat transfer comparison between DVA and SVA at 2000 RPM, 9.8 bar GMEP

3.3. Heat Transfer Sensitivity Study

Previous section discussed and showed that increased heat transfer from gas to engine walls, head and piston of SVA was responsible for offsetting benefits of faster combustion which was particularly pertinent in the 2000 RPM case. Heat transfer calculations were based on the Angelberger model (Angelberger *et.al.* 1997), and used the following equation:

$$q_w = -\frac{\rho_w c_p u_\tau T_w}{\theta^+} \ln\left(\frac{T}{T_w}\right), \text{ where} \quad (3.3)$$

q_w heat transfer

ρ_w density

C_p specific heat capacity at constant pressure

u_τ friction velocity

T_w temperature

$$\theta^+ = \begin{cases} Pr y^+, & y^+ \leq 13.2 \\ 2.075 \ln(y^+) + 3.9, & y^+ > 13.2 \end{cases} \quad (3.4)$$

$$u_\tau = \frac{u - u_w}{u^+} \quad (3.5)$$

$$u^+ = \begin{cases} y^+, & y^+ \leq y_m^+ \\ \frac{1}{\kappa} \ln(E y^+), & y^+ > y_m^+ \end{cases} \quad (3.6)$$

The y_m^+ satisfies the equation $y_m^+ = \frac{1}{\kappa} \ln(E y_m^+)$, κ is the von Karman constant ($\kappa = 0.4$) and E is an empirical coefficient which depends on the turbulence model which is $E = 9$ for k- ϵ RNG. Pr is the Prandtl number, u is the tangential fluid velocity and u_w is the tangential wall velocity. For the k- ϵ model the y^+ is given by: $y^+ = \rho C_\mu^{1/4} k^{1/2} y / \mu$ with $C_\mu = 0.085$ for k- ϵ RNG model.

Each of the parameters that affect heat transfer, presented in Equation 3.3, was scrutinised to understand its contribution towards the heat transfer delta between DVA and SVA.

Starting with density (ρ_w), it was concluded that this parameter did not play important role in contributing towards delta between DVA and SVA due to the fact that trapped mass at IVC was same for DVA and SVA, hence no difference in charge density.

Specific heat capacity (C_p) is a function of gas composition and its temperature. Given that different amount of residuals were trapped (higher for SVA), this parameter was investigated further. At 3.6 bar GMEP, 1500 RPM condition DVA's residual content was lower than SVA's

at 7% vs 8.1%. Higher RGF concentration of SVA contributed towards higher in-cylinder gas temperature which was recorded at 25K higher than DVA at IVC for 3.6 bar GMEP and 30K higher for 7.9 bar GMEP and 9.8 bar GMEP. C_p was found to be only less than 1% higher for SVA. That result suggests that C_p contribution towards the difference can be marginalised, however to gain an extra level of confidence extreme case of 40% RGF at 700K was simulated and compared against 5% RGF. C_p only increased by 3% for the aforementioned 8x increase in RGF, hence its contribution was concluded to be negligible.

Friction velocity (u_τ), as shown above in equation 3.5 is directly proportional to $u - u_w$ which represents the difference between tangential gas velocity next to the walls and tangential wall velocity. In this scenario u_w is typically equal to zero for all surfaces.

As it had already been presented earlier (Figure 3.15) temperature varies between DVA and SVA valvetrain operation, hence should be investigated further.

Given the shape of heat transfer plots (Figures 3.16, 3.27 and 3.38), and the fact that IVC occurred at 90 °CA bTDC_f, so-called “closed cycle” simulations were designed to decouple temperature and flow field impact on the total heat transfer. “Closed cycle” refers to part of the compression stroke past IVC, when no extra air is being admitted or expelled from the cylinder, thus keeping gas composition and trapped mass constant.

“Closed cycle” simulations were undertaken to study sensitivity of the heat transfer to previously identified “main drivers” that were believed to have major effects on its magnitude. Gas temperature at IVC and flow field were the only variables, whilst the simulations were initialised with the same C_p and density (applying DVA values to all cases) and homogenous distribution of in-cylinder pressure, gas composition and temperature to effectively erase any mixing history from the full cycle simulations. It ensured that all closed cycle simulations could be compared like-for-like in terms of temperature without interference from any full cycle developed temperature field.

Table 3.2 represents 4 points designed for 1500 RPM, 3.6 bar GMEP condition to test implications of flow field and temperature on heat transfer effects. It was colour-coded (grey representing DVA, orange SVA) and it can be seen that points 1 and 3 were “pure” DVA and SVA cases, whilst point 2 simulated DVA flow field with SVA temperature and point 4 simulated SVA flow field with DVA temperature.

Table 3.2 – Representation of flow field and temperature conditions used in heat transfer sensitivity study for 1500 RPM, 3.6 bar GMEP

Run N°	1	2	3	4
Flow Field	DVA	DVA	SVA	SVA
T _{IVC}	387	412	412	387

All subsequent plots produced during this sensitivity study include 4 closed cycle simulation points (as outlined in the table 3.2 above) and full cycle DVA and SVA results. Full cycle data was added to ensure that conclusions drawn in this study are well matched to previous analysis of heat transfer data and confirm relevance of the closed cycle approach.

Figure 3.39 representing 1500 RPM, 3.6 bar GMEP operating condition shows that closed cycle simulations were representative of the full cycle (which includes mixing history and temperature spacial distribution) only after about 15°CA after they were being initialised and validate approach outlined earlier. It can be seen that under this speed and load, maximum heat transfer delta between DVA and SVA was approximately 500 W (23.6% increase). Delta that came from flow field effects (grey to burgundy lines) was 10.5%, whilst increased in-cylinder temperature (grey to blue) resulted in 12.5%. As a result, temperature difference could be attributed to be a major heat transfer driver between DVA and SVA for 1500 RPM, 3.6 bar case. As per earlier discussion that initial temperature difference was a result of higher residuals content for SVA.

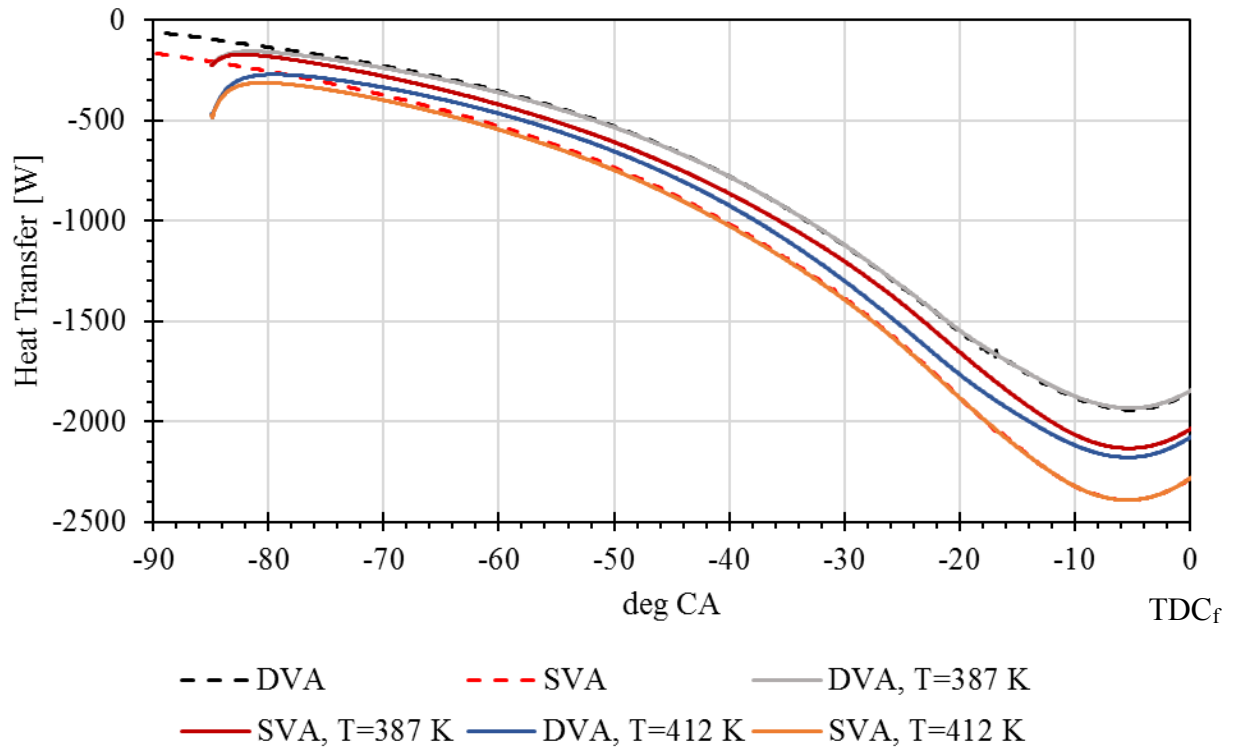


Figure 3.39 – Total heat transfer at 1500 RPM, 3.6 bar GMEP, showing the effect of applying equivalent temperature and equivalent flow field

Same methodology was applied to 1500 RPM, 7.9 bar GMEP running condition. Table 3.3 outlines temperatures being used for closed cycle simulation set-up.

Table 3.3 - – Representation of flow field and temperature conditions used in heat transfer sensitivity study for 1500 RPM, 7.9 bar GMEP

Run N ^o	1	2	3	4
Flow Field	DVA	DVA	SVA	SVA
T _{IVC}	379	404	404	379

Closed cycle simulations were again well matched with their full cycle equivalents, with only minor discrepancy (about 1.5%) for “pure” SVA scenario within the last 20°CA. The maximum heat transfer delta between DVA and SVA was 50% (grey to orange). Fixed flow field, temperature increase of 25K resulted in 13.5% increase in heat transfer, whilst the effect of variable flow field at a constant temperature (379K) was quantified to contribute 34% heat

transfer delta. Contrary to the lower load, flow field effects had significantly higher contribution compared to the temperature (approximately 2.5 times the temperature effects).

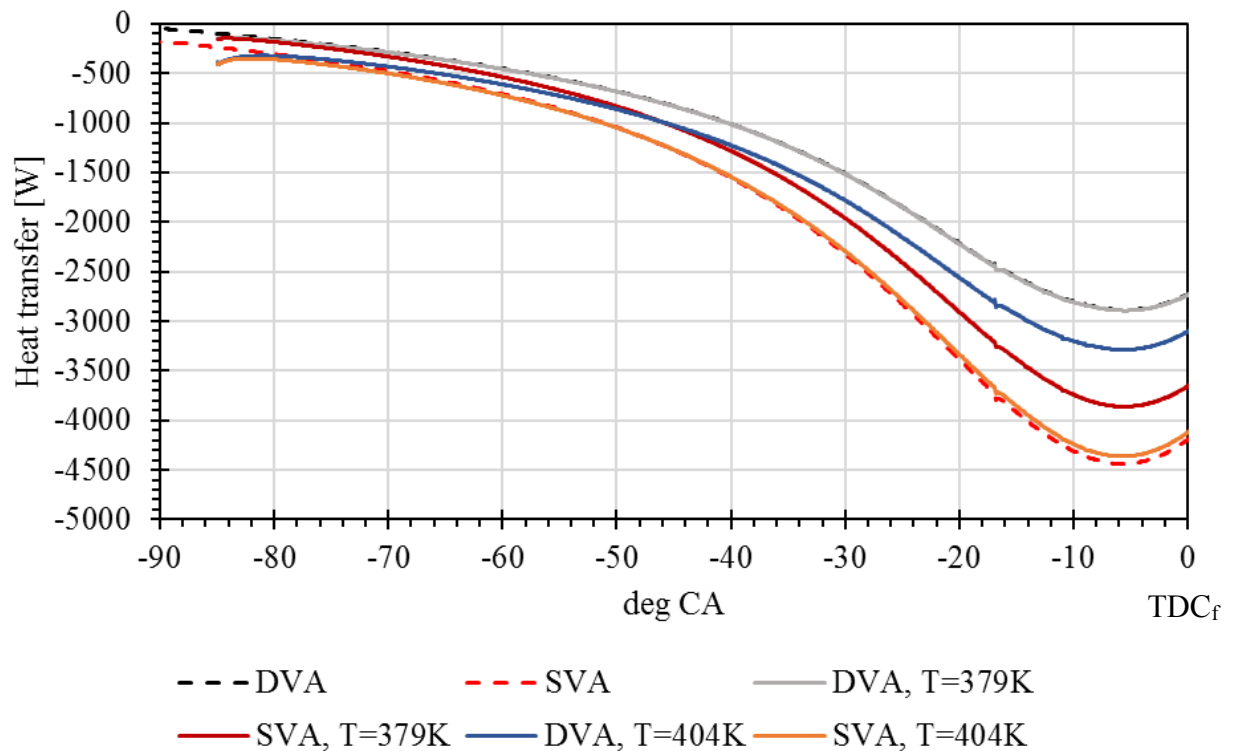


Figure 3.40 - Total heat transfer at 1500 RPM, 7.9 bar GMEP, showing the effect of applying equivalent temperature and equivalent flow field

Lastly the 2000 RPM, 9.8 bar GMEP point was subjected to the same heat transfer sensitivity analysis. Temperature difference between DVA and SVA at IVC was 5K higher than for both 1500 RPM cases.

Table 3.4 – Representation of flow field and temperature conditions used in heat transfer sensitivity study for 2000 RPM, 9.8 bar GMEP

Run N°	1	2	3	4
Flow Field	DVA	DVA	SVA	SVA
T _{IVC}	384	414	414	384

Heat transfer delta between DVA and SVA was recorded at almost 3000W, a 56% increase for SVA scenario. Once the flow field remained constant, but temperature was increased by 30K, heat transfer rose by 16%. The change in flow field for equivalent temperatures (384K) resulted in heat transfer increase of 36%. Same as for the 1500 RPM, 7.9 bar GMEP, flow field was found to be main driver of the heat transfer delta between DVA and SVA, contributing more than twice as much as temperature effects.

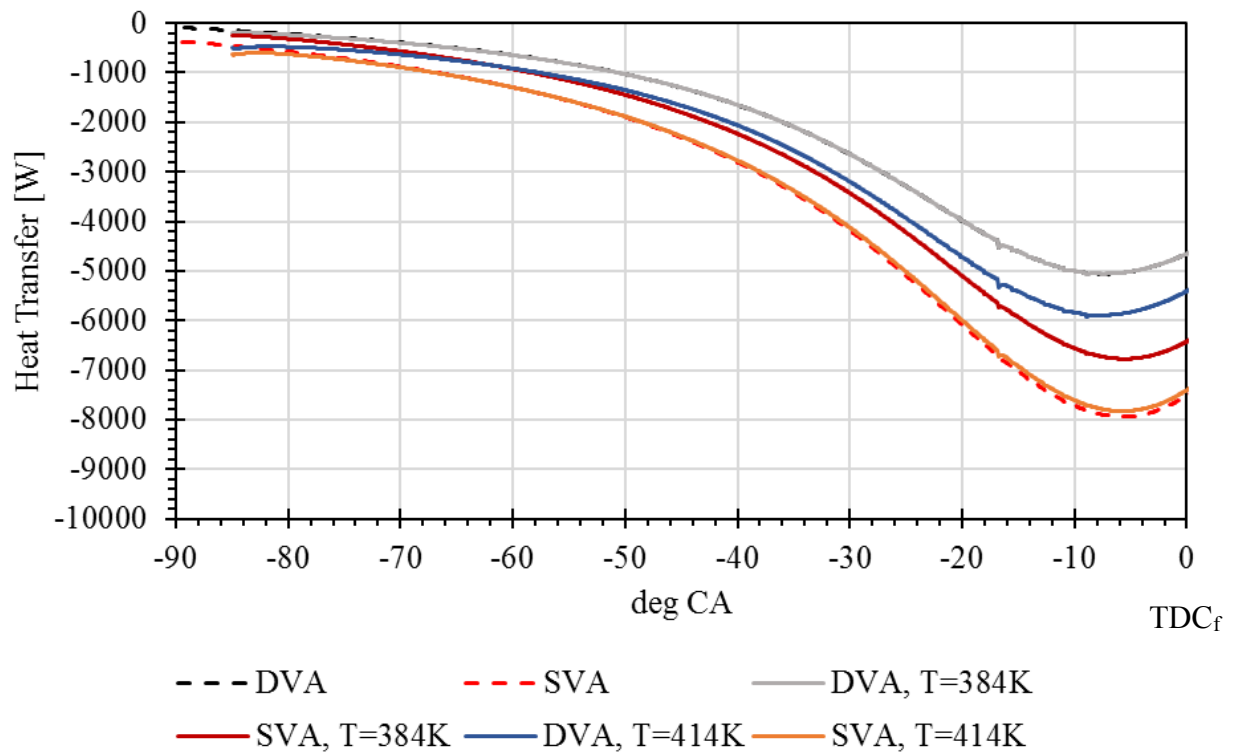


Figure 3.41 - Total heat transfer at 2000 RPM, 9.8 bar GMEP, showing the effect of applying equivalent temperature and equivalent flow field

Based on the heat transfer sensitivity analysis performed for all three points of this study, it can be concluded that at low engine loads dominant factor for increased heat transfer is higher in-cylinder temperature of SVA (caused by increased residuals content), whilst for the higher loads, flow field effects i.e. higher gas velocity close to the engine walls become the prevailing mechanism of increasing SVA's heat transfer. Finally, it should be noted that while the flow field effects varied significantly across 3 examined points (11-36%), temperature effects were fairly consistent for all 3 loads (12.5-16%), especially for both 1500 RPM points where the difference was recorded at 12.5 and 13.5%.

3.4. Emissions

Given the current regulations within the automotive industry and unprecedented level of efforts being put into lowering emissions, with especially heavy focus on NO_x , no study can be completed without analysis of emissions formation.

Tailpipe emissions were measured using Horiba MEXA ONE (details on measurement principles for all species can be found in Chapter 2).

In addition to aforementioned NO_x which plays major role, CO, CO_2 and total unburned hydrocarbons (THC) were studied.

3.4.1. 1500 RPM, 3.6 bar GMEP

First experimental point of 3.6 bar GMEP, 1500 RPM exhibited significant differences in CO production under stoichiometric conditions. DVA condition produced 20-25% more CO for all EGR cases (0-20%), with a trend of small reduction with increasing EGR dilution. Once the excess air dilution was introduced, CO levels fell by 80-90% (compared to stoichiometric values) and were the same for DVA and SVA, also showing very little sensitivity to either EGR dilution or air/fuel equivalence ratio (shown as single colour turquoise area in Figure 3.42).

CO_2 production was identical for both DVA and SVA, and only showed sensitivity towards excess air dilution represented by horizontal isolines in Figure 3.42.

Trends for both CO and CO_2 formation were in-line with expectations set for lean mixtures based on the emissions formation presented by Heywood (1988). High availability of oxygen within the mixture ensured complete burn of carbon elements of the fuel to CO_2 , thus insignificant levels of CO when operating lean. CO_2 concentration, measured as percentage of analysed exhaust gas, decreased with increased air/fuel equivalence ratio due to more inert gases passing through the chamber and not taking place in the combustion process.

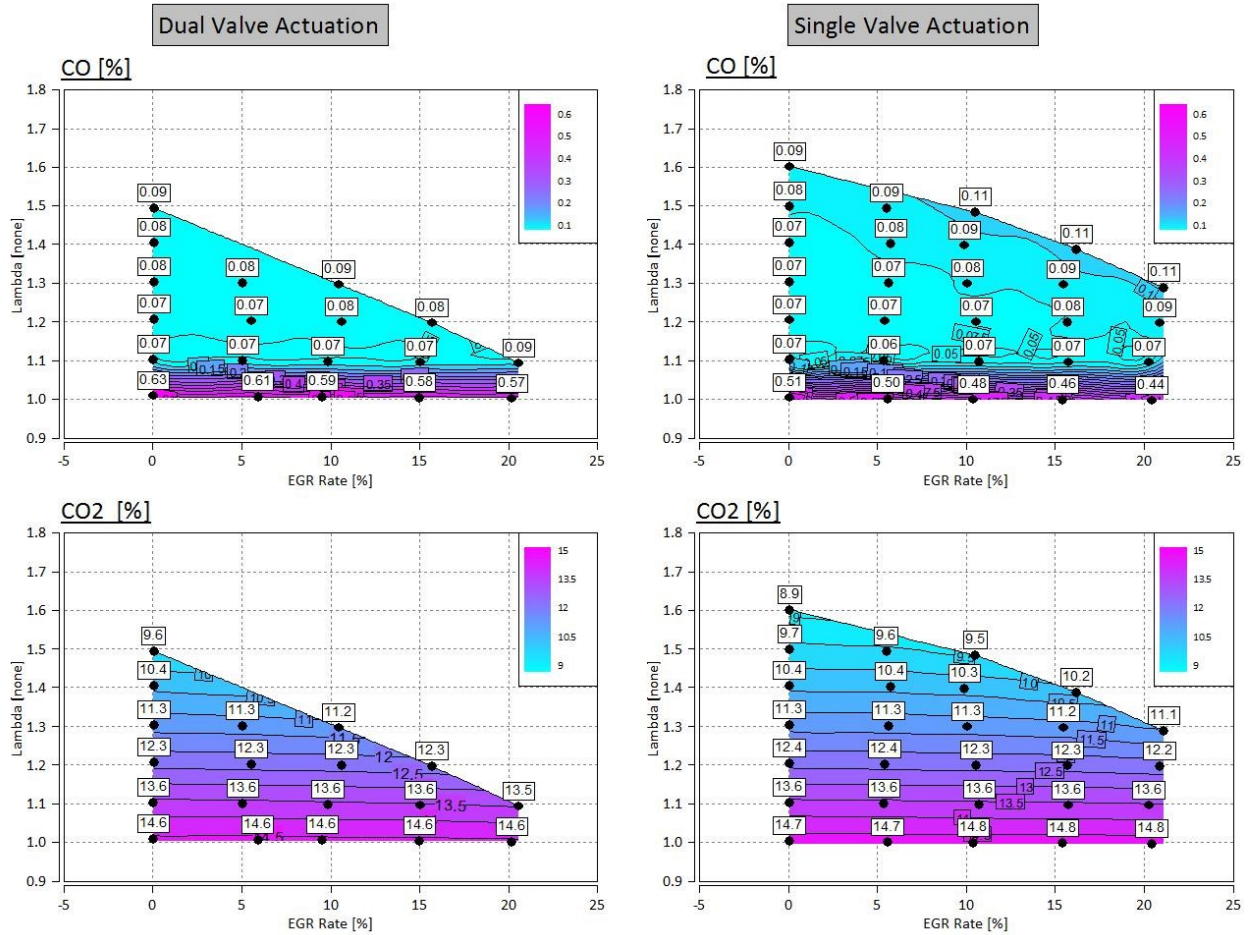


Figure 3.42 – CO and CO₂ emissions comparison for DVA and SVA at 1500 RPM, 3.6 bar GMEP

Total hydrocarbons (THC) were plotted and it can be seen in Figure 3.43 that SVA generally achieved values lower than DVA, with exception of stoichiometric line and 0% EGR, lambda 1.1 condition, when comparing equivalent dilution levels. THC emissions can have several different sources, however they are all linked with mixture formation within the cylinder and flame propagation during the combustion process, being quite sensitive to local air/fuel equivalence ratio within the cylinder. These are discussed in greater detail later in this chapter, together with mixture preparation.

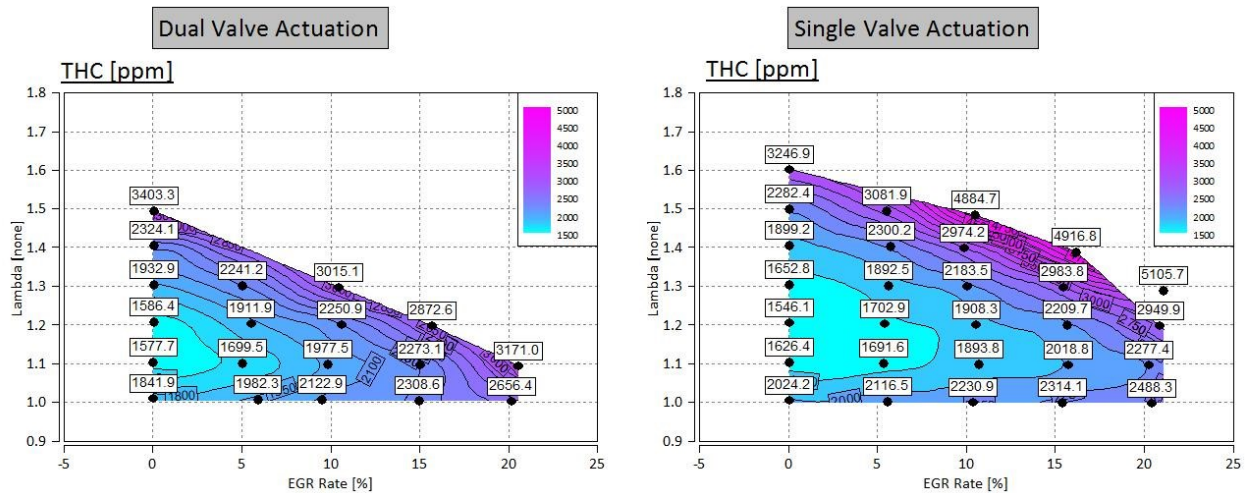


Figure 3.43 – Total hydrocarbons emissions comparison for DVA and SVA at 1500 RPM, 3.6 bar GMEP

Nitrogen oxides (NO_x) production was found to be in line with literature (Heywood 1988, Lumley 1999), and highest emissions of these species occurred at lambda 1.1 for each EGR rate. EGR dilution also proved to be more efficient NO_x suppressant than excess air due to its reduced in-cylinder temperatures associated with this form of charge dilution and CO_2 being much more effective in increasing heat capacity of the cylinder charge, per unit mass of fuel, than N_2 (NO_x formation being directly proportional to T^2). Contour maps presented in Figure 3.44 indicated some inconsistencies in trends between DVA and SVA, therefore new method of plotting data was introduced. Each EGR level for both DVA and SVA was plotted against lambda on the x-axis and NO_x engine out emissions on the y-axis (Figure 3.45). Revised approach provided a clear picture, showing both DVA and SVA always producing peak NO_x values at lambda 1.1 (2366 ppm and 2573 ppm respectively for $\lambda=1.0$). Interestingly, SVA's NO_x emissions were always higher than DVA's for lambda 1.0-1.1, however as the air/fuel equivalence ratio increased delta between the two decreased. Depending on the percentage of EGR dilution, DVA and SVA lines crossed, and after that intersection NO_x emissions were higher for DVA. The aforementioned intersection points appear to be correlated with EGR dilution, as it occurred at lambda 1.3 for 0% EGR, and earlier at around lambda 1.2 for 5% and 10% EGR levels. At 15% EGR level, DVA and SVA both peaked at lambda 1.1 achieving very similar NO_x values (~20 ppm difference). Lastly, the optimum points found earlier for reducing GSFC (adhering to 3% CoV of NMEP criteria) were highlighted with green in Figure 3.45. It clearly shows that SVA strategy which allows for much greater dilution was very effective in reducing NO_x levels by nearly 80% in this case.

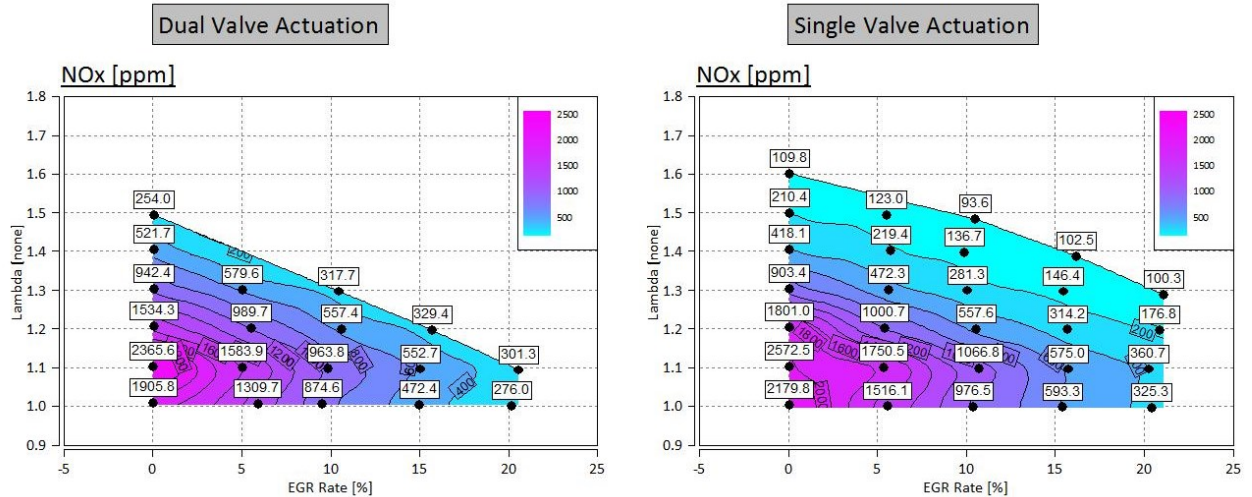


Figure 3.44 - NO_x emissions comparison for DVA and SVA at 1500 RPM, 3.6 bar GMEP

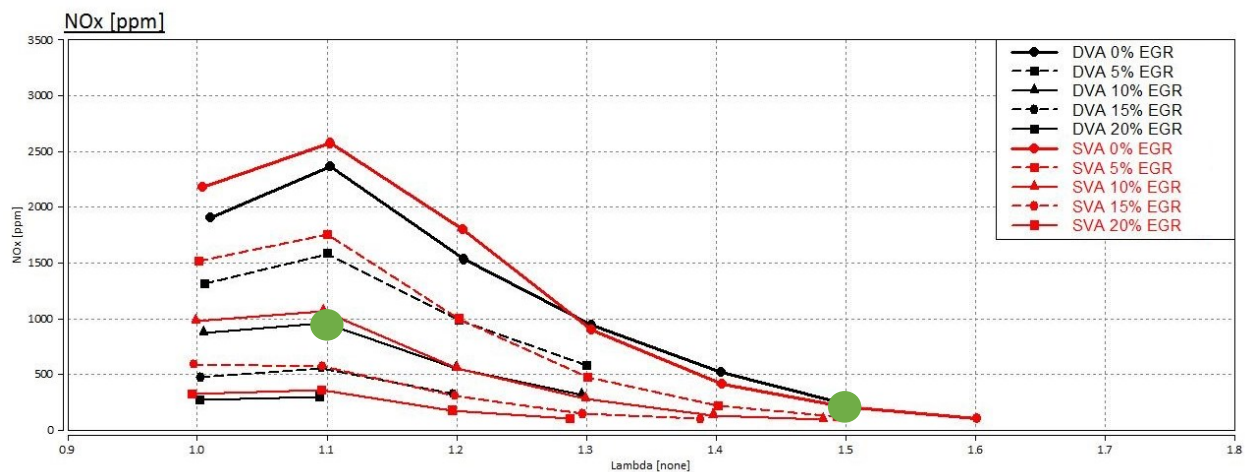


Figure 3.45 - NO_x emissions analysis by EGR percentage for DVA and SVA at 1500 RPM, 3.6 bar GMEP

3.4.2. 1500 RPM, 7.9 bar GMEP

CO and CO₂ emissions were almost identical, once the load increased to 7.9 bar GMEP. Same trends were captured, and for the completeness of analysis contour plots were produced and can be found in the Appendix section on page 287.

Unburned hydrocarbons, however exhibited different behaviour which is clearly presented in Figure 3.46. SVA produced significantly lower THC emissions for all measurement points. Big turquoise area in Figure 3.46 shows reduction of 30-35% compared to DVA points of

equivalent dilution. It can also be seen in Figure 3.47, where data was split by EGR levels, that THC emissions for all cases reached levels close to their minimum at about lambda 1.1 and remained almost constant up to the point, where combustion stability deteriorated and they increased due to partial burn cycles (possibly misfires for extreme cases) and incomplete combustion events.

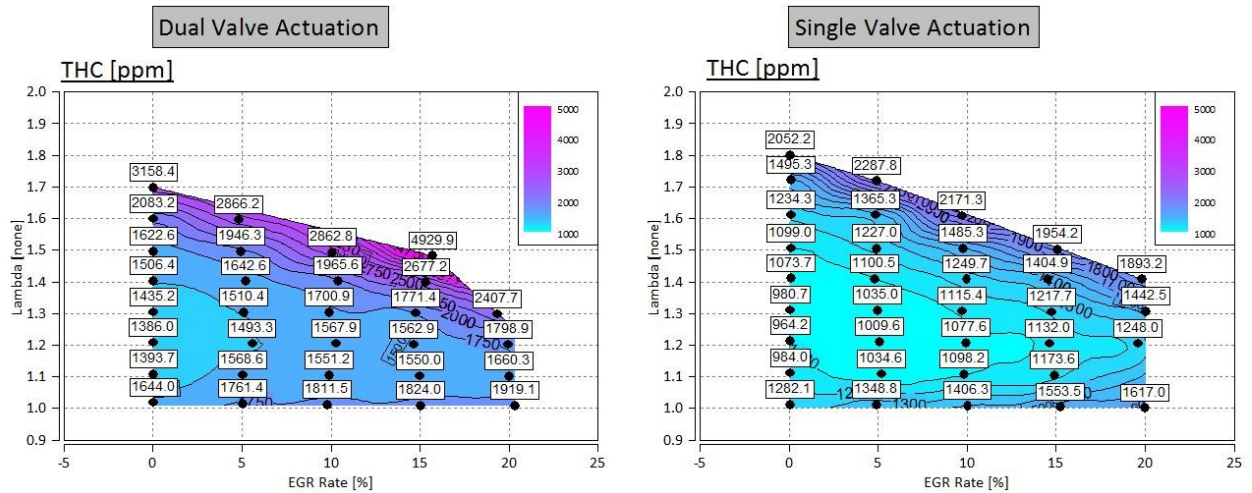


Figure 3.46 - Total hydrocarbons emissions comparison for DVA and SVA at 1500 RPM, 7.9 bar GMEP

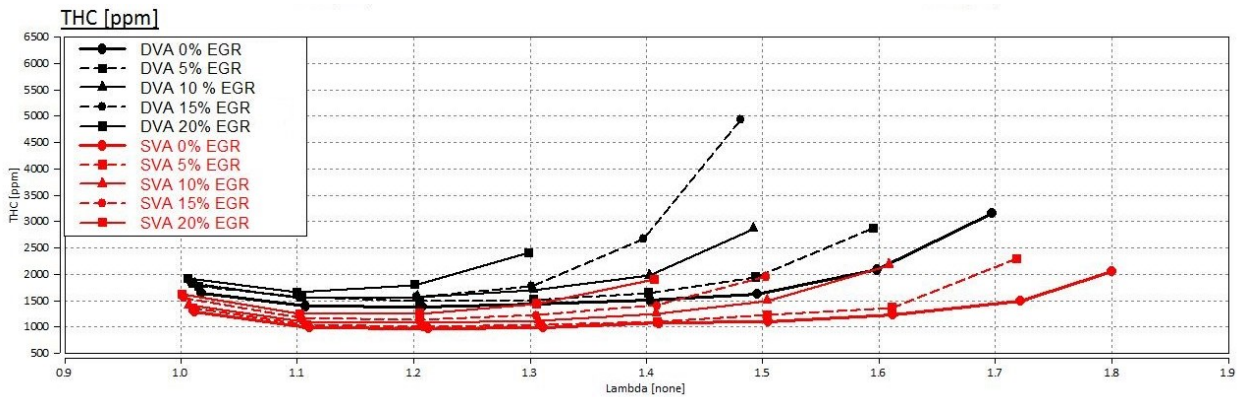


Figure 3.47 - THC emissions analysis by EGR percentage for DVA and SVA at 1500 RPM, 7.9 bar GMEP

NO_x formation exhibited trends similar to the ones observed for lower load condition, however levels of NO_x (at $\lambda=1.1$ peak) were roughly twice as high as the ones examined for 3.6 bar GMEP condition and were recorded at 4364 ppm and 5176 ppm for DVA and SVA respectively. An increase can be attributed to higher in-cylinder temperatures and pressure resulting from load increase. Intersection points between DVA and SVA for the same EGR

levels shifted to higher lambda values i.e. 1.45 vs 1.3 lambda for 0% EGR. Comparison of NO_x for GSFC optimised points showed reduction of approximately 85% when applying SVA strategy.

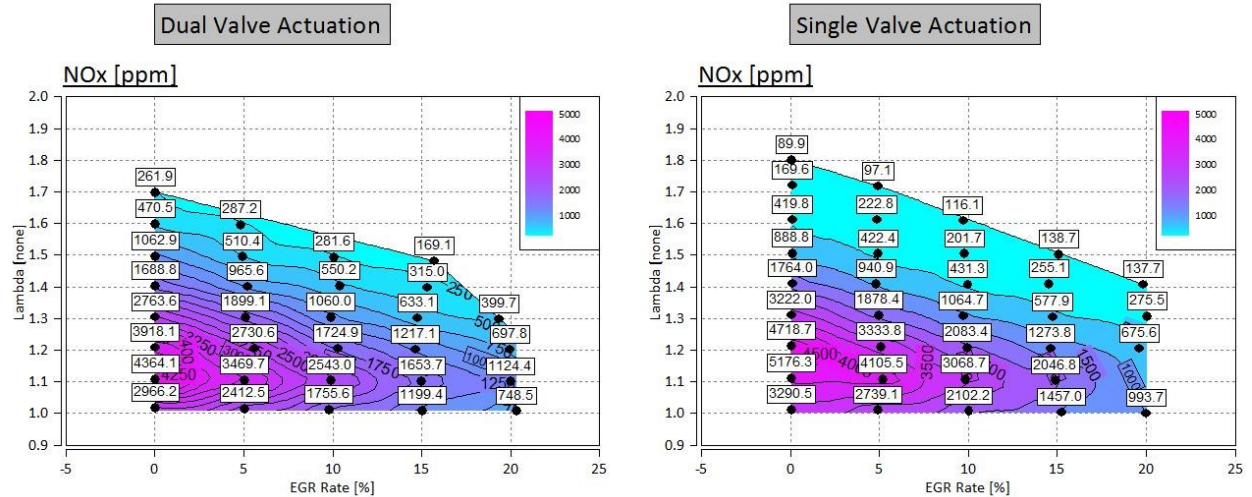


Figure 3.48 - NO_x emissions comparison for DVA and SVA at 1500 RPM, 7.9 bar GMEP

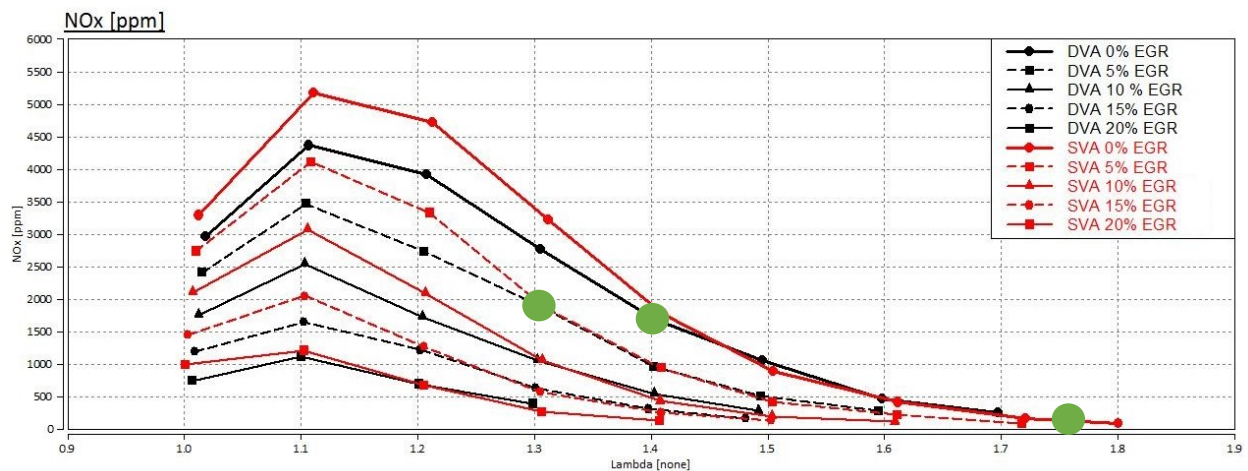


Figure 3.49 - NO_x emissions analysis by EGR percentage for DVA and SVA at 1500 RPM, 7.9 bar GMEP

3.4.3. 2000 RPM, 9.8 bar GMEP

Lastly emissions analysis was performed for 2000 RPM, 9.8 bar GMEP running condition. As per two previously analysed points, CO and CO₂ production followed the same trends described earlier (plots can be found in Appendix on page 287). Unburned hydrocarbons were

much lower for both DVA and SVA (compared to 1500 RPM low and medium load cases), with the same general behaviour. As presented in Figure 3.51, SVA widened the gap by which it outperforms DVA, now set at approximately 40%.

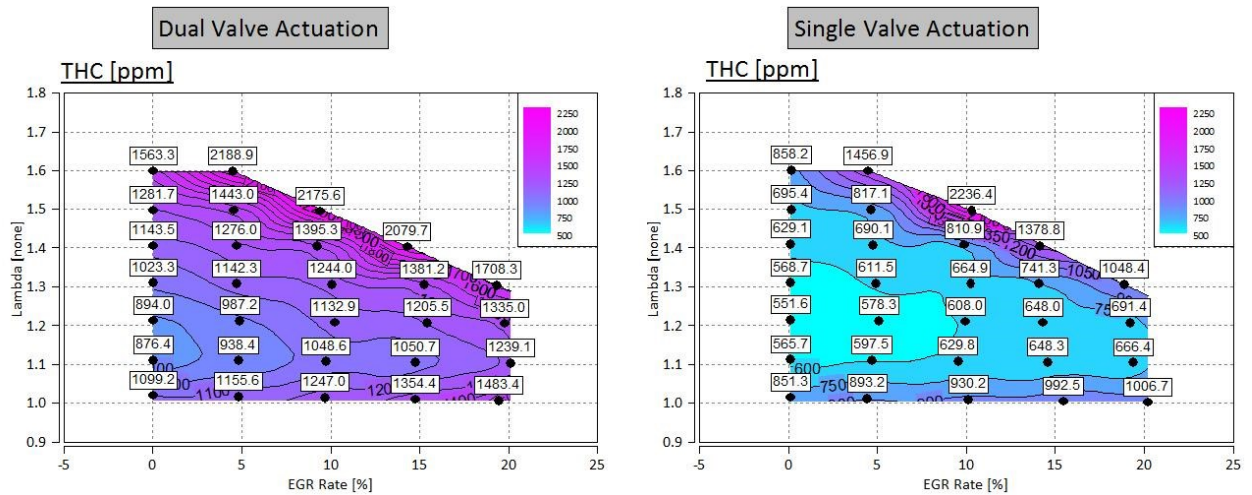


Figure 3.50 - Total hydrocarbons emissions comparison for DVA and SVA at 2000 RPM, 9.8 bar GMEP

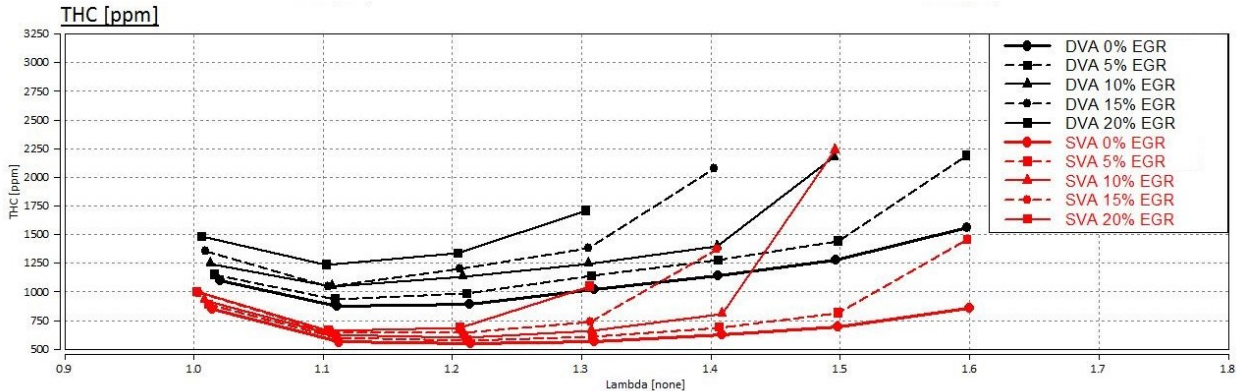


Figure 3.51 - THC emissions analysis by EGR percentage for DVA and SVA at 2000 RPM, 9.8 bar GMEP

NO_x behaviour was again similar to earlier analysed cases and in-line with expectations set by literature (Heywood 1988, Lumley 1999). Interestingly, there were no intersections between DVA and SVA lines of any given EGR dilution. Also, NO_x reduction for GSFC optimised point was far smaller at only 50%, which can be attributed to much smaller gap in air/fuel equivalence ratio between DVA and SVA points optimised for fuel consumption.

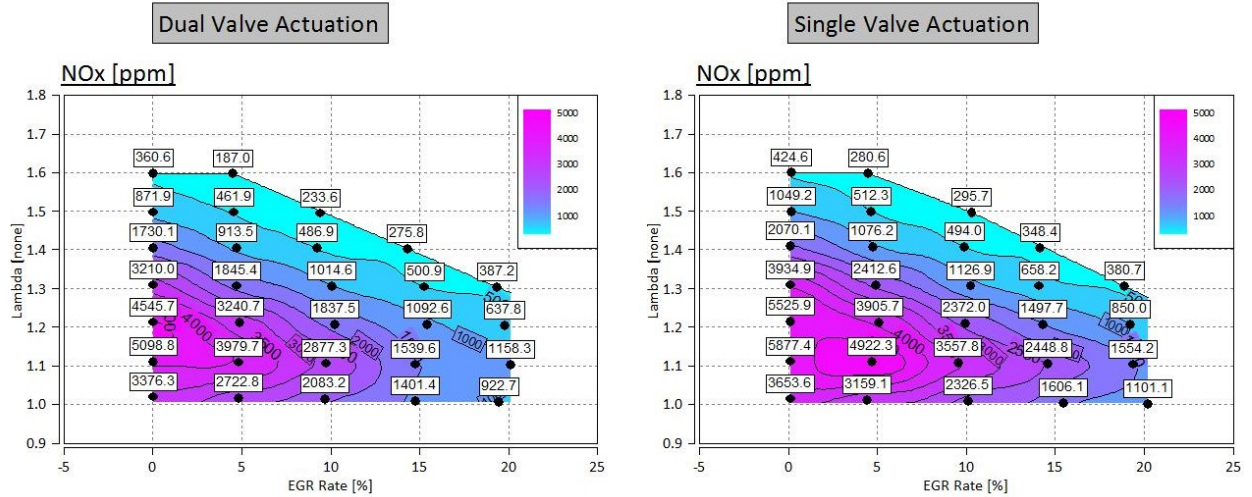


Figure 3.52 - NO_x emissions comparison for DVA and SVA at 2000 RPM, 9.8 bar GMEP

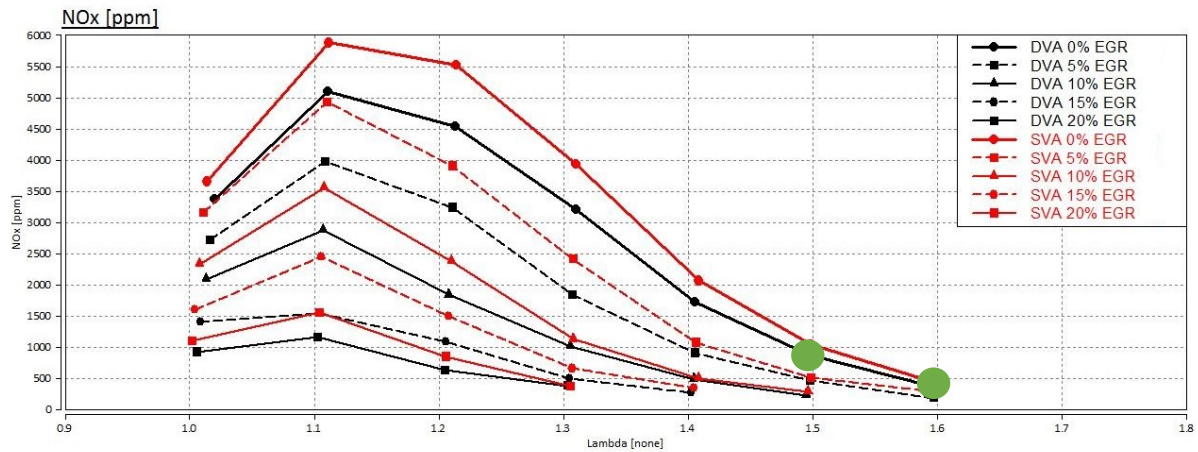


Figure 3.53 - NO_x emissions analysis by EGR percentage for DVA and SVA at 2000 RPM, 9.8 bar GMEP

3.4.4. Mixture Preparation Effects

Emissions formation, especially NO_x trends and intersections seen between DVA and SVA lines, prompted investigation into the mixture preparation within the cylinder close to the combustion event. Insights into the mixture preparation which affects species formation required fuel injection to be modelled in addition to air and residuals presented in the previous sections. It was dictated by the paramount importance of air/fuel equivalence ratio in the process of NO_x and THC production. The Lagrangian multi-phase treatment was used to model the fuel injection, details of which can be found in Kountouriotis (2019). It should be noted

that the aforementioned Lagrangian approach included turbulent dispersion, gravity effects and inter-droplet collisions based on the O'Rourke (1981) model and the Reitz and Diwakar (1986) model was used for the droplet break-up. Start of injection was matched to the experimental value of $300^{\circ}\text{CA bTDC}_f$ (details of why this injection angle was used can be found in Chapter 2). Single-component fuel represented by *iso*-Octane was used. Further details and validation studies performed to ensure fuel spray modelling accurately represents experimental conditions can be found in Kountouriotis (2019).

1500 RPM, 7.9 bar GMEP was used as “base” case in the mixture preparation analysis due to more pronounced THC differences between DVA and SVA (compared to 3.6 bar GMEP) and presence of NO_x intersections of DVA and SVA plots for a given EGR level (as opposed to 2000 RPM condition).

Firstly, the spacial distribution of air-fuel mixture was presented for both DVA (Figure 3.54) and SVA (Figure 3.55). It immediately stands out that, regardless of the same global lambda for both cases, employing DVA strategy resulted in much higher stratification within the cylinder close to TDC_f . Fuel rich region (blue colour) was formed on the exhaust side, whilst lean region (red) occupied intake side during the end of compression stroke. Spark plug vicinity had lambda values close to 1.0, however “lean pocket” was formed on the exhaust side of the spark plug at ignition ($15^{\circ}\text{CA bTDC}_f$).

SVA strategy presented vastly different behaviour. Almost single-coloured images provide evidence that the mixture was very homogenous throughout the cylinder, without any rich or lean pockets. Only very small portion of the chamber, close to the intake valves, exhibited slightly leaner air-fuel mixture at ignition ($10^{\circ}\text{CA bTDC}_f$).

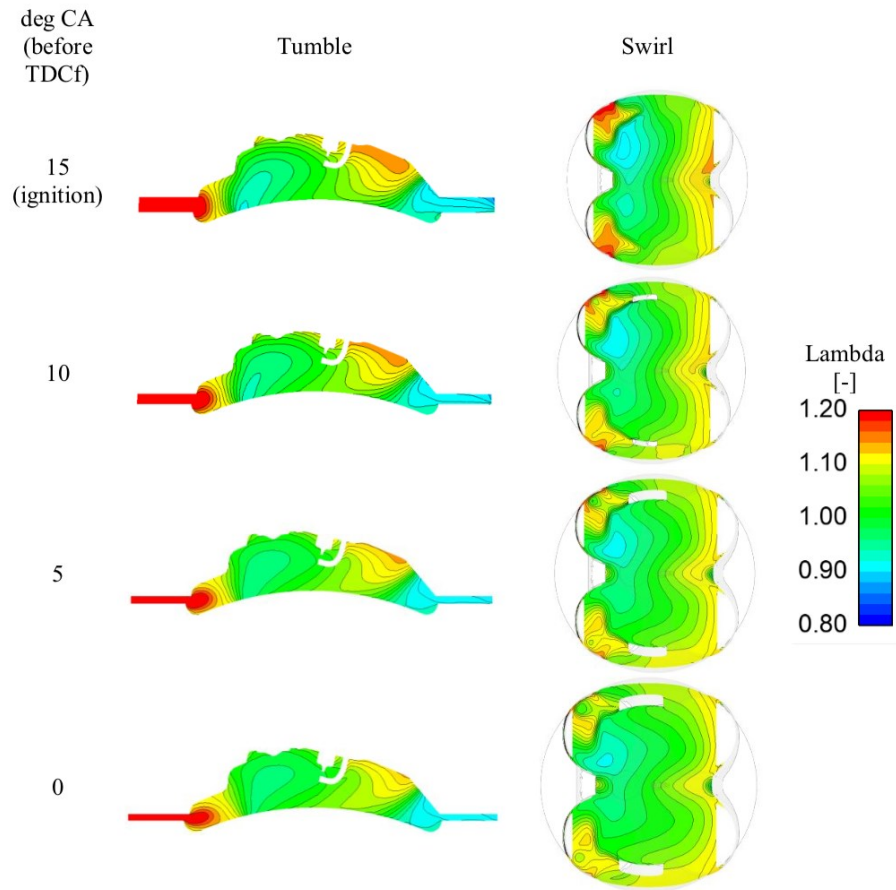


Figure 3.54 – Spatial distribution of in-cylinder lambda for DVA between 15°CA bTDCf and TDC, $\lambda=1.0$, 1500 RPM, 7.9 bar GMEP (Kountouriotis, 2019)

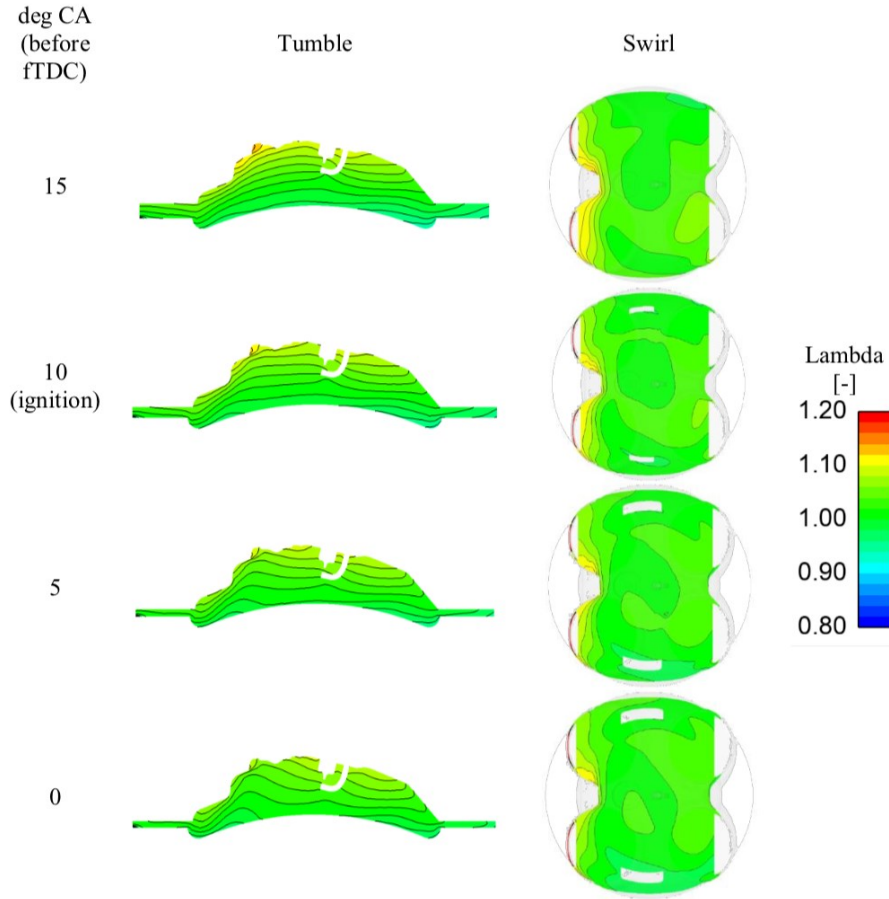


Figure 3.55 - Spatial distribution of in-cylinder lambda for SVA between 15°CA bTDCf and TDC, $\lambda=1.0$, 1500 RPM, 7.9 bar GMEP (Kountouriotis, 2019)

An effort was made to quantify the observed homogeneity of the mixture, and Homogeneity Index (HI) was introduced in the following form:

$$HI = 1 - CoV_{\lambda} = 1 - \frac{\sigma_{\lambda}}{\bar{\lambda}}, \text{ where} \quad (3.7)$$

σ_{λ} – standard deviation of lambda values

$\bar{\lambda}$ – average lambda

Once plotted (Figure 3.56 representing last 60°CA within the compression stroke up to 20°CA after TDC_f), it can be clearly seen that homogeneity of SVA was always higher than DVA

which is in good correlation with observations made earlier. Homogeneity index values also constantly increased for both strategies throughout the cycle, and given that SVA strategy always required smaller ignition angle advance due to faster burn (for a given dilution condition), the difference between the two valvetrain operating modes increased even further. They were quantified as 0.97 and 0.94 at ignition for SVA and DVA respectively.

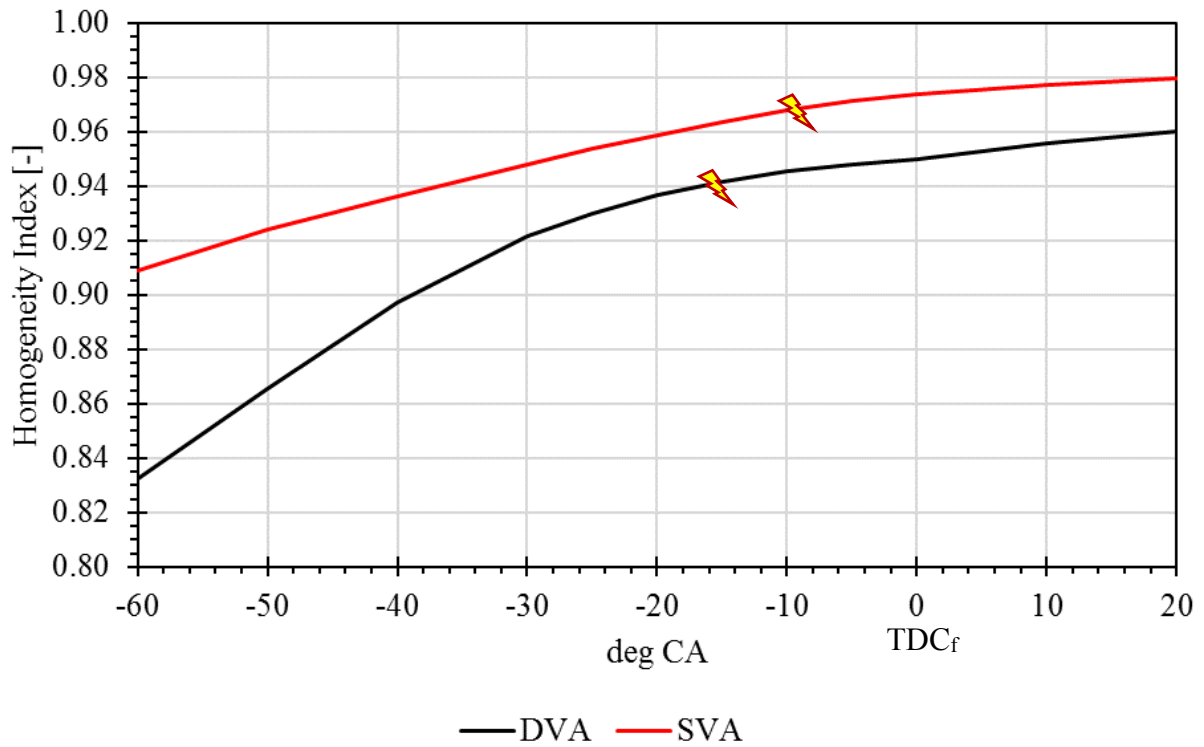


Figure 3.56 – Homogeneity Index comparison between DVA and SVA for $\lambda=1.0$ at 1500 RPM, 7.9 bar GMEP

Next parameter investigated in the emissions context was the fuel film mass forming on piston and liner surfaces. It should be noted that injector nozzles and spray pattern were designed to primarily avoid liner wetting which would have consequences in liquid fuel mixing with the oil film. Such mixing would have detrimental effect on the fuel evaporation (Zhang *et al.*, 2018). It is evident (Figure 3.57) that the design criteria were met, as liner wetting was marginal when compared to piston surface. Film mass on the piston varied significantly between DVA and SVA strategies. It is clearly shown that adoption of valve deactivation significantly reduced the maximum accumulation (0.48 mg vs 0.81 mg), which also happened about 5 °CA earlier within the cycle and fuel evaporated much quicker. Rate of evaporation was similar, however much higher accumulation in the wetting process required longer time (delay of approximately

20 °CA). Such behaviour was caused by the SVA, having stronger in-cylinder flow, being more effective in disintegrating spray plumes.

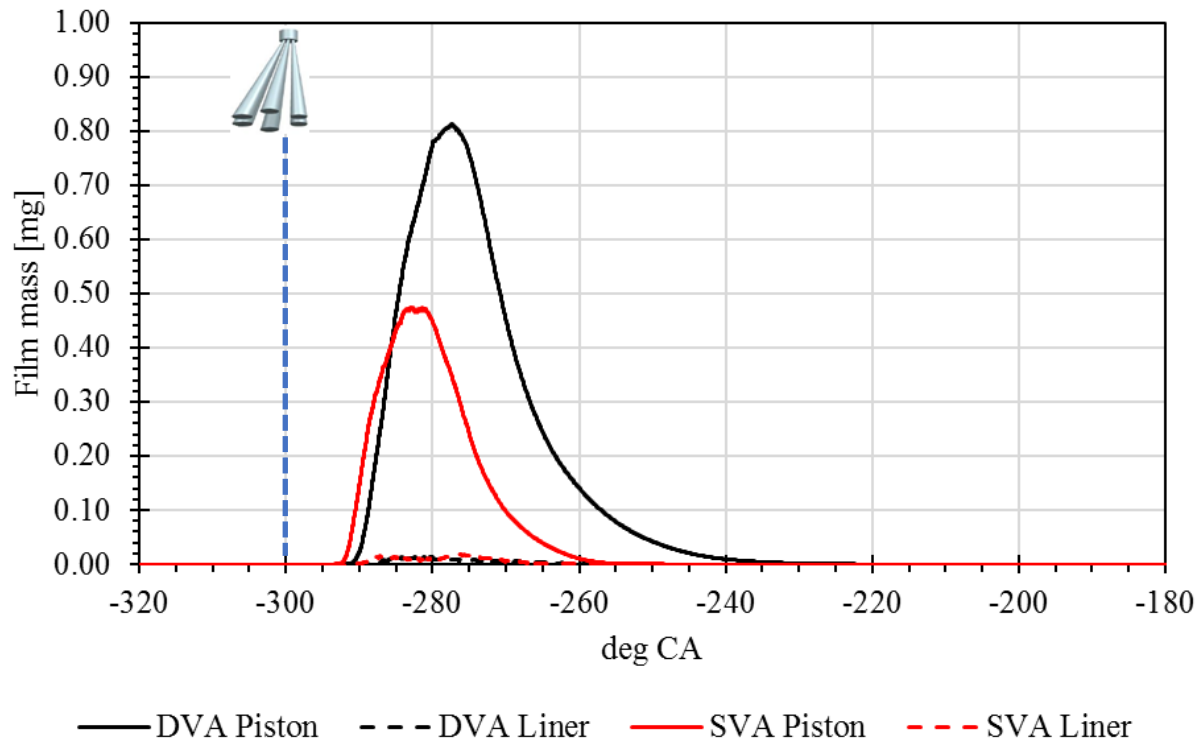


Figure 3.57 – Accumulated liquid fuel film mass on piston surface and cylinder liner comparison between DVA and SVA for $\lambda=1.0$ at 1500 RPM, 7.9 bar GMEP

Spatial distribution of liquid film was also investigated around its peak (snapshot taken at 280 °CA bTDC_f) and it showed two very different patterns, presented in Figure 3.58 (Kountouriotis, 2018). As expected, DVA produced symmetric pattern which can be attributed to symmetric design of the injector and tumble motion acting only along a single axis. SVA pattern did not follow any symmetry, as it was subjected to combination of both swirl and tumble. Important observation was also the fact that under DVA operation, fuel film formed within the exhaust valve cut-outs on the piston surface. Numerical simulations also showed full fuel evaporation (past 220°CA bTDC_f), however in real engine with some deposit built-ups on the piston surface, fuel trapped in such notches might not fully evaporate and contribute towards higher THC production, as deposits are known to absorb HC (Heywood, 1988). Also, single component iso-octane was used in CFD estimation, whilst SCRE was running on EU6 fuel that contained heavier components. According to the fuel certificate of analysis, provided by the

manufacturer, Final Boiling Point (F.B.Pt) was measured at 180.3° which is 81°C higher than a standard F.B.Pt of isooctane (99.3°C), thus Figure 3.57 significantly overestimates point of complete evaporation of the fuel film. Therefore, increased fuel impingement on the piston surface, together with lower homogeneity of the mixture, with clearly visible rich and lean regions within the cylinder at ignition, are believed to be main factors contributing towards higher concentration of measured unburned hydrocarbons under DVA operation. It is acknowledged that different injector configurations under SVA operation could offset this effect.

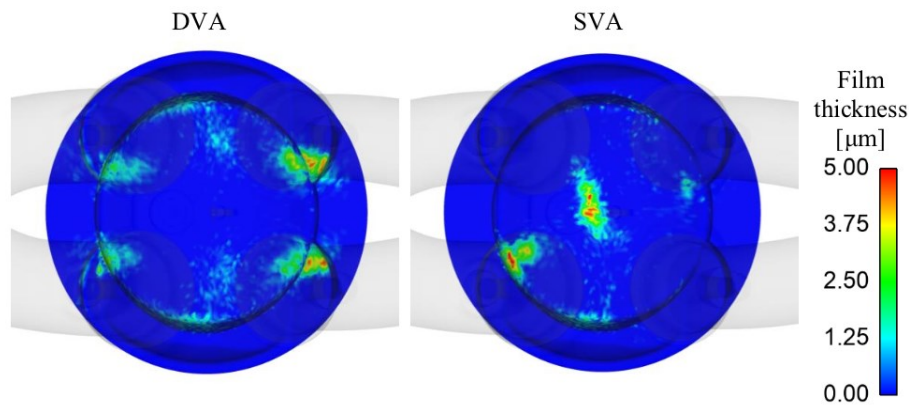


Figure 3.58 – Liquid fuel film thickness at the piston surface for DVA and SVA at 20°C_A after SOI, $\lambda=1.0$, 1500 RPM, 7.9 bar GMEP (Taken from Kountouriotis, 2018)

Lastly, to provide better understanding of the emissions formation (mainly NO_x), lambda distribution within the cylinder was studied for artificially introduced spheres centred around the spark plug gap. They ranged between 10 mm and 60 mm and were created in 10 mm increments, as presented in Figure 3.59. Only cylinder domain of each sphere was taken into account for data generation.

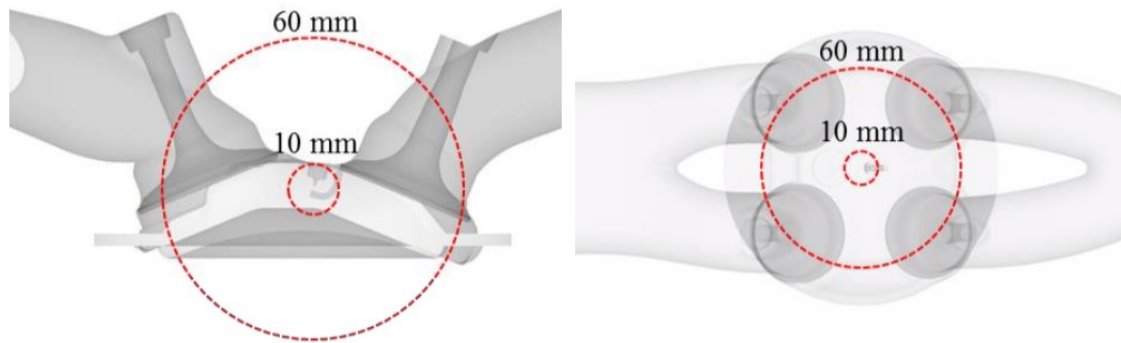


Figure 3.59 – Representation of the smallest and largest spheres centred around the spark plug gap.

NO_x formation behaviour observed for both DVA and SVA agreed with trends outlined in the literature, however providing explanation for quantitative differences between the two valvetrain operating strategies required further analysis of mixture preparation. Based on the literature and experimental findings, it was assumed that highest NO_x production occurs around the $\lambda=1.1$ region. Based on this assumption, a region of lambda 1.05 to 1.15, considered to be the worst-case scenario for NO_x formation was scrutinised for three different global lambda cases: $\lambda=1.0$, $\lambda=1.1$ and $\lambda=1.2$. Knowing that NO_x formation mostly takes place in the post-flame gases, analysis was extended to 20°CA aTDC_f, and percentage of the selected spheres' volume taken by the aforementioned lambda range [1.05,1.15] was plotted.

At lambda 1.0, DVA (Figure 3.60) right after ignition, the 10mm sphere contained 30% of the region considered to be the worst for NO_x formation, however it dropped at a high rate ($\sim 4\%/^\circ\text{CA}$) for the next 5°CA . At ignition timing, SVA (Figure 3.61) had 46% of the mixture within 10mm sphere in that region, followed by much lower reduction for the next 5°CA ($\sim 0.5\%/^\circ\text{CA}$). As the sphere diameters increased, the percentage of volume being within the [1.05,1.15] lambda region decreased, and it can be attributed to the fact that spheres capture bigger volume of the cylinder for which global lambda is 1.0. It should be noted that for all spheres centred around the spark plug, volume adhering to the earlier defined criteria was always higher for SVA, which is in line with higher NO_x production seen in experimental data. Only exception was the total cylinder values past ignition, however it can be attributed to lower homogeneity of DVA and presence of both lean and rich regions observed earlier. These regions were close to the walls, thus not captured by the spheres.

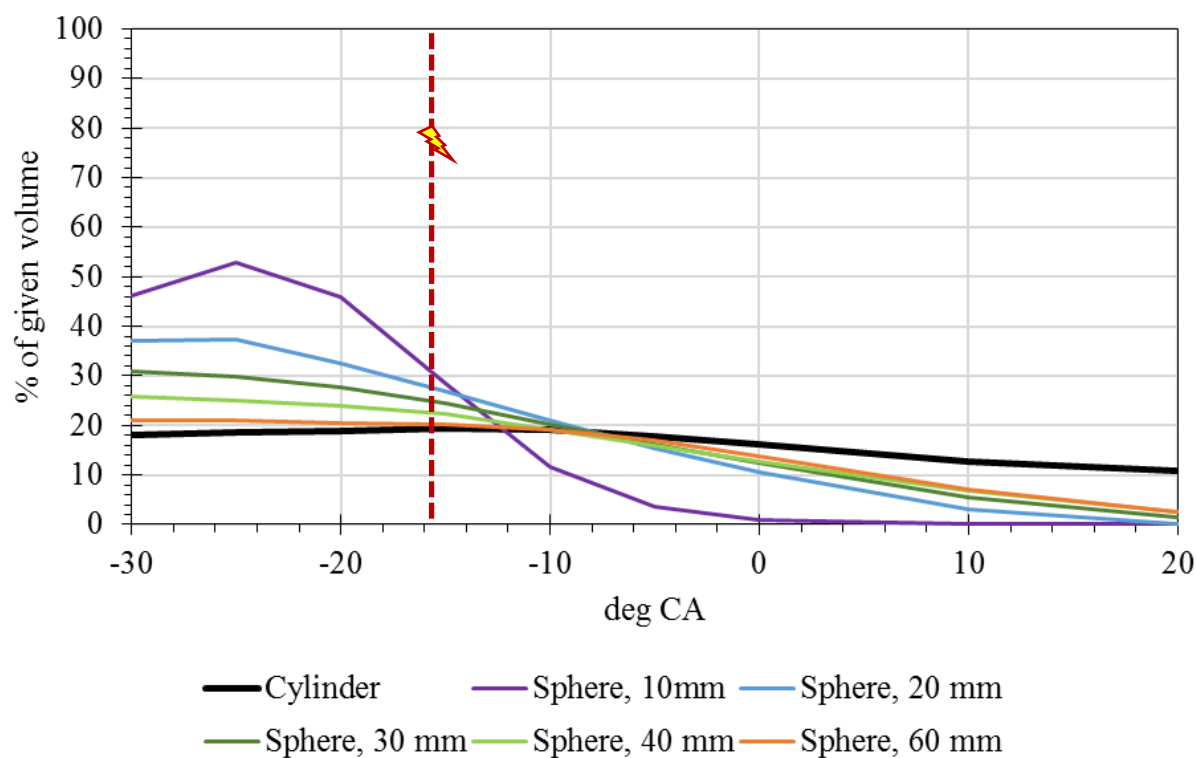


Figure 3.60 - Percentage of the cylinder and specific spherical volumes occupied by a mixture in the λ range of [1.05, 1.15] for DVA at global $\lambda=1.0$, 7.9 bar GMEP, 1500RPM

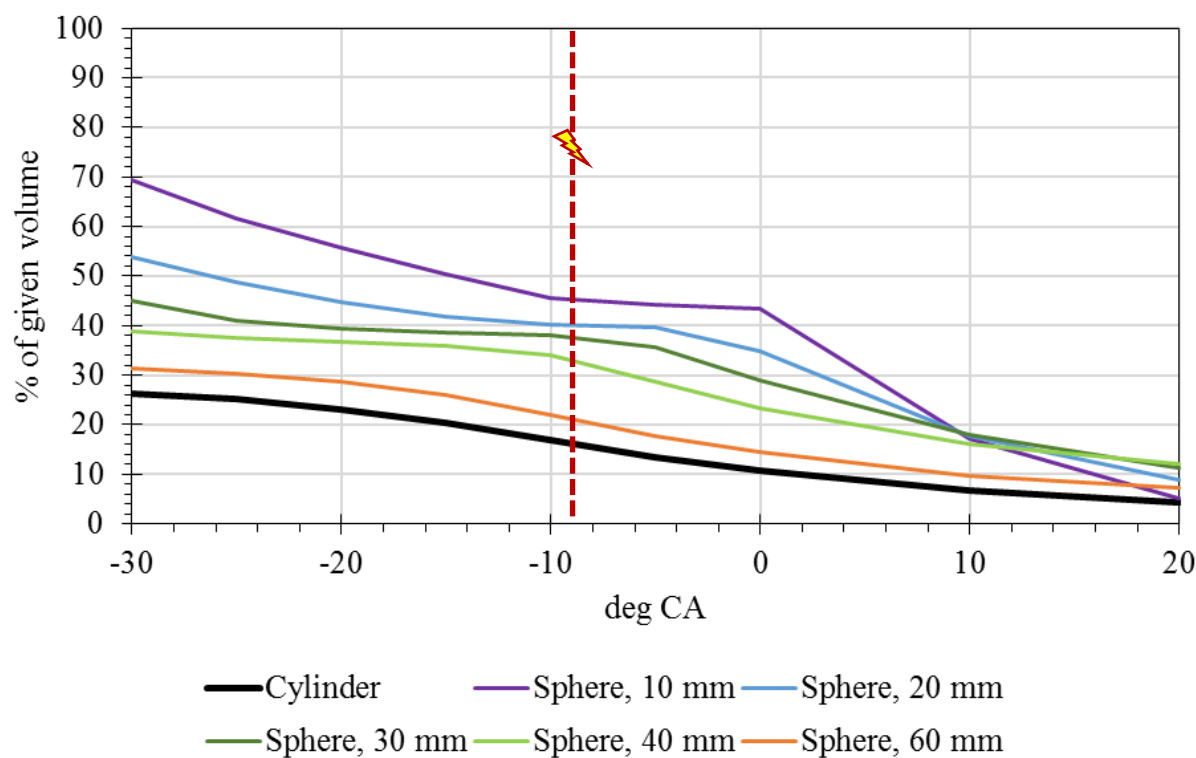


Figure 3.61 - Percentage of the cylinder and specific spherical volumes occupied by a mixture in the λ range of [1.05, 1.15] for SVA at global $\lambda=1.0$, 7.9 bar GMEP, 1500RPM

Increasing global lambda to 1.1, exhibited trends showing that much higher percentage of the respective volumes were within the high NO_x production region for both DVA and SVA (Figures 3.62 and 3.63). Past ignition point, spheres of sizes 10-40 mm had 50-60% of their volumes within the assumed high NO_x production range when operated with DVA strategy, whilst same sphere sizes operated with SVA had 90-98%. That difference was much more pronounced than at lambda 1.0 condition, and it appeared to be in good agreement with experimental results that showed larger delta in NO_x concentration, once dilution was increased. Higher homogeneity of SVA was again clearly visible, as all the spheres and total in-cylinder percentage taken by the [1.05,1.15] lambda region converged to ~95%-100%, whilst DVA data had more spread and lower values throughout the analysed part of the cycle.

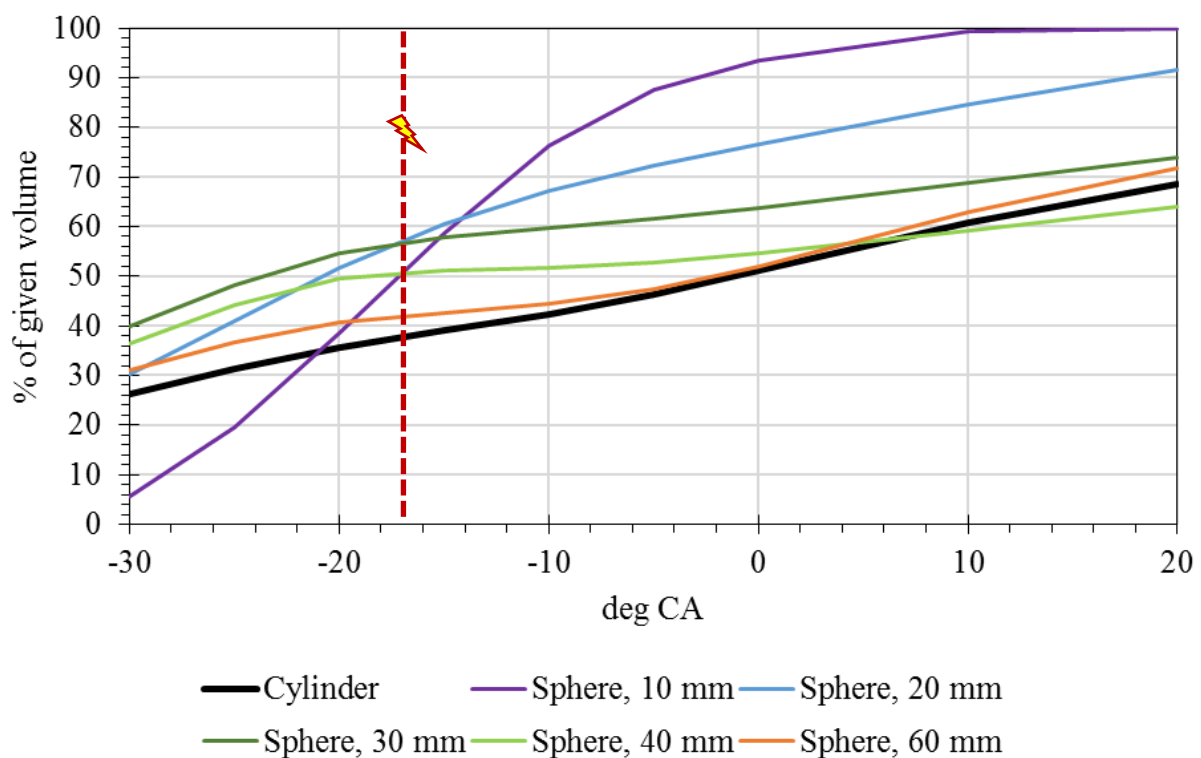


Figure 3.62 - Percentage of the cylinder and specific spherical volumes occupied by a mixture in the λ range of [1.05, 1.15] for DVA at global $\lambda=1.1$, 7.9 bar GMEP, 1500RPM

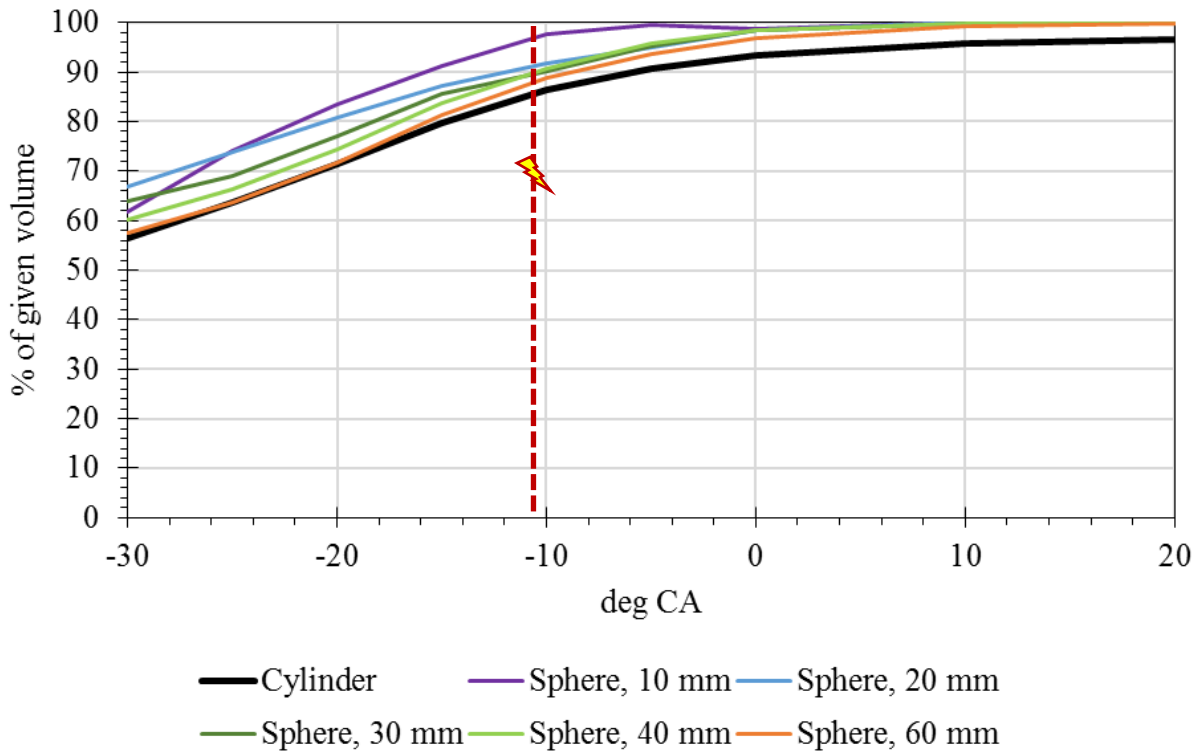


Figure 3.63 - Percentage of the cylinder and specific spherical volumes occupied by a mixture in the λ range of [1.05, 1.15] for SVA at global $\lambda=1.1$, 7.9 bar GMEP, 1500RPM

Lastly, the excess air dilution was increased to $\lambda=1.2$, and the same analysis was performed. Overall percentage fell significantly compared to $\lambda=1.1$, however it was kept above the $\lambda=1.0$ case for both DVA and SVA, which is in-line with engine out NO_x emissions. These global trends were reassuring and confirmed the hypothesis formulated earlier about the importance of contribution towards NO_x production coming from the [1.05,1.15] regions within the cylinder. It was noted that lines representing concentration of specific lambda range for 10-40 mm spheres past the ignition had opposite sign gradients for DVA and SVA. Smallest, 10 mm spheres had 0% concentration for both DVA and SVA, however 20, 30, 40 mm spheres appeared to be quite well-matched during ignition ($\pm 5\%$) under two valvetrain operating strategies. As mentioned earlier, past the ignition point concentration for DVA increased for these spheres, whilst SVA concentration decreased and it might be attributed to higher stratification achieved by DVA and SVA being closer to the nominal $\lambda=1.2$ around the spark plug. When total in-cylinder concentration was studied, it was found that SVA maintained approximately 10% higher values throughout which is correlated with the engine out NO_x emissions.

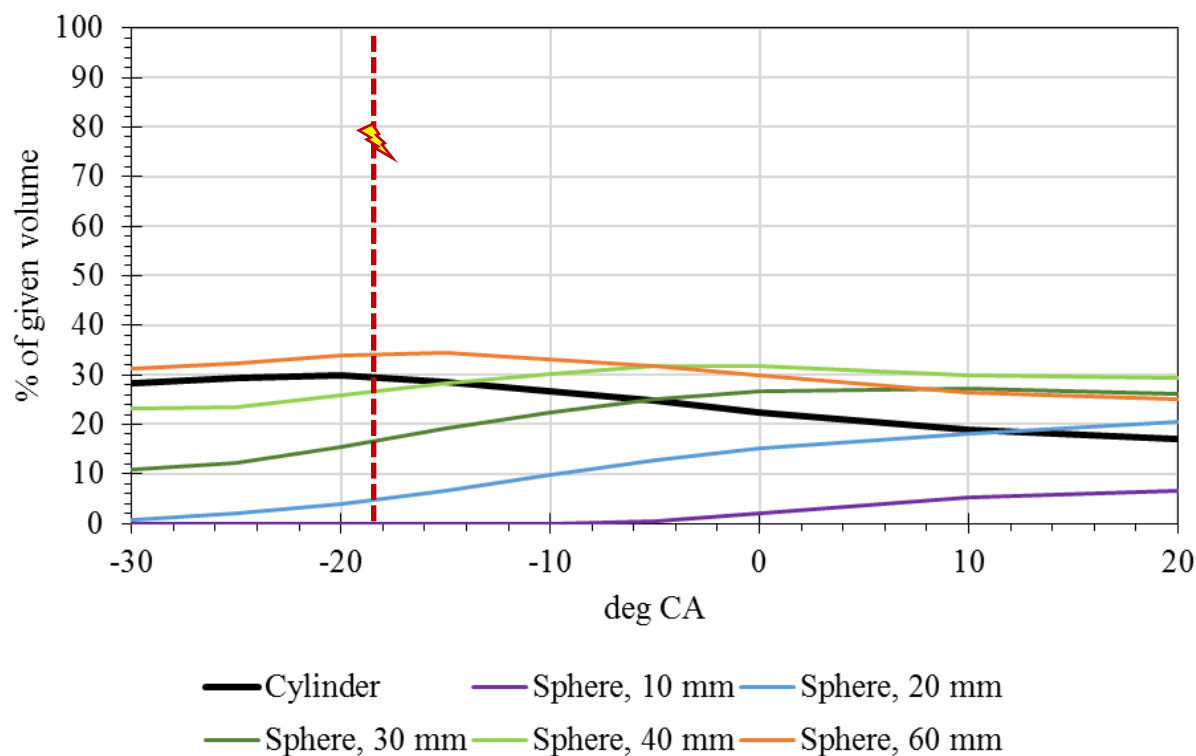


Figure 3.64 - Percentage of the cylinder and specific spherical volumes occupied by a mixture in the λ range of $[1.05, 1.15]$ for DVA at global $\lambda=1.2$, 7.9 bar GMEP, 1500RPM

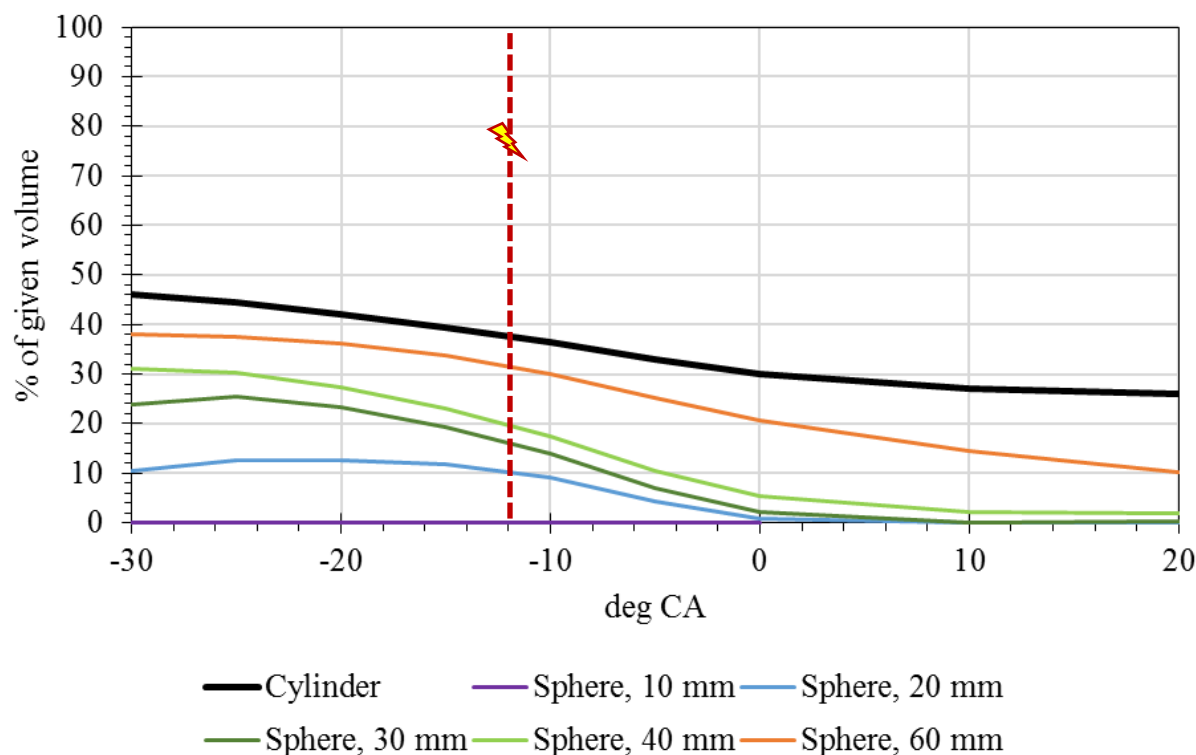


Figure 3.65 - Percentage of the cylinder and specific spherical volumes occupied by a mixture in the λ range of $[1.05, 1.15]$ for SVA at global $\lambda=1.2$, 7.9 bar GMEP, 1500RPM

Based mostly on the trends seen for volumes of spheres of 40 mm and larger, it can be concluded that there was a sharp increase in volumes within the specified range of [1.05, 1.15] lambda, when total dilution was increased from $\lambda=1.0$ to $\lambda=1.1$. Further increase to $\lambda=1.2$ brought decrease in volumes, however they stayed higher than the base $\lambda=1.0$ which also follows experimental results. The biggest differences between DVA and SVA were seen at $\lambda=1.1$, followed by $\lambda=1.2$, which was again seen in experiments, however the difference was not as pronounced as the simulations might suggest (delta of 812 ppm and 801 ppm for $\lambda=1.1$ and $\lambda=1.2$ respectively).

To further understand differences in the NO_x emissions, Zeldovich mechanism which predicts NO production in the post-flame gases was taken from Heywood (1988).

$$\frac{d[NO]}{dt} = \frac{6 \cdot 10^{16}}{T^{0.5}} \exp\left(\frac{-69090}{T}\right) [O_2]_e^{0.5} [N_2]_e \quad (3.8)$$

It is evident that the rate of production is strongly dependent on the temperature (exponential term). For the purpose of illustrating that dependence, in-cylinder temperature was calculated using the ideal gas law (Figure 3.66). Trapped mass and molecular weight of the exhaust gases were extracted from CFD simulations, whilst experimental, IndiCom acquired in-cylinder pressure values (average over 300 consecutive cycles) were used. It was duly noted that such calculation is only an approximation at best and was performed for both DVA and SVA. Additional point of $\lambda=1.5$ was added to the analysis, as at this point difference between DVA and SVA in NO_x emissions was marginal.

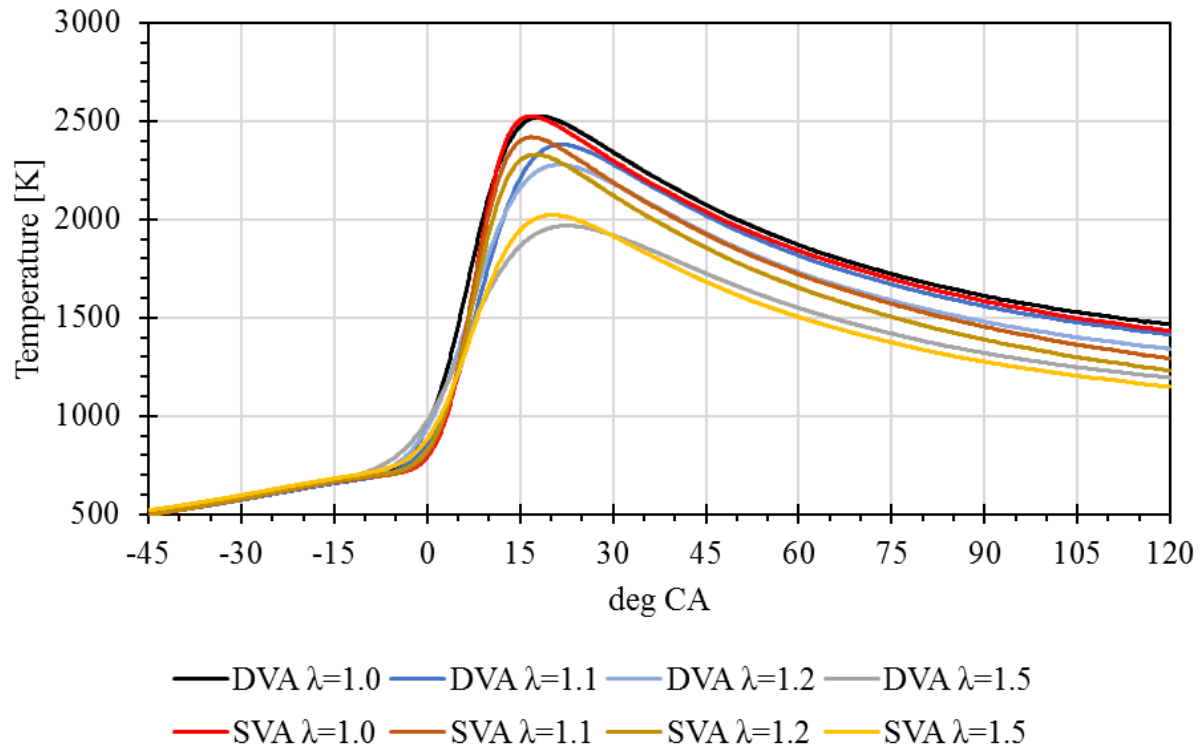


Figure 3.66 – Temperature derived from the in-cylinder pressure traces using the ideal gas law for 1500 RPM, 7.9 bar GMEP

Calculated temperatures were used to assess the contribution of the temperature term from equation 3.8 and were presented in Figure 3.67. Note that values are arbitrary in this plot, as it was constructed for illustration of relative differences between investigated points. Exponential relationship resulted in approximately 80% reduction (1535 units vs 305 units for DVA) attributed to about 100K temperature decrease from $\lambda=1.0$ to $\lambda=1.1$. Further 100K peak temperature decrease in DVA, resulted in another 70% reduction in the temperature related term (305 units vs 90 units). Also, the relatively small temperature differences between DVA and SVA (as seen in Figure 3.66) were significantly amplified (especially for $\lambda=1.1$ and $\lambda=1.2$), showing that higher SVA NO_x levels were linked to combustion temperature. Temperatures at $\lambda=1.5$ for both DVA and SVA were around 500K lower than $\lambda=1.0$, resulting in the arbitrary temperature term being two orders of magnitude lower, hence they were not plotted in Figure 3.67.

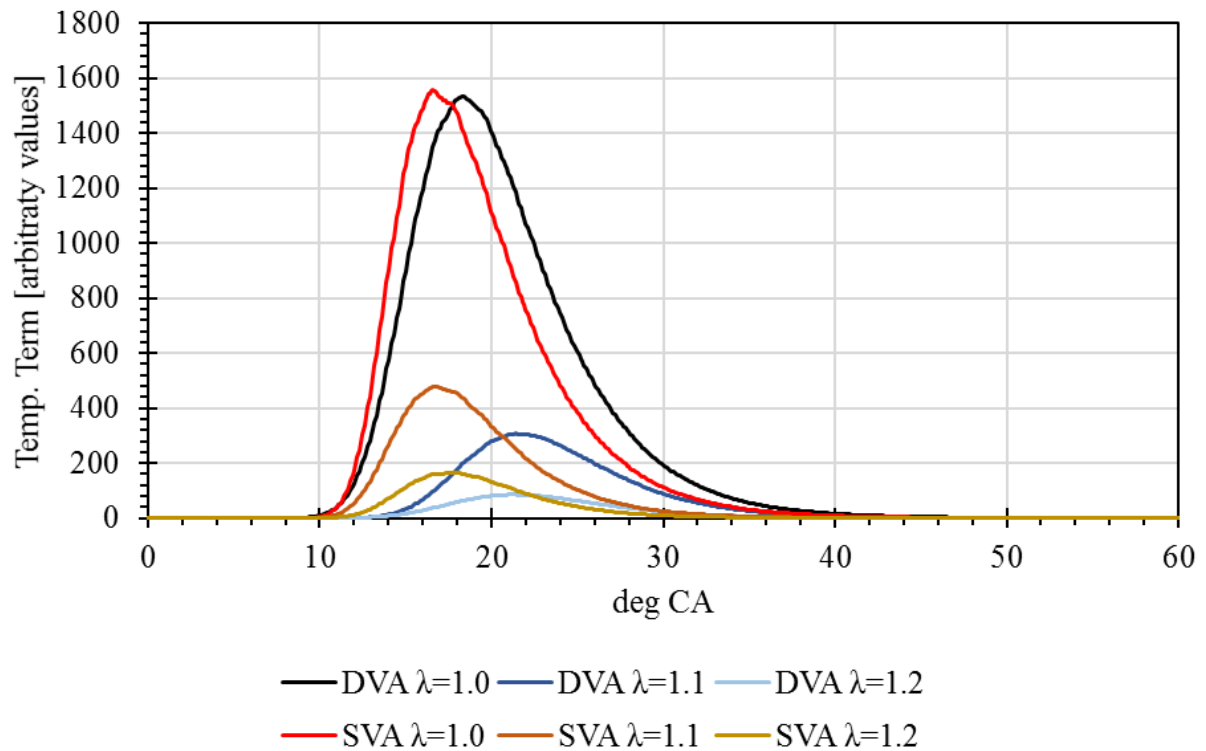


Figure 3.67 – Temperature term of Zeldovich mechanism equation predicting NO production for 1500 RPM, 7.9 bar GMEP

It was duly noted that not all phenomena could be successfully modelled, such as incomplete combustion (becoming more prevalent with increased dilution), carbon deposits, fuel-oil mixing, HC outflow from the crevices and they all contribute towards emission formation, however the main hypothesis about the species formation was able to successfully link experimental data and numerical simulations. Such modelling can be effective in estimating whether concepts are worth progressing towards expensive prototyping and experimental testing.

3.5. Concluding Remarks

A thorough investigation of running the Single Cylinder Research Engine (SCRE) with Single Valve Actuation (SVA) strategy, where one of the intake valves remained completely shut during the air intake part of the cycle, provided in-depth understanding of benefits and shortfalls of employing this strategy across various speed and load points compared to a standard Dual Valve Actuation (DVA) mode. Experimental data proved that SVA strategy was an effective mean for increasing dilution tolerance (both running lean and feeding external EGR) irrespective of engine speed and load. These findings were supplemented by CFD simulations which generated important insights that enabled detailed investigation of the in-cylinder motion, heat transfer and emissions formation. Conclusions reached upon completion of this study are as follows:

- Symmetric design of AJ-20 “high tumble” intake ports did not produce any swirl under DVA operation. Once an SVA strategy was introduced, swirl motion became dominant and resulted in increased in-cylinder bulk air motion, generating increased TKE at ignition.
- Increased TKE which is a governing parameter for speed of flame propagation and combustion duration resulted in significantly shorter combustion under SVA operation across all running conditions. The majority of the difference between SVA and DVA was manifested in the initial stage of combustion (0-10% MFB), whilst later stages of the combustion process were less sensitive (10-90% MFB).
- SVA valve action also trapped more residuals within cylinder. It was caused by the lower admission (and expelling) of fresh air during the intake and compression part of the cycle, resulting in higher in-cylinder temperatures being recorded for SVA. Higher in-cylinder temperatures recorded under SVA running conditions, in turn resulted in increased laminar burning velocity which was yet another factor contributing towards faster SVA combustion.
- Heat transfer effects were found to be offsetting faster burn for SVA. Increased heat transfer to the engine walls, cylinder head and piston for SVA resulted in lack of fuel

consumption benefit under stoichiometric conditions and marginal GSFC improvements for equivalent dilution levels under most conditions.

- NVH derived 3% CoV of NMEP combustion stability target was reached with lower dilution levels for DVA compared to SVA. Once the engine was operated targeting the limit, fuel consumption benefit of SVA was realised. Depending on the running condition, the results were:
 - At 1500 RPM, 3.6 bar GMEP lambda was increased from 1.15 to 1.5 at 0% EGR, resulting in 2.9% GSFC reduction and from 1.0 to 1.16 at 17.7% EGR, resulting in 3.7% GSFC reduction.
 - At 1500 RPM, 7.9 bar GMEP lambda was increased from 1.38 to 1.74 at 0% EGR, resulting in 0.8% GSFC reduction and from 1.08 to 1.34 at 20% EGR, resulting in 3.1% GSFC reduction.
 - At 2000 RPM, 9.8 bar GMEP lambda was increased from 1.53 to 1.70 at 0% EGR, without realising any fuel consumption reduction and from 1.23 to 1.25 at 20% EGR, resulting in 0.2% (negligible) GSFC reduction.
- Heat transfer analysis identified two main drivers affecting the heat transfer differences between DVA and SVA – averaged in-cylinder temperature delta between the two and flow field effects (higher gas velocity close to the engine walls).
- Temperature difference was found to be dominant effect on total heat transfer delta between DVA and SVA for low load, whilst flow field effects become prevailing mechanism for higher loads. Unfavourable heat transfer effects were experimentally captured through compression polytropic coefficients calculations that were lower for SVA compared to DVA for any given dilution.
- Heat transfer delta between DVA and SVA was increasing both in absolute and relative terms with load increase. Under stoichiometric conditions it was found to be 500 W (23.6% increase), 1500 W (50% increase) and 3000 W (56% increase) for 1500 RPM, 3.6 bar GMEP; 1500 RPM, 7.9 bar GMEP and 2000 RPM, 9.8 bar GMEP respectively.
- Optimum level of enhanced in-cylinder motion caused by SVA should exist, where benefits of faster burn are not being offset by the detrimental effects of significantly increased heat transfer, while running under high dilution. It is assumed that such operation is possible within low to low-medium load region of engine operation and results in GSFC benefit of adopting SVA.

- Mixture preparation analysis showed that SVA benefited from more homogenous mixture than DVA at ignition. By introducing Homogeneity Index (HI), it was quantified to be 0.97 vs 0.94 for 1500 RPM, 7.9 bar GMEP.
- More homogenous mixture of SVA had an effect on lowering the THC emissions. It is believed that rich pockets (formed mainly on the exhaust side of the chamber) found in cylinder under DVA operation contributed towards higher THC production.
- THC emissions were also affected by higher fuel impingement on the piston during the intake stroke for DVA. Liquid film recorded for DVA was almost twice as high as SVA, recorded at 0.81 mg vs 0.48 mg. It was a result of SVA motion being more effective in disintegrating the spray plumes.
- NO_x emissions were found to be higher for SVA than DVA across all measured points at 2000 RPM, 9.8 bar GMEP, whilst lower speed and load conditions exhibited trends in which NO_x formation was initially higher for SVA within $\lambda=1.0$ to $\lambda=1.2$ region, followed by the two matching and intersecting once the air/fuel equivalence ratio was increased. Such behaviour was also sensitive to EGR dilution and load. Increased EGR rate made DVA and SVA trend lines meet at lower lambda, whilst increasing the load pushed them apart.
- Artificial spheres, centred around the spark plug gap, were introduced to investigate air/fuel equivalence ratio distribution and aid in NO_x formation analysis. Hypothesis of the highest NO_x production taking place when local lambda is within [1.05,1.15] lambda was confirmed for 1500 RPM, 7.9 bar GMEP case.
- The Zeldovich mechanism of NO_x production in post-flame gases was studied and it was shown that temperature plays a critical role due to its exponential term. It was found that at 1500 RPM, 7.9 bar GMEP increasing lambda by 0.1 (vs stoichiometry) resulted in the in-cylinder peak temperature being lowered by about 100K (~4%) and approximately 80% reduction in the temperature component within the NO_x formation equation. Temperature contribution was also found to be very similar for both DVA and SVA.

Chapter 4

Staggered Valve Lift Operation

4.1. Preliminary Remarks

Chapter 3 presented findings on how the dilution tolerance and thus fuel efficiency of an AJ-20 SCRE engine was increased when adopting a Single Valve Actuation (SVA) – the complete closure of one of the intake valves – over Dual Valve Actuation (DVA). It was shown that an increased in-cylinder flow and TKE contributed to shorter, more efficient combustion, however there was also noticeable penalty in increased heat transfer to the engine walls, piston and cylinder head. That detrimental heat transfer effect was becoming more prevalent at higher engine loads.

It was concluded that there must be an optimum point, where charge motion is enhanced by introducing swirl motion which promotes faster burn, whilst kept below its maximum to lower the undesirable heat transfer effects.

With a view to broadening the understanding of the SVA vs. DVA trade-offs and identifying the most efficient running mode, investigation making use of staggered lift capabilities of the CVVL system was launched.

The CVVL valvetrain system enabled utilising either Late Intake Valve Opening (LIVO) or Early Intake Valve Closure (EIVC) modes at various maximum lifts on any of the valves independently of the other. Figure 4.1 represents valve lift strategies for LIVO and EIVC respectively, with 1 mm maximum lift increments used for this study (up to 5 mm maximum lift). It should be noted that LIVO strategy was dictated by the mechanical limitations of the CVVL system. It ran on symmetric camshaft profile with a fixed centre-point. That design enabled delay in opening the valve, however moving IMOP beyond the “base” position was not possible. Using the outlined profiles, an optimised staggered valve lift strategy (i.e. one of the valves running in full lift mode, whilst the other performing LIVO or EIVC at a specified lift) was utilised under dilute conditions to assess the dilution tolerance and heat transfer effects associated with such operating mode.

It should be noted that due to engine test cell maintenance and engine being used for other testing required by the project sponsoring company, it was rebuilt after the initial valve deactivation study described in Chapter 3. The rebuild resulted in a slightly lower compression ratio of 10.860:1 (vs 11.006:1 used previously), and it required re-running both DVA and SVA baseline results to ensure most accurate, direct comparisons could be made. Experimental results were obtained for all three points (1500 RPM, 3.6 bar GMEP; 1500 RPM, 7.9 bar GMEP and 2000 RPM, 9.8 bar GMEP), however based on the findings from Chapter 3, the full set of CFD simulations, including fuel spray modelling complemented only 1500 RPM, 7.9 bar GMEP running condition.

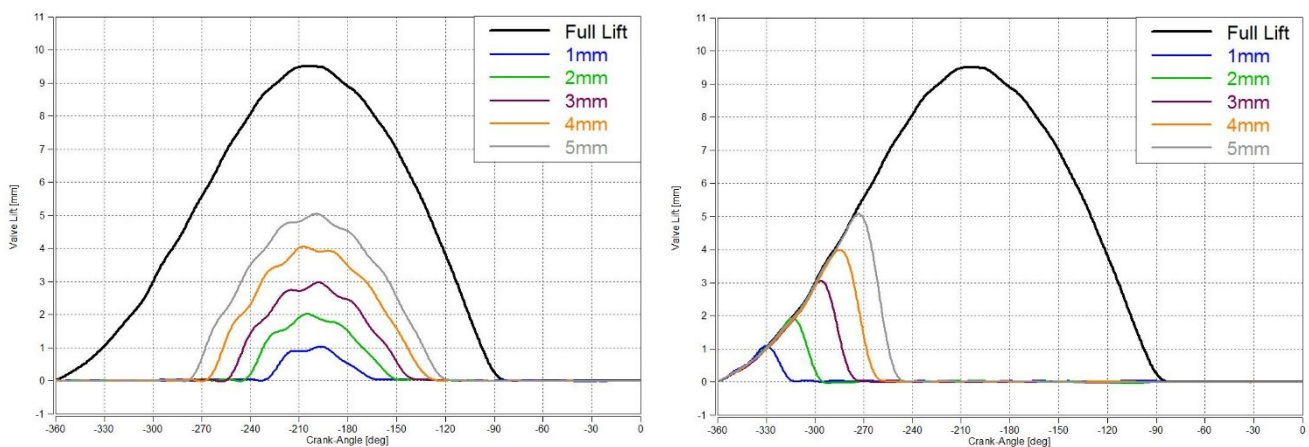


Figure 4.1 – LIVO and EIVC valve lift profiles used in the study; ranging from 1 to 5mm max lift

4.2. Results

4.2.1. 1500 RPM, 3.6 bar GMEP

4.2.1.1 Late Intake Valve Opening (LIVO) Operation, Lambda Study

First set of experiments run on SCRE were the LIVO lambda swings at 0% EGR, 1500 RPM, 3.6 bar GMEP. As per previous studies, combustion stability limit of 3% CoV of NMEP was used to identify maximum dilution level that can be tolerated by the engine. It was found that introduction of LIVO strategy presented benefits closely matched to these observed for SVA. Figure 4.2 clearly shows that all LIVO modes (especially 1-4 mm max lift on the second valve) follow stability trend of the SVA (red) line and outperformed DVA by a healthy margin.

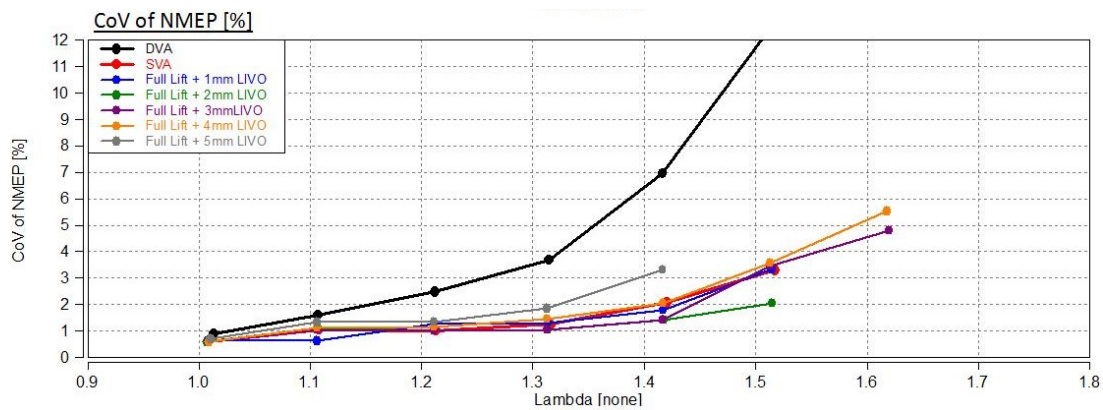


Figure 4.2 – Combustion stability of LIVO cases for lambda sweep at 1500 RPM, 3.6 bar GMEP

Such behaviour can be attributed to significantly faster burn throughout the cycle. Ignition to 10% MFB and 10 to 90% MFB displayed in Figure 4.3 showed interesting clustering of the trend lines. SVA and LIVO up to 3 mm maximum lift were characterised by almost identical burn durations, whilst 5mm lift was closely matched to DVA condition. Initial stage (ignition to 10% MFB) was about 6°CA faster at stoichiometric condition (12°CA vs 18°CA) for SVA and 1-3 mm maximum lift cases vs DVA and 5 mm lift, and as the dilution is increased, the difference became more pronounced and reached almost 10 °CA at $\lambda=1.5$. Later stages of combustion (10 to 90% MFB) presented constant offset of about 3°CA between the two aforementioned clusters and it was independent of dilution increase.

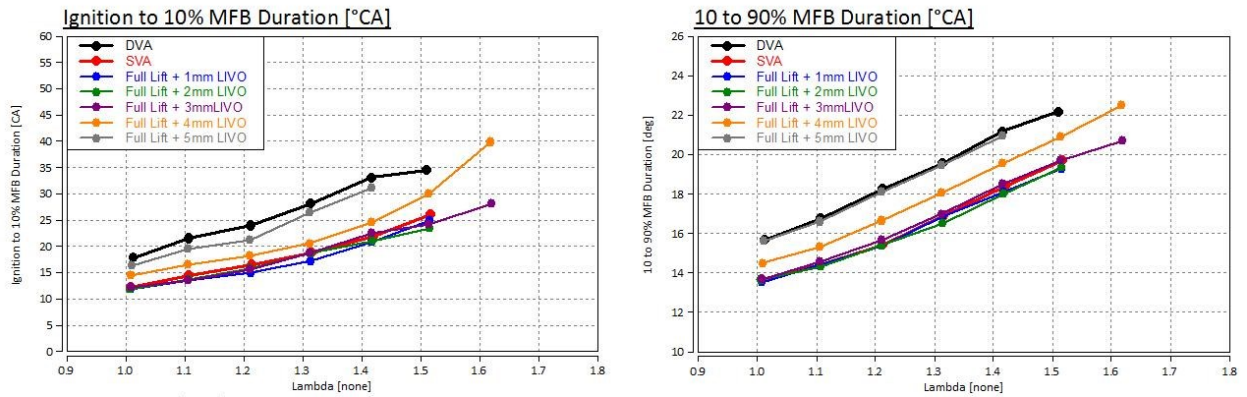


Figure 4.3 – Burn duration analysis of LIVO cases for lambda swings at 1500 RPM, 3.6 bar GMEP

Knowing that burn durations are correlated with TKE, which is a function of the in-cylinder flow, tumble and swirl motions were plotted in Figures 4.4 and 4.5. Opening the second valve up to 5 mm had negligible effect on the tumble motion, however swirl motion was affected, especially past BDC. It is in-line with expectations, given that second valve opened much later within the cycle (at approximately mid induction stroke for the most advanced 5 mm LIVO scenario, and respectively later for lower lifts, as shown in Figure 4.1). Swirl was again the governing motion within the cylinder, as it caused reverse tumble and swirl/tumble ratio (taken as peak to peak) was approximately 1.8 for all LIVO cases.

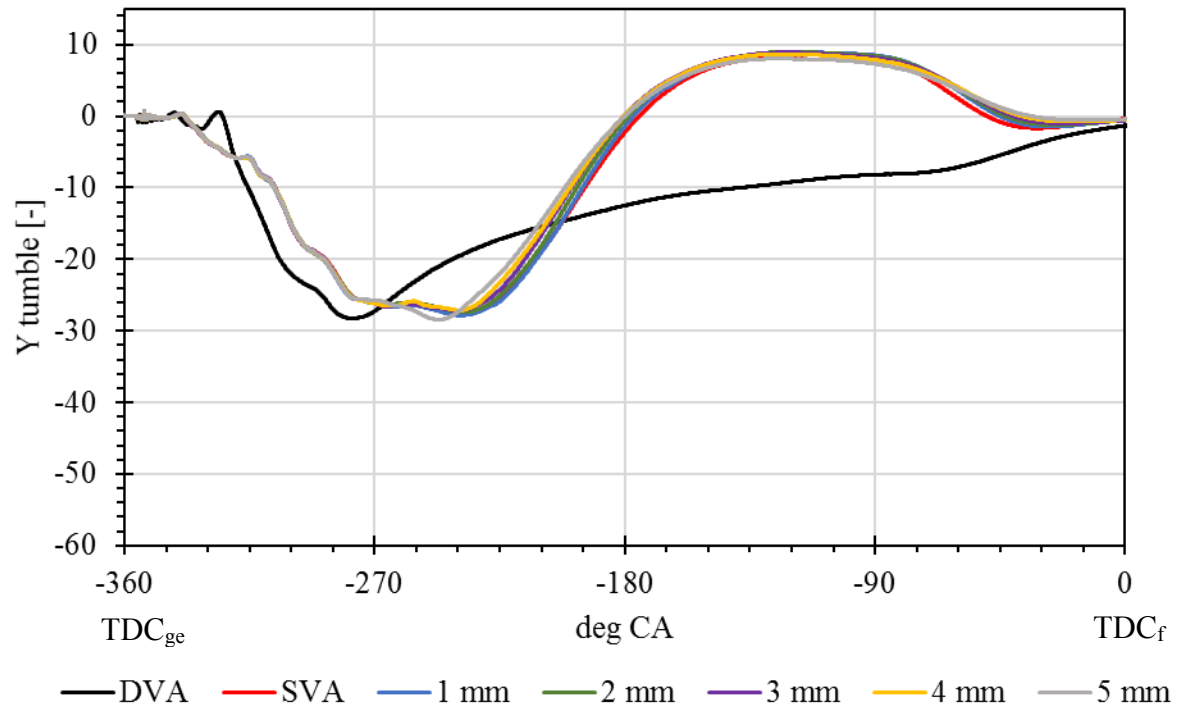


Figure 4.4 – Tumble motion comparison between various max lift LIVO cases at 1500 RPM, 3.6 bar GMEP, $\lambda=1.0$

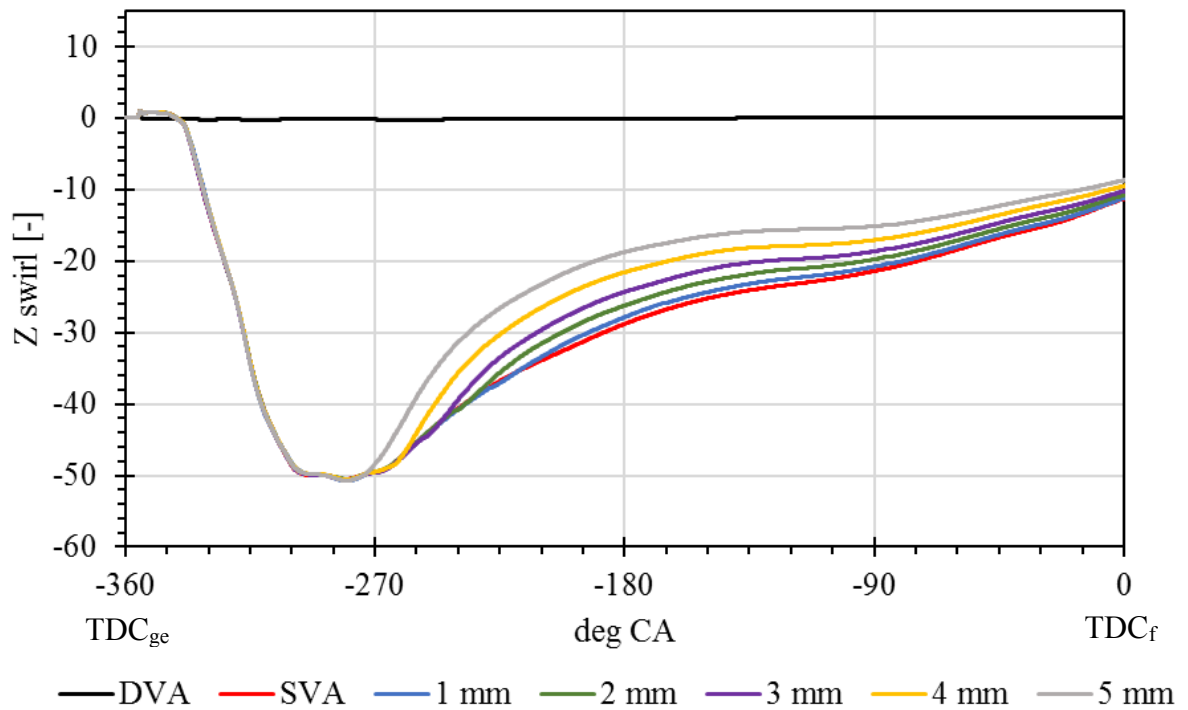


Figure 4.5 - Swirl motion comparison between various max lift LIVO cases at 1500 RPM, 3.6 bar GMEP, $\lambda=1.0$

Differences in swirl during the compression stroke were believed to have a sizeable effect on the TKE. Presented in Figure 4.6, it can be seen that at ignition (15°CA bTDC_f for DVA, 11°CA bTDC_f for SVA) SVA and all the LIVO cases benefited from increased TKE compared to DVA. There was also pronounced cluster of SVA and 1-3 mm max lift LIVO cases, which agrees with trends discussed earlier for reduced burn durations and increased combustion stability. Decrease in TKE at ignition was quantified for 5 mm max lift LIVO case which exhibited pronounced reduction in burn durations at approximately 15% (compared to SVA and 1-3 mm LIVO), whilst it was close to 20% for DVA.

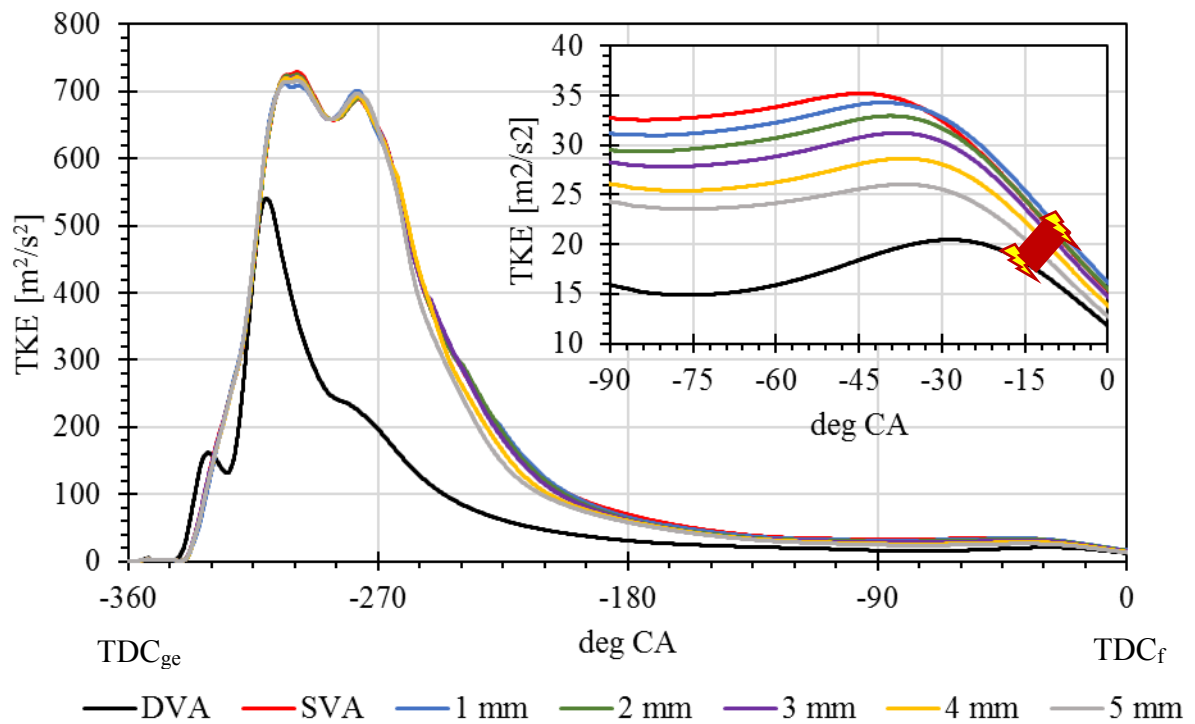


Figure 4.6 – TKE comparison between various max lift LIVO cases at 1500 RPM, 3.6 bar GMEP, $\lambda=1.0$ with a zoom on mid-stroke compression to TDC region

Having discussed how stability was significantly improved and burn durations reduced due to increased TKE, GSFC effects of introducing LIVO operation were quantified. Adhering to 3% CoV of NMEP best results were achieved for 3 mm LIVO at $\lambda=1.4$, where fuel consumption dropped to 233 g/kWhr vs the 240 g/kWhr of DVA at $\lambda=1.2$, a reduction of 3%. This result

was also 1.5% better than the minimum achieved by SVA at $\lambda=1.5$, and within 1% of the 1 mm, 2 mm and 4 mm values, all achieved at $\lambda=1.5$.

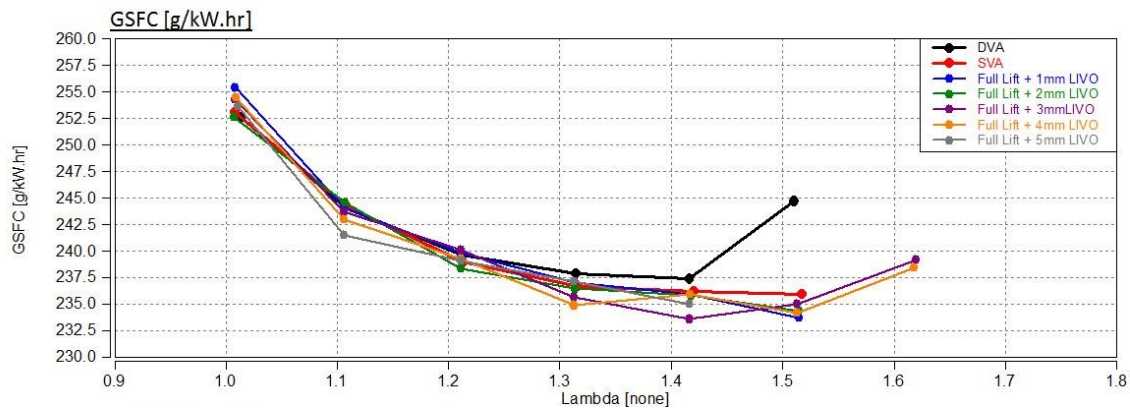


Figure 4.7 – GSFC of LIVO cases for lambda sweep at 1500 RPM, 3.6 bar GMEP

Analysis performed in Chapter 3, proved that heat transfer largely offsets benefits of increased TKE and faster burn when adopting SVA, hence same analysis was introduced for assessment of staggered lift assisted by LIVO strategy on the second intake valve. In-cylinder average temperatures for all LIVO cases were found to be closely matched to SVA (within 1K), whilst DVA was running about 15K cooler. It was attributed to the amount of fresh charge being admitted to cylinder and then expelled during the intake and compression strokes when running DVA. Introducing LIVO strategy (regardless of the max lift being used) had limited effect on that quantity, thus temperature and residuals were matched more closely to SVA. Opening second valve relatively late within the cycle did not allow for significant charge cooling. Closed cycle heat transfer curves representing $\lambda=1.0$ condition for all valve lifts were plotted in Figure 4.8. Difference in in-cylinder temperature between DVA and all other cases was clearly visible from BDC onwards. As per discussion in Chapter 3, enhanced charge motion (represented by swirl and tumble motions magnitude) had detrimental effect on the heat transfer and it can be seen in Figure 4.5 that reduced swirl corresponded to reduced heat transfer in Figure 4.8. Clear progression could be seen between all LIVO modes, and as the maximum lift on the second valve increased, the negative effects of heat transfer decreased. Having said that, heat transfer (similarly to tumble and swirl) was more closely matched to SVA rather than DVA. Relative differences in heat transfer were calculated to quantify the change in heat transfer. It was normalised to DVA and SVA being 0% and 100% respectively. Relative reductions were as follows: 2%, 13%, 17%, 34% and 43% for corresponding maximum lifts on the second valve of 1, 2, 3, 4 and 5 mm.

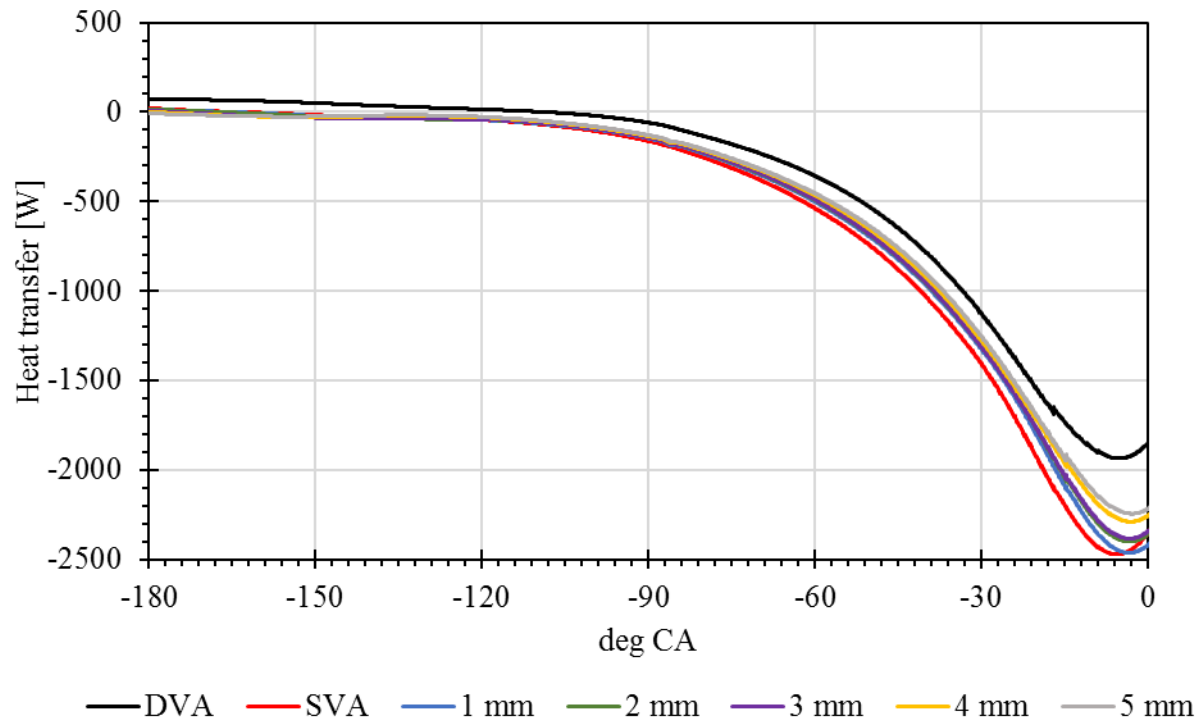


Figure 4.8 – Heat transfer comparison between various max lift LIVO cases at 1500 RPM, 3.6 bar GMEP, $\lambda=1.0$

Lastly, after analysing combustion and fuel efficiency, emissions under LIVO operation were scrutinised. CO and CO₂ trends did not provide any insightful information that could be used to distinguish between various operating modes, however unburned hydrocarbons and NO_x emissions varied between DVA, SVA and LIVO. THC emissions were generally the highest for DVA (with exception of stoichiometric condition), however they were not the lowest for SVA. 1 mm maximum lift LIVO case appeared to have the lowest THC emissions, whilst they were only up to 5% lower compared to SVA. The highest THC production displayed in Figure 4.9 for 3 mm and 4 mm maximum lift cases under LIVO operation at $\lambda=1.6$ should be neglected as combustion stability was worse than 5% CoV of NMEP. These data points represent contribution from partial burn cycles (characterised by high THC production and not suitable for practical engine calibration).

NO_x emissions were in line with trends found earlier in Chapter 3. SVA and 1-3 mm LIVO cases represented almost identical, highest values within lambda [1.0, 1.2] range, whilst DVA was characterised with the lowest NO_x production within this region. Starting at $\lambda=1.3$, onwards all tested conditions collapsed onto the single line and differences became almost negligible. It was attributed to much reduced in-cylinder temperatures and less favourable NO_x

production conditions, resulting in NO_x levels dropping below 1000 ppm. It also proved that valvetrain action was much less effective in tackling the NO_x production than increasing the air-fuel equivalence ratio which dramatically decreases NO_x levels past its maximum at $\lambda=1.1$. Further analysis on emissions formation, supported by CFD modelled fuel injection can be found in section 4.2.3 discussing 7.9 bar GMEP load case. It was done to ensure consistency with Chapter 3, where this exact point was modelled.

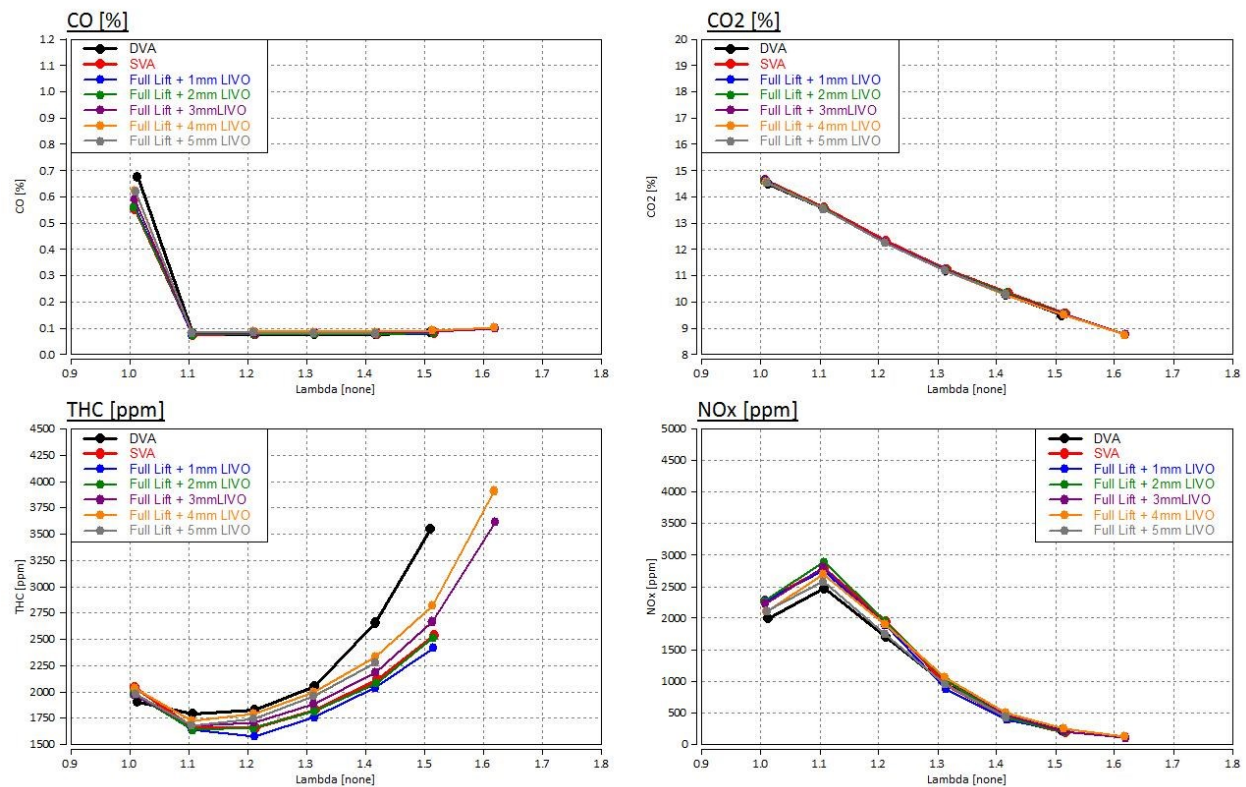


Figure 4.9 – Emissions formation comparison for lambda sweep between various max lift LIVO cases at 1500 RPM, 3.6 bar GMEP

4.2.1.2 Early Intake Valve Closing (EIVC) Operation, Lambda Study

Early Intake Valve Closing (EIVC) mode was used to run set of experiments on SCRE to generate the same set of lambda swings (as LIVO presented in 4.2.1.1) as part of initial investigation into the possible benefits of employing such valvetrain strategy. Again, it was experimentally proven that this method is another viable mechanism for extending dilution tolerance and improving stability of combustion compared to traditional DVA operation. Figure 4.10 shows that all 1-5 mm maximum lifts on the second valve running in EIVC mode

were better matched to SVA rather than DVA, offering substantial reduction in cyclic variability. Limit of $\lambda=1.2$ reached by DVA, whilst adhering to 3% CoV of NMEP (NVH based) criteria, was increased to $\lambda=1.5$ for 1 mm, 2 mm and 3 mm max lift EIVC conditions.

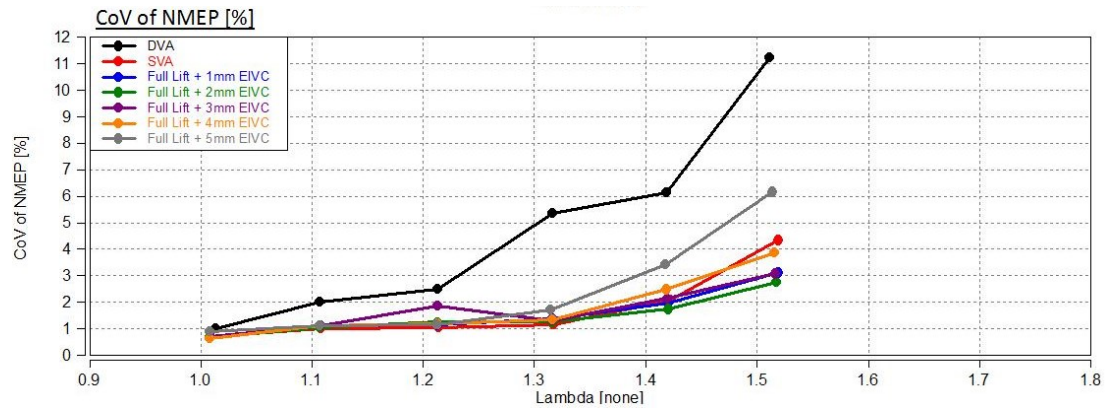


Figure 4.10 - Combustion stability of EIVC cases for lambda sweep at 1500 RPM, 3.6 bar GMEP

Contrary to the LIVO strategy, not all EIVC lifts were characterised by combustion faster than DVA. Only SVA, 1 mm and 2 mm EIVC presented significantly shorter burn durations (all three can be seen clustered in Figure 4.11), whilst 3 mm max lift was only fractionally faster than DVA scenario. Interestingly, 4 mm and 5 mm EIVC cases showed very similar behaviour, and whilst their ignition to 10% MFB was very similar to that of DVA (especially up to $\lambda=1.4$), 10 to 90% MFB was noticeably longer throughout the entire lambda sweep. Such turning point observed at 3 mm max lift employing EIVC strategy suggested that in-cylinder behaviour and mixture preparation differed from what has been seen up to this point (both SVA and LIVO vs DVA).

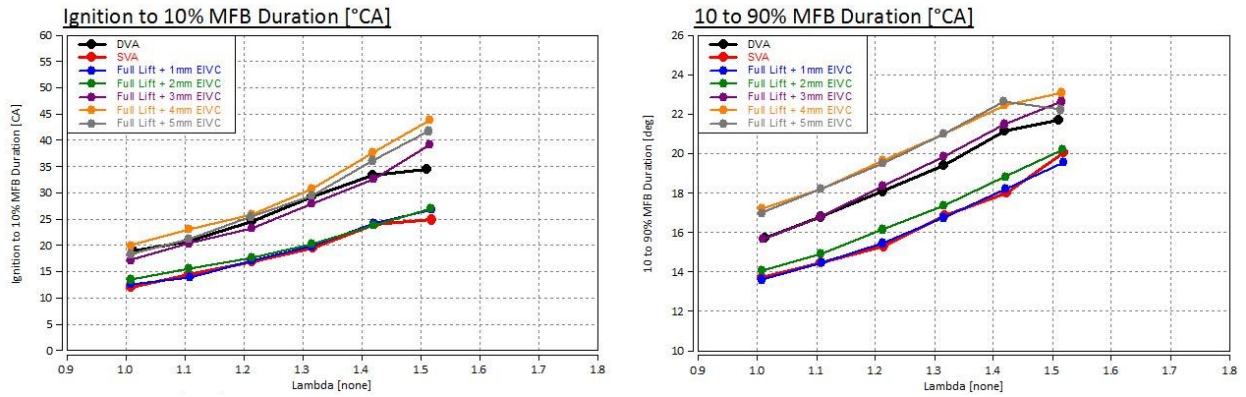


Figure 4.11 - Burn duration analysis of EIVC cases for lambda swings at 1500 RPM, 3.6 bar GMEP

After duly noting the detrimental effect of opening second valve more than 3 mm in EIVC mode on the burn durations, in-cylinder flow characteristics were examined in detail. It was found that tumble motion of 1 mm and 2 mm max lift cases was in good agreement with SVA scenario (1 mm and SVA were almost line on line in Figure 4.12 representing tumble motion within the intake and compression strokes). Both 1 mm and 2 mm cases exhibited strong reverse tumble past BDC. Increasing the lift by another 1 mm, to 3 mm case showed much bigger step. Reverse tumble was much less pronounced and started about 30 °CA later within the cycle, however it was still noticeable. Maximum, 5 mm EIVC lift examined in this study did not produce any reverse tumble motion, whilst 4 mm case was borderline, with a maximum value reached for reverse tumble being only 1 (vs 9 for SVA and 1 mm lift).

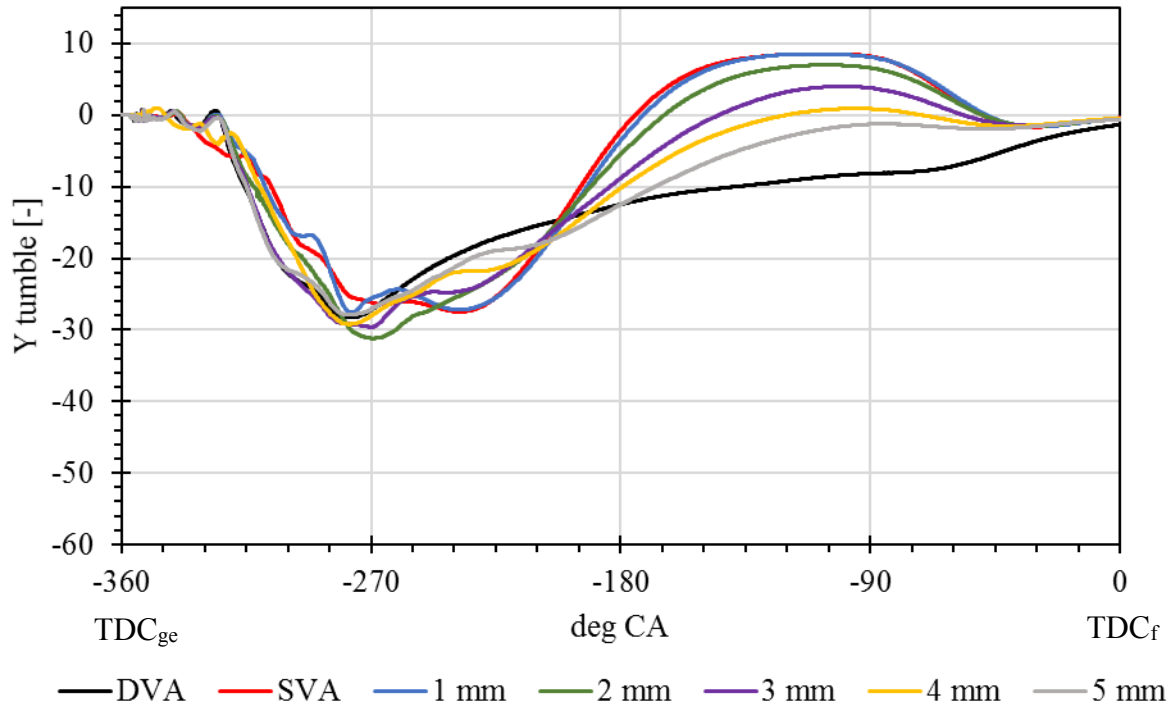


Figure 4.12 - Tumble motion comparison between various max lift EIVC cases at 1500 RPM, 3.6 bar GMEP, $\lambda=1.0$

Swirl motion analysis produced step like plot which can be seen in Figure 4.13. As expected, SVA was characterised by the strongest swirl motion, which was reduced as the second valve opened in 1 mm increments. Compared against Figure 4.5 showing swirl for LIVO cases, it is obvious that the fact that both valves open simultaneously from the beginning dramatically reduced swirl formation (keeping it at 0 up to closing of the second valve). If the second valve is not promptly closed during the initial phase of the valves opening, when jet velocity sharply rises, then the overall swirl formation is delayed and only fraction of the SVA potential can develop. Also comparing EIVC swirl to LIVO presented previously, big separation was noticeable in the last half of the compression stroke where ignition took place. It is believed that this significantly affects burn durations presented earlier through reduced TKE.

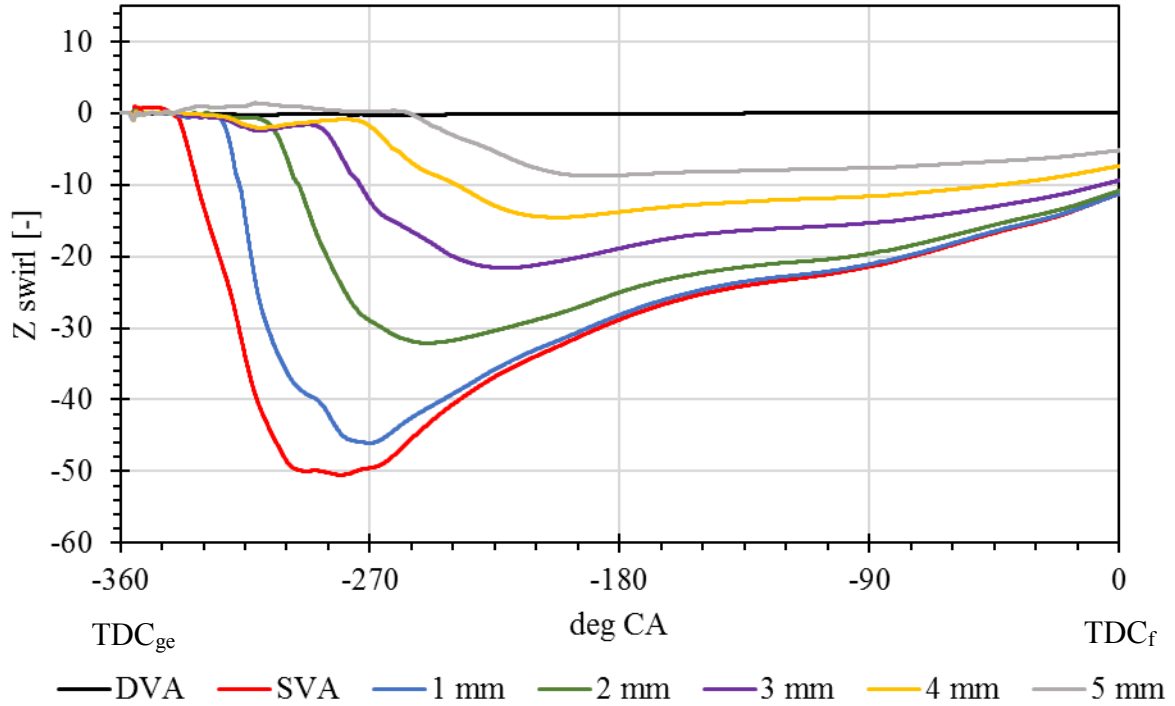


Figure 4.13 – Swirl motion comparison between various max lift EIVC cases at 1500 RPM, 3.6 bar GMEP, $\lambda=1.0$

After showing how EIVC staggered lift operating mode had the in-cylinder flow field different from SVA and LIVO, parameter having the biggest effect on burn durations and combustion stability was analysed. TKE (Figure 4.14) showed very interesting trends that were very well correlated with burn duration characteristics presented in Figure 4.11. Same clustering as the one already described for combustion metrics was exhibited for TKE in the ignition zone. Fast burning SVA, 1 mm and 2 mm lift EIVC cases represented the highest TKE at ignition, forming a cluster at around $23\text{--}24 \text{ m}^2/\text{s}^2$, whilst DVA and 3 mm case recorded about $19 \text{ m}^2/\text{s}^2$. 4 mm and 5 mm cases were also very close together at approximately $17 \text{ m}^2/\text{s}^2$ at ignition. These differences were not only qualitatively representative of the burn durations (thanks to the formed clusters), but also quantitatively as the difference in burn durations was slightly larger between [SVA, 1mm, 2mm lifts] and [DVA, 3mm lift] than it was between [DVA, 3mm lift] and [4mm, 5mm lift] for EIVC cases. Reduced TKE observed when employing EIVC at low load condition agrees with the results presented by Scheidt *et al.* (2014). In their study engine run in EIVC mode, with valve opening to about half of its maximum lift (comparable to 4 mm and 5 mm lifts used in this study) dramatically reduced TKE in the ignition region.

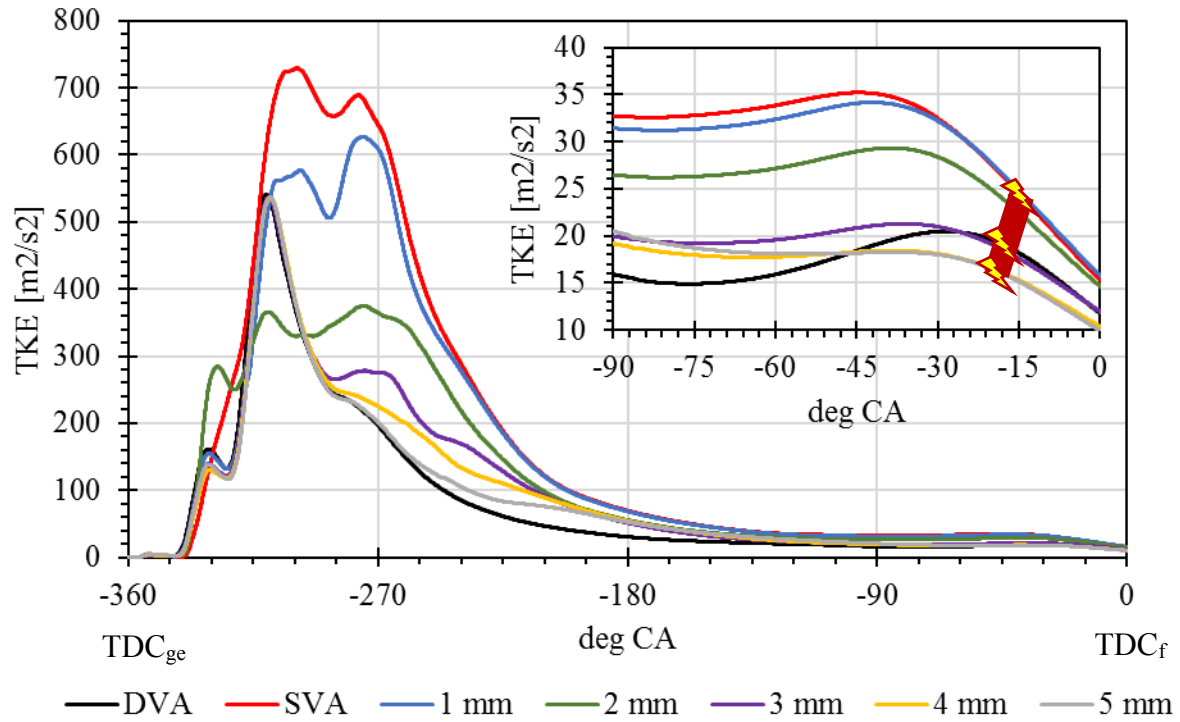


Figure 4.14 - TKE comparison between various max lift EIVC cases at 1500 RPM, 3.6 bar GMEP, $\lambda=1.0$ with a zoom on mid-stroke compression to TDC region

GSFC recorded by SCRE running in the EIVC mode was found to be the lowest for 1 mm, 2 mm and 3 mm EIVC max valve lift cases at $\lambda=1.4$. The minimum was recorded at 232 g/kWhr which was marginally better than LIVO which achieved 233 g/kWhr (Figure 4.7). GSFC line plotted vs λ for 3 mm EIVC case plateaued around its minimum at $\lambda=1.3$ and remained at such low level until $\lambda=1.5$, thus it was concluded to be the optimum running condition from fuel consumption perspective. Emissions and especially NO_x formation is discussed later in the thesis. It is acknowledged that fuel consumption analysis cannot be concluded without taking these into consideration.

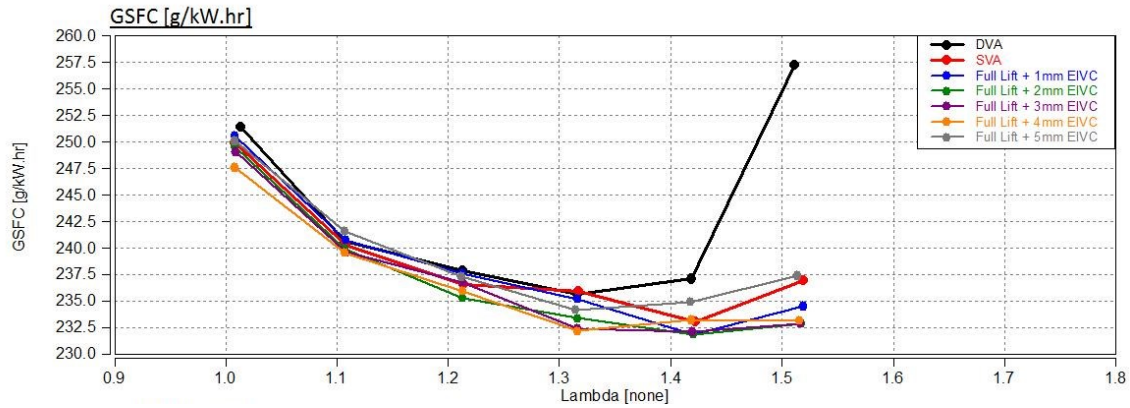


Figure 4.15 - GSFC of EIVC cases for lambda sweep at 1500 RPM, 3.6 bar GMEP

Having already established that heat transfer of SVA and LIVO cases offset benefits of faster burn, EIVC's heat transfer was plotted (Figure 4.16). It was found that EIVC heat transfer profiles were noticeably different than their LIVO counterparts. Opening second valve in EIVC mode resulted in heat transfer being significantly reduced, compared to SVA, once the maximum lift was 3 mm or higher.

Relative reductions were calculated in the same manner as for LIVO and they were as follows: 9%, 24%, 56%, 74% and 88% for corresponding maximum lifts on second valve of 1, 2, 3, 4 and 5 mm. These deltas were significantly different from ones noted for LIVO and summarised in Table 4.1. It can be seen that for each valve lift, heat transfer reduction was more than doubled for EIVC compared to LIVO (with exception of 2 mm lift where it was close to being nearly twice as high). It provided a clue as to why 3 mm maximum lifts for both EIVC and LIVO were optimum from the fuel consumption perspective, regardless of their vastly different burn durations. Whilst LIVO benefited more from the increased speed of combustion, EIVC suffered far less heat transfer increase compared to baseline DVA condition.

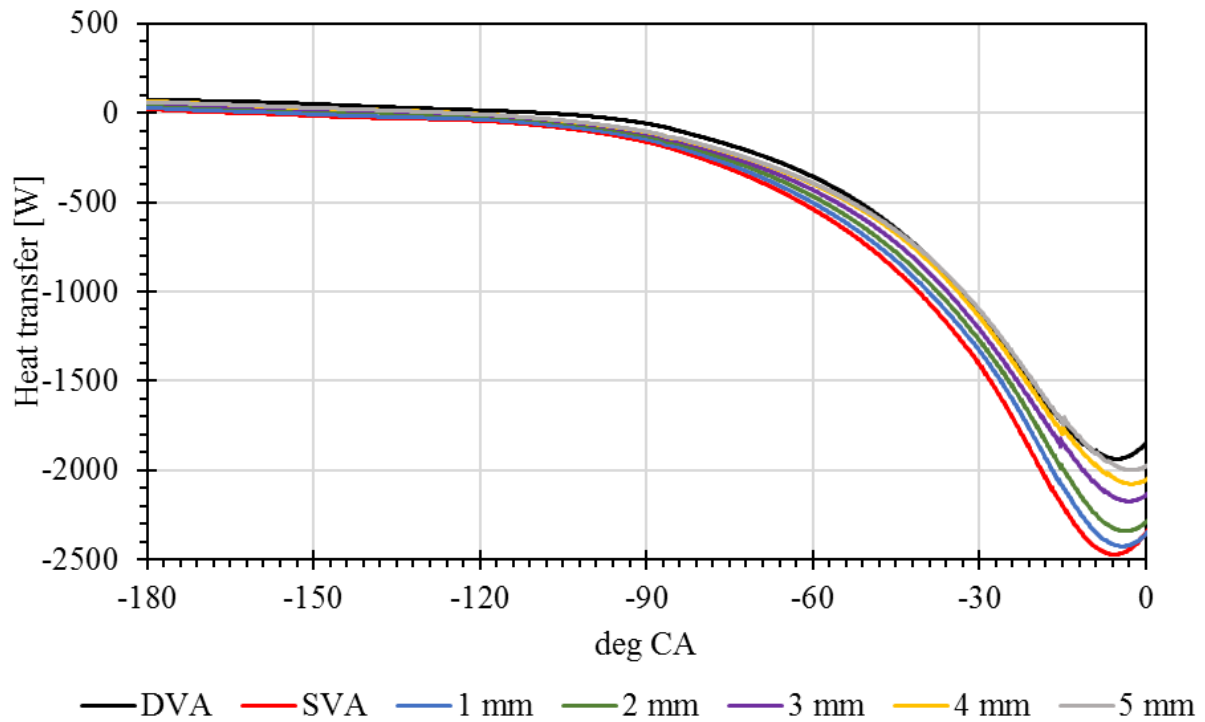


Figure 4.16 - Heat transfer comparison between various max lift EIVC cases at 1500 RPM, 3.6 bar GMEP, $\lambda=1.0$

Table 4.1 – Heat transfer reduction vs SVA baseline comparison of LIVO and EIVC cases at various maximum lifts at 1500 RPM, 3.6 bar GMEP, $\lambda=1.0$

Maximum valve lift on second valve	Percentage decrease of heat transfer compared to SVA [%]	
	LIVO	EIVC
1 mm	2	9
2 mm	13	24
3 mm	17	56
4 mm	34	74
5 mm	43	88

Emissions formation for EIVC was almost identical to that observed for LIVO with respect to CO and CO₂. THC formation was again the highest for DVA (past $\lambda=1.1$), however it was the lowest for SVA. Under LIVO condition 1 mm lift was characterised by the THC readings lower

than SVA which was not the case for EIVC operation, and 1 mm lift was only matched to SVA up to $\lambda=1.3$. All other lifts appeared to be better matched to DVA up to $\lambda=1.3$, from which point onwards, there was a more even spread and THC emissions drastically increased. It was linked to pick up in combustion variation at this point, presented in Figure 4.10. NO_x emissions were the lowest for DVA in the highest production zone, up to $\lambda=1.2$, however there were no clear trends for EIVC lifts, other than the fact that lower lifts tended to be better matched to SVA rather than DVA, and characterised by relatively high production (especially in the $1.0 < \lambda < 1.2$ zone). Further analysis of emissions formation is presented for 1500 RPM, 7.9 bar GMEP test point, later in this chapter.

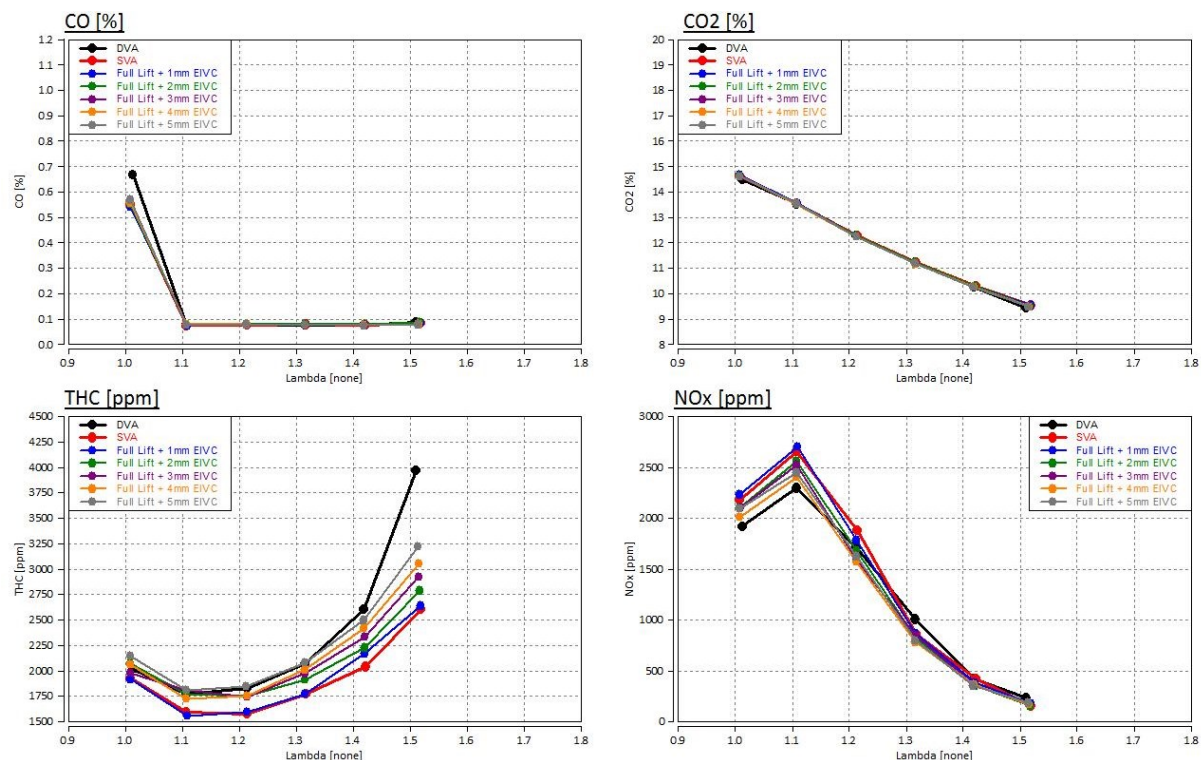


Figure 4.17 - Emissions formation comparison for lambda sweep between various max lift EIVC cases at 1500 RPM, 3.6 bar GMEP

4.2.1.3 Optimum

Once the LIVO and EIVC lambda swings were analysed, the optimum valve lifts were chosen for both of the strategies. Main criteria determining the selection was the minimised fuel consumption (GSFC) and subjective balance between burn duration benefit and heat transfer penalty. 3 mm maximum lifts were chosen for both strategies based on the lambda swings. Having fixed the staggered lift setting for LIVO and EIVC, such an optimised condition was subjected to full dilution sweep which included both running lean and feeding EGR (similar to method presented in Chapter 3). Contrary to method presented in Chapter 3, all the points recorded in this study adhered to the 3% CoV of NMEP criteria, hence all the points presented in the contour plots were valid for analysis purposes.

Starting with combustion metrics and assessment of burn durations, it was shown that LIVO required less ignition advance to achieve the same 50% MFB at 8°CA aTDC_f. Ignition advance also proved to be dilution dependent, with delta between LIVO and EIVC of 2.3°CA at stoichiometric condition, growing to approximately 9°CA when subjected to 20% EGR ($\lambda=1.0$) and 11°CA at $\lambda=1.5$ (0% EGR). Most of this difference could be attributed to early stage of the combustion process – ignition to 10% MFB, which is a flame development region. Differences in this parameter were again highly sensitive to dilution levels of SCRE, whilst later stages of combustion (10 to 90% MFB) were characterised by fairly constant offset which was not as affected by dilution levels. It was also much lower, kept at 1.2-3°CA.

Drawing on the results discussed earlier, differences in burn durations could be directly linked to TKE levels, and they were presented in Figures 4.6 and 4.14 earlier in this chapter. They were produced for $\lambda=1.0$ case, however sensitivity studies have shown that increasing lambda or EGR dilution increased plenum pressure, but changes in TKE were qualitatively negligible. According to baseline TKE values, 3 mm LIVO could range between 23 and 31 m²/s² in the ignition region, whilst 3 mm EIVC fell within the 18 to 22 m²/s². These ranges also showed that TKE profile of EIVC was much flatter (4 m²/s² range), whilst LIVO had more sharply rising TKE with increasing dilution (8 m²/s² range). It also provided valid explanation as to why the burn duration difference between two strategies was getting larger with increasing ignition advance, yielding sensitivity to dilution.

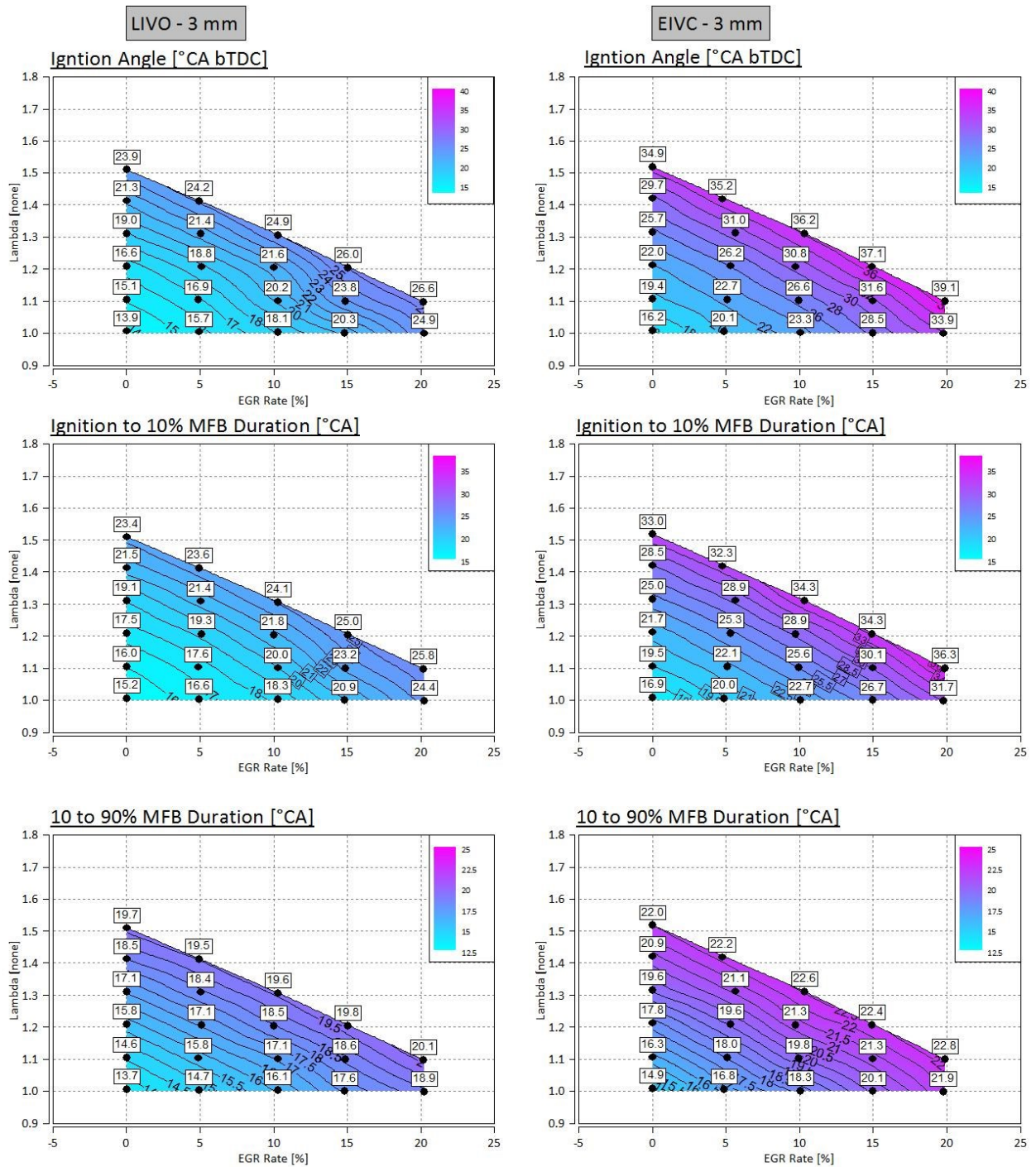


Figure 4.18 – Ignition angle, Ignition to 10% MFB and 10 to 90% MFB comparison between 3mm LIVO and 3mm EIVC at 1500 RPM, 3.6 bar GMEP

Regardless of the faster burn observed for 3 mm max lift LIVO case, its GSFC was worse for most of the points recorded during the study. Colour-coded Figure 4.19 shows clearly that EIVC plot had much bigger patches of light blue colour representing low end of GSFC measurements. At stoichiometric condition EIVC outperformed LIVO by 1.3%. 3 mm EIVC

reached minimum GSFC at $\lambda=1.3$, 0% EGR, recorded at 225.5 g/kWhr, whilst optimum GSFC of LIVO condition was measured at 231.1 g/kWhr for $\lambda=1.5$, 0% EGR, resulting in 2.4% difference between their most optimised points.

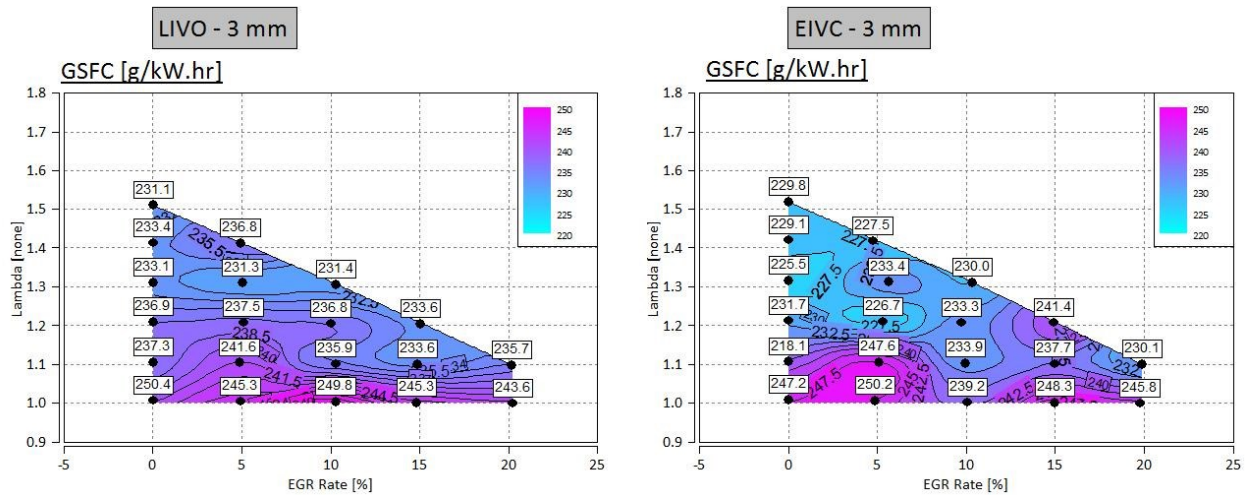


Figure 4.19 – GSFC comparison between 3mm LIVO and 3mm EIVC cases at 1500 RPM, 3.6 bar GMEP

Emissions formation was analysed for two species (eliminating CO and CO₂ based on the previous discussion and results presented in Chapter 3 for DVA vs SVA) – unburned hydrocarbons and NO_x. Maps plotted with EGR Rate on x-axis and Lambda on the y-axis (Figure 4.20 and 4.21) clearly represented trade-off between these two species. Area of the lowest THC production was also characterised by the highest NO_x emissions – lambda [1.1, 1.2] without any EGR. It could be attributed to high stability, high temperature region with excess oxygen and complete, efficient combustion taking place. Whilst the minimum THC was about 6% lower for EIVC at 1642 ppm, other points on the map tended to be characterised by fractionally higher production of that pollutant.

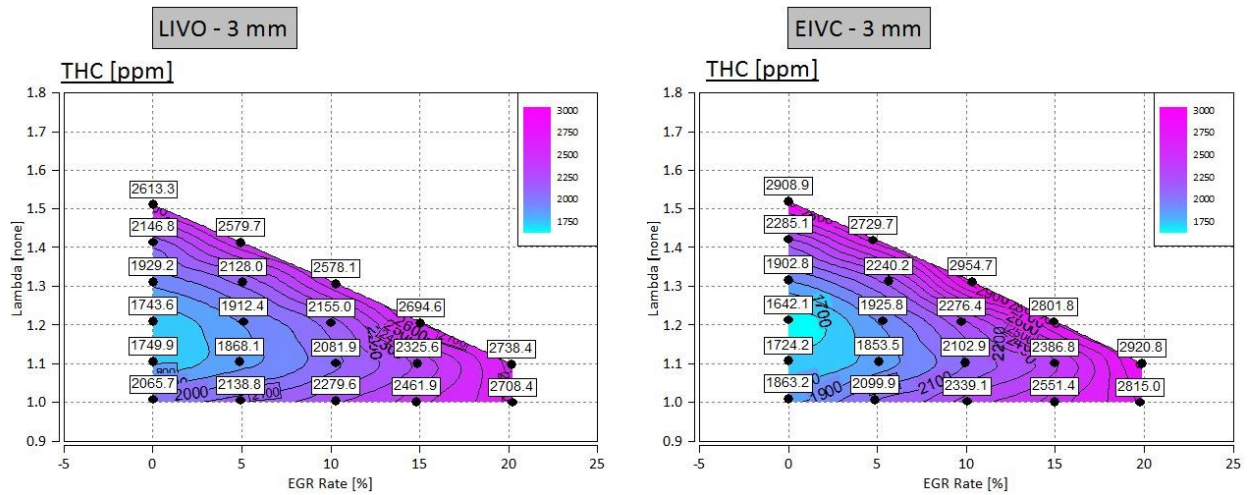


Figure 4.20 – Unburned hydrocarbons emissions comparison between 3mm LIVO and 3mm EIVC at 1500 RPM, 3.6 bar GMEP

NO_x emissions appeared to be lower for EIVC rather than LIVO strategy used on the second valve (up to 15%) and these differences should be linked to mixture preparation differences coming from significantly different in-cylinder motion. These differences are studied in more detail later in this chapter for 1500 RPM, 7.9 bar GMEP case.

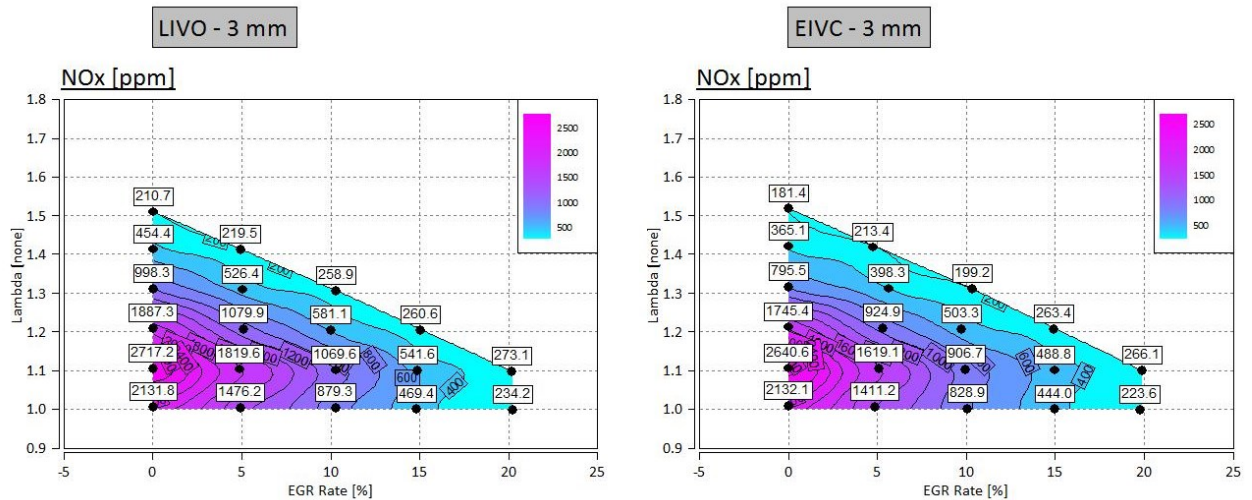


Figure 4.21 – NO_x emissions comparison between 3mm LIVO and 3mm EIVC at 1500 RPM, 3.6 bar GMEP

To sum up 1500 RPM, 3.6 bar GMEP condition, a plot showing maximum GSFC reduction (relative to stoichiometric DVA, 0% EGR) achieved for each strategy run on SCRE was

created. It can be seen in Figure 4.22 that points which exhibited the biggest fuel consumption reduction form an “envelope”, where utilising SVA was subjected to the highest overall dilution (highest lambda values for any given EGR rate), whilst optimum DVA points sat much lower in the plot (0.2 - 0.3 lambda difference for all EGR levels). 3 mm LIVO points were much closer to SVA with the biggest difference of lambda 0.1 at 5% EGR. 3 mm EIVC started at the same optimum dilution as DVA for 0% EGR, and as the EGR rate increased, it moved closer to LIVO and SVA profiles.

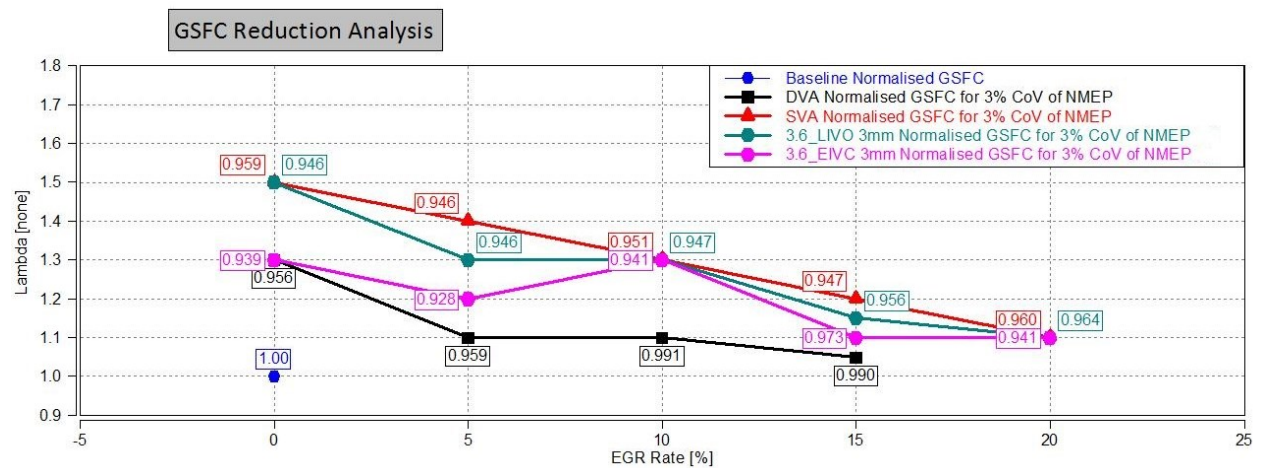


Figure 4.22 – Maximum GSFC reduction achieved for DVA, SVA, 3mm LIVO and 3mm EIVC normalised to baseline stoichiometric, 0% EGR DVA GSFC at 1500 RPM, 3.6 bar GMEP

These trends were in-line with the flow field within the cylinder and TKE at ignition, which was better matched to SVA for LIVO and DVA for EIVC.

The highest GSFC reduction overall for each condition was summarised in Table 4.2. 3 mm EIVC came out as the optimum condition that achieved 7.2% GSFC reduction, followed by LIVO and SVA at 5.4% and DVA at 4.4%. These result show that hypothesis stated in conclusions of Chapter 3 was successfully validated. SVA and DVA were optimised by the use of second valve operated in staggered lift mode. Opening second valve in 3mm LIVO mode achieved reduction in heat transfer losses (compared to SVA), whilst largely keeping the benefits of increased TKE at ignition and faster burn duration. These changes proved to balance each other, as the optimum GSFC for SVA and 3 mm LIVO remained unchanged.

Similar analogy can be noted for 3 mm EIVC and DVA. In this case small heat transfer penalty of closing second valve early was relatively small, whilst increased in-cylinder motion resulted

in higher TKE, especially for advanced ignition at diluted conditions. It achieved the highest GSFC benefit of all strategies tested at 1500 RPM, 3.6 bar GMEP.

Table 4.2 – Summary of the highest GSFC benefit achieved for all tested valve strategies at 1500 RPM, 3.6 bar GMEP

Valve Strategy	Lambda [-]	EGR Rate [%]	Benefit achieved over stoichiometric DVA [%]	Penalty over the best condition overall [%]
DVA	1.3	0	4.4	3.0
SVA	1.4	5	5.4	1.9
3mm LIVO	1.5; 1.3	0; 5	5.4	1.9
3mm EIVC	1.2	5	7.2	-

All the analysis up to date was done using the gross metrics (excluding pumping contributions). For completeness of the analysis conditions summarised in Table 4.2 were subjected to indicated overall thermal efficiency (fuel conversion efficiency) calculation outlined in Equation 4.1 (Heywood, 1988).

$$\eta_f = \frac{W_c}{m_f Q_{HV}}, \text{ where} \quad (4.1)$$

W_c indicated work output of engine cycle¹

m_f mass of fuel

Q_{HV} fuel heating value²

Calculating overall thermal efficiency enabled quantifying losses associated with pumping loops for all tested points and presented how effective each strategy was in converting chemical energy of the fuel into useful work output. It needs to be remembered that this was low speed and load, throttled point, hence overall efficiency of such operation is not close to the optimum range of 35-40% characterising most efficient running conditions of modern DISI engines.

¹ W_c was calculated using NMEP of given condition

² Fuel heating value was represented by Net Heat of Combustion [MJ/kg] supplied by Haltermann Carless in Certificate of Analysis delivered for every batch of fuel used.

Table 4.3 – Thermal efficiency results achieved for the GSFC optimised conditions at 1500 RPM, 3.6 bar GMEP

Valve Strategy	Lambda [-]	EGR Rate [%]	η_f [%]	$\Delta\eta_f$ vs the best η_f achieved [%]
DVA	1.3	0	31.38	3.6
SVA	1.4	5	32.07	1.2
3mm LIVO	1.5; 1.3	0; 5	32.07	1.2
3mm EIVC	1.2	5	32.56	-
DVA	1.0	0	30.41	6.6

It should also be noted that subjected to fuel conversion efficiency calculation, SVA and LIVO achieved the same result (to 2nd d.p.), however calculations were based on different NMEP and fuel flow data (3.084 bar NMEP and 0.519867 kg/hr fuel flow vs 3.098 bar NMEP and 0.522243 kg/hr fuel flow for SVA and 3 mm LIVO respectively). Results shown in Table 4.3 concluded that use of advanced valvetrain strategy at low speed and load condition was a viable mean of increasing engine thermal efficiency by up to 6.6% compared to stoichiometric DVA and 3.6% for optimised DVA running under diluted conditions.

4.2.2. 1500 RPM, 7.9 bar GMEP

4.2.2.1 LIVO Operation, Lambda Study

Increasing the load to 7.9 bar GMEP at 1500 RPM, presented increased stability of combustion and there was noticeable difference in combustion metrics compared to lower load. New clustering can be seen in Figures 4.23 and 4.24 representing combustion stability and burn durations. DVA's lambda sweep presented much less stable combustion and violated 3% CoV on NMEP (NVH criteria) at around $\lambda=1.35$, whilst all the LIVO (1-5 mm max lift) cases and SVA comfortably reached dilution of $\lambda=1.6$. Combustion durations were also significantly longer for DVA than for all the other LIVO and SVA cases in both ignition to 10% MFB and 10 to 90% MFB stages. Offset was fairly constant at 8-10°CA for initial stage of combustion and ~3°CA in the 10-90% MFB. Burn durations of SVA and LIVO lifts of 1-4 mm max lift on the second valve were indistinguishable from one another and they clustered onto a single line.

Only applying maximum lift of 5 mm presented deteriorated performance in speed of combustion. It proved that as the load increased, engine performance became less sensitive to late intake valve opening of up to 5 mm. It can be attributed to higher jet velocities in the early phases of induction stroke and strong turbulence formation that only started to be weakened by advancing opening of the second valve past 270°CA bTDC_f for the 5 mm scenario which was slightly more than halfway through induction stroke (part of the cycle, where piston velocity was the highest).

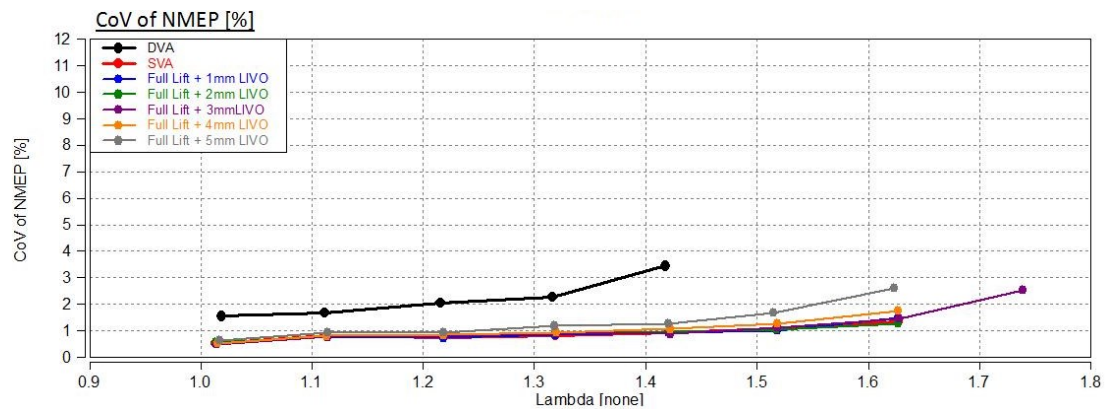


Figure 4.23 - Combustion stability of LIVO cases for lambda sweep at 1500 RPM, 7.9 bar GMEP

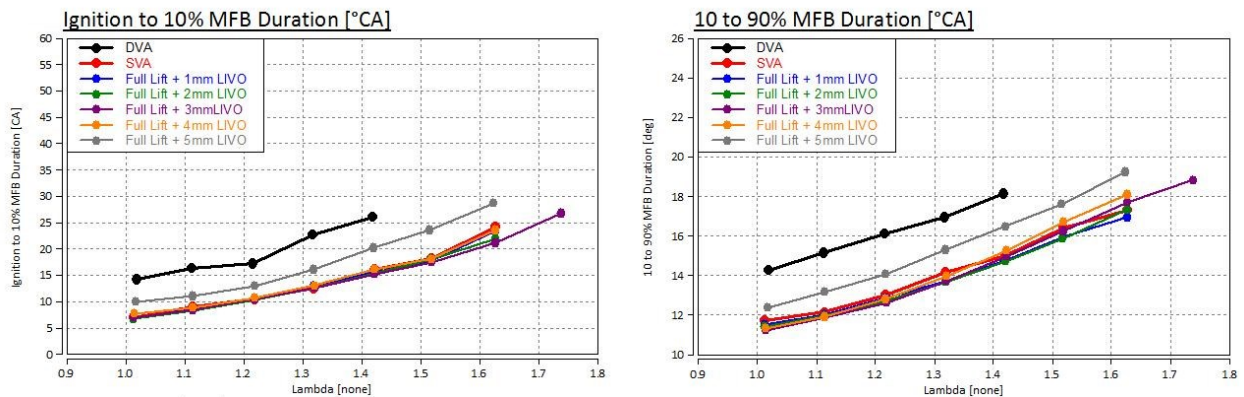


Figure 4.24 - Burn duration analysis of LIVO cases for lambda swings at 1500 RPM, 7.9 bar GMEP

Such behaviour was also represented by TKE shown in Figure 4.25. DVA's peak TKE value ($350 \text{ m}^2/\text{s}^2$) was only half of that reached for all the other cases. Then past 270°CA, it can be seen that due to opening of the second valve, line representing 5 mm LIVO case started dropping faster than other LIVO cases and SVA. In the ignition zone, it can be noted that SVA

and 1-3 mm lift cases were ignited at almost the same TKE of about $25 \text{ m}^2/\text{s}^2$, whilst 5 mm LIVO lift and SVA had $22 \text{ m}^2/\text{s}^2$ and $19 \text{ m}^2/\text{s}^2$ respectively.

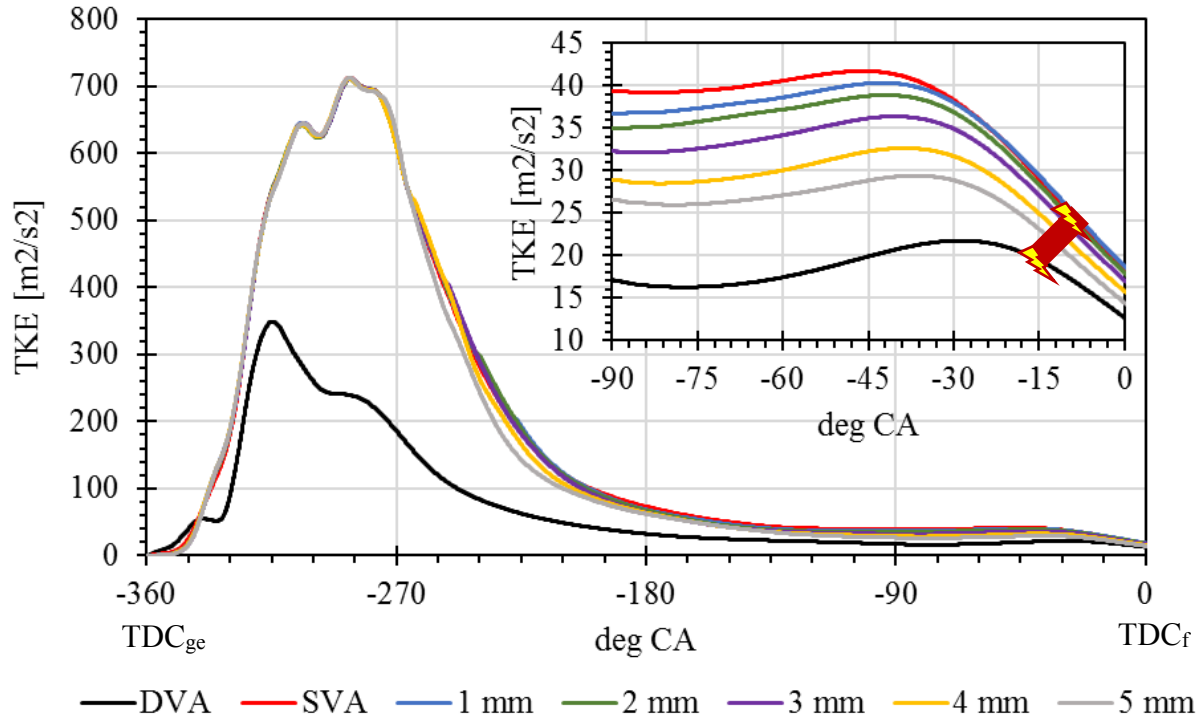


Figure 4.25 - TKE comparison between various max lift LIVO cases at 1500 RPM, 7.9 bar GMEP, $\lambda=1.0$ with a zoom on mid-stroke compression to TDC region

Fuel consumption measured as GSFC, was not significantly different to trends already seen and described for lower load LIVO lambda sweep. Again, 3 mm max lift LIVO case on the second lift provided most stable performance and consistently low fuel consumption between lambda 1.5 and 1.6. It outperformed DVA by 3%, however did not achieve any quantifiable benefit compared against SVA (difference of 0.1% that was within the measurement error).

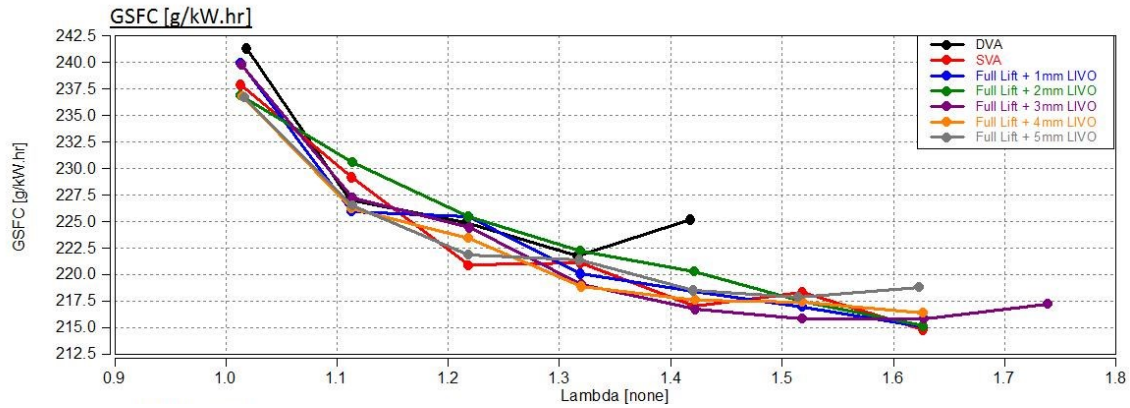


Figure 4.26 - GSFC of LIVO cases for lambda sweep at 1500 RPM, 7.9 bar GMEP

Heat transfer plot portraying its development during compression stroke (Figure 4.27) was qualitatively almost identical to Figure 4.8 produced for 1500 RPM, 3.6 bar GMEP. Knowing the results of heat transfer analysis and sensitivity study presented in Chapter 3, it can be said that quantitatively differences have other origin i.e. flow field effects are more pronounced than temperature differences, however for the purpose of assessing staggered lift using the LIVO strategy, it can be said that trends had little sensitivity to load and LIVO results were still better matched to SVA, showing that significant heat transfer penalty was associated with delayed opening of the second valve. Using the same methodology introduced previously for 1500 RPM, 3.6 bar GMEP case, relative heat transfer deltas were measured at 9%, 15%, 22%, 31% and 43% for lift of 1, 2, 3, 4 and 5 mm respectively. These differences were noticeably larger for 1-3 mm LIVO lifts compared to 3.6 bar GMEP load case, but very similar for 4 mm and 5 mm. Similarly to lower load scenario, gas temperatures of LIVO cases were well matched to SVA with the differences contained to 1K (whilst DVA was 25K cooler).

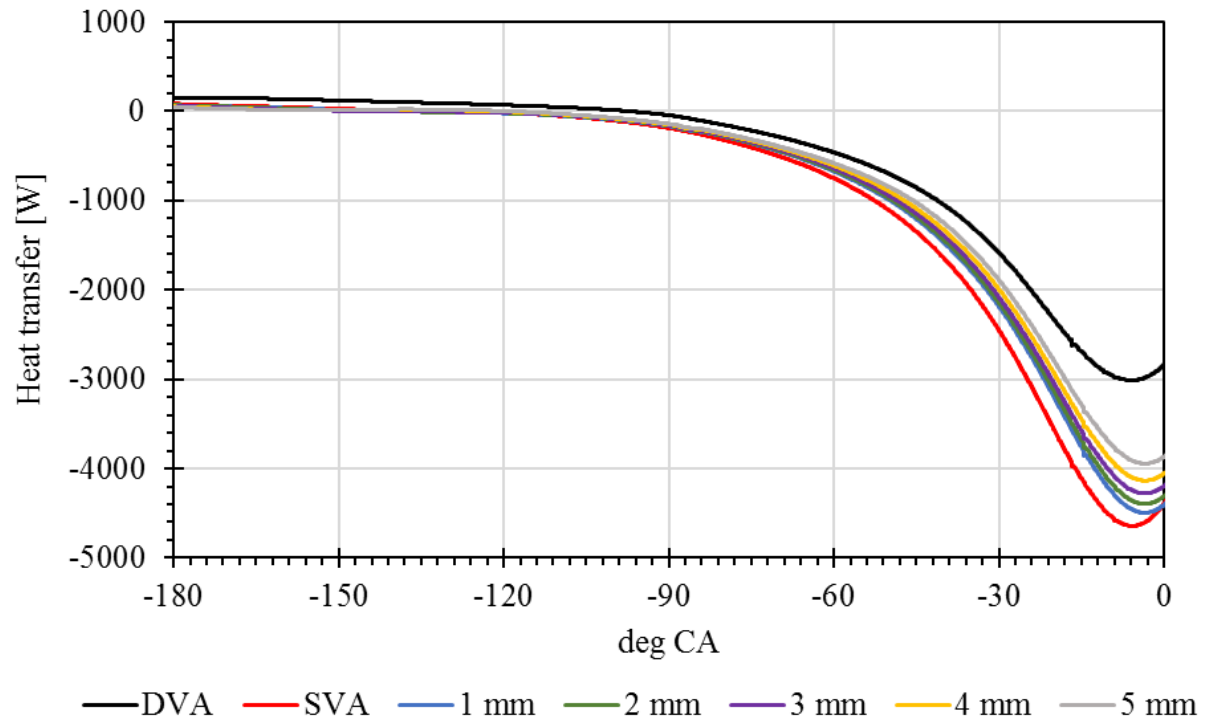


Figure 4.27 - Heat transfer comparison between various max lift LIVO cases at 1500 RPM, 7.9 bar GMEP, $\lambda=1.0$

Emissions formation analysis of lambda sweep at 1500 RPM, 7.9 bar GMEP (Figure 4.28) showed that introducing staggered lift in LIVO mode on the second valve had a sizeable impact on production of unburned hydrocarbons, and fairly limited impact on NO_x emissions. General shapes of the plots have already been discussed in Chapter 3, thus only differences between various valve lifts are subject of discussion. THC production represented trend, where maximum production was achieved under DVA operation, whilst minimum was obtained for SVA. As the opening of the second valve was increasing, the THC production was also increased, however it was not linear. SVA, 1 mm and 2 mm LIVO valve lifts were characterised by similar production levels, whilst increasing the lift by another 1 mm, started resulting in more substantial increases. It is believed that such behaviour was mainly linked to mixture formation and its homogeneity.

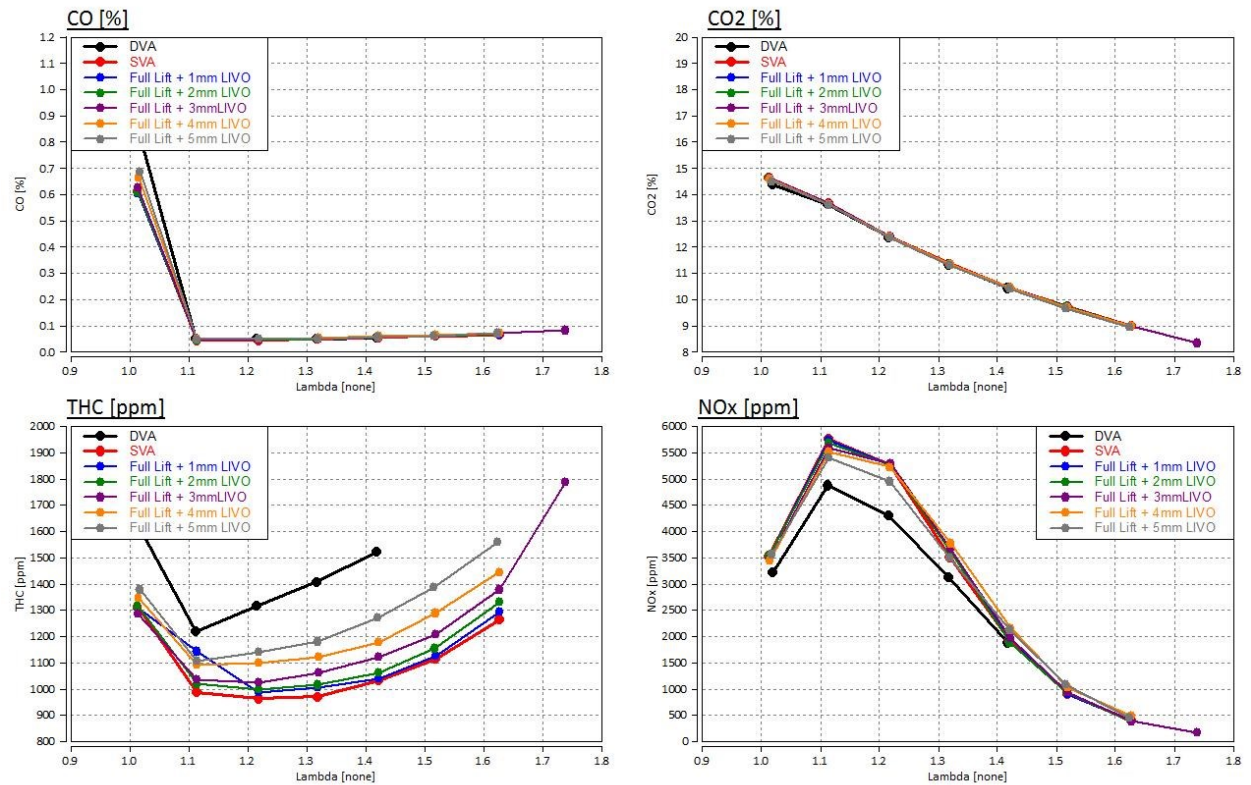


Figure 4.28 - Emissions formation comparison for lambda sweep between various max lift LIVO cases at 1500 RPM, 7.9 bar GMEP

To confirm the hypothesis related to homogeneity of the mixture, Homogeneity Index (introduced in Chapter 3) was plotted between 60°CA bTDC_f and 20°CA aTDC_f (Figure 4.29). It was in-line with the previously set expectations, given the very progressive, almost step-like, behaviour observed earlier for tumble, swirl, TKE, and now HI. Another factor which is believed to have positive impact on reducing the THC formation, was the fact that combustion started later within the cycle for LIVO cases and SVA (progressively), thus homogeneity was also increasing with time, making the delta with DVA even larger.

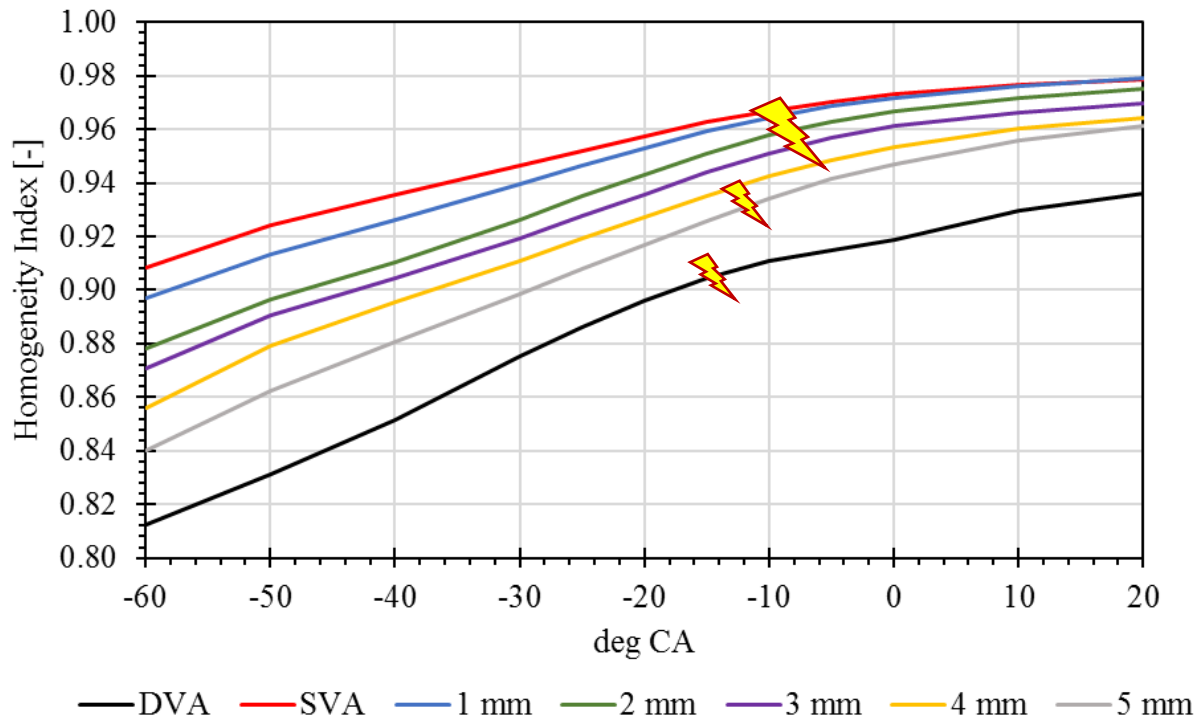


Figure 4.29 – Homogeneity Index comparison between various max lift LIVO cases for $\lambda=1.0$ at 1500 RPM, 7.9 bar GMEP

Fuel film on the piston was also scrutinised to see if there were any differences, however they were insignificant and mirrored SVA scenario (Figure 4.30). It was expected because injection preceded opening of the second valve (for all 1-5 mm lifts), hence spray plumes were subjected to flow field very closely matched to that of SVA during initial phase of fuel injection.

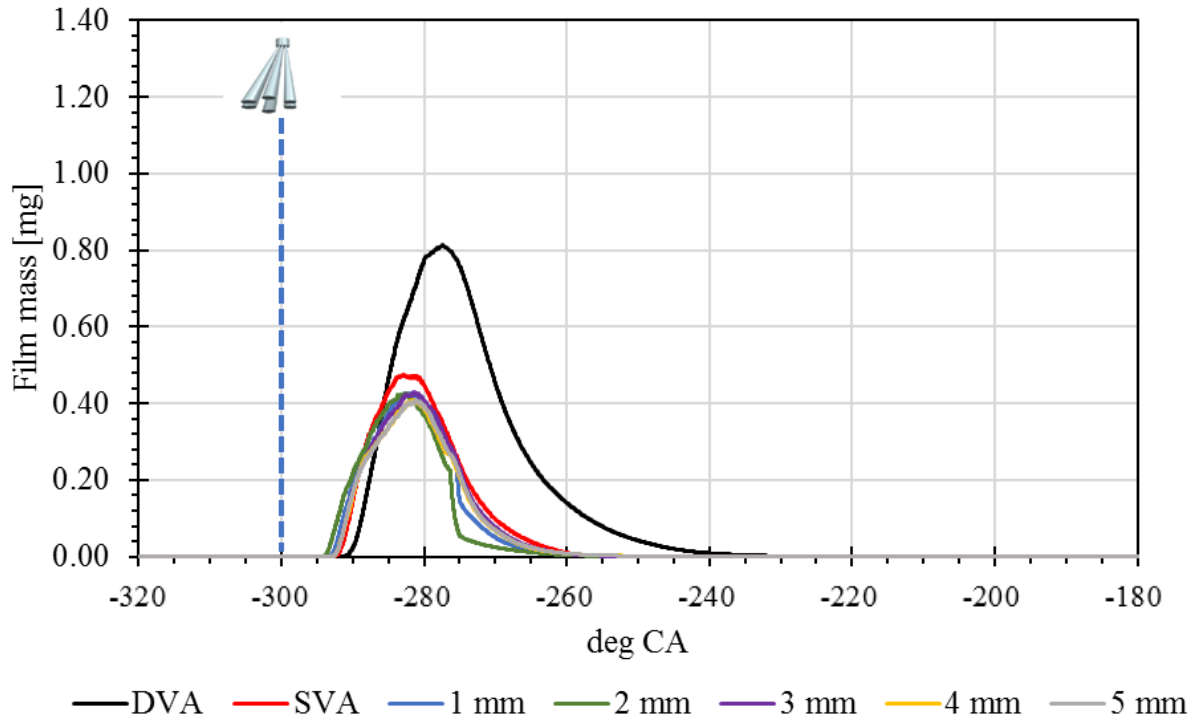


Figure 4.30 – Accumulated liquid fuel film mass on the piston surface comparison between various max lift LIVO cases for $\lambda=1.0$ at 1500 RPM, 7.9 bar GMEP

Regardless of the very small quantitative differences in NO_x production for all LIVO cases, methodology introduced in Chapter 3 was used, and plots representing amount of artificially introduced spheres taken by volume deemed the most favourable for NO_x production (λ [1.05, 1.15]) were produced (Figure 4.31). It should be noted that all lift cases were characterised by well-matched overall in-cylinder (depicted with black lines in Figure 4.31) values showing how much of the volume was occupied by the specific mixture, especially in the 30 to 10°CA bTDC_f, where values were kept at 20-30%. Also 30 mm, 40 mm and 60 mm spheres exhibited fairly consistent behaviour throughout the investigated 50°CA range, being kept at around 30-50% and 20-40% respectively. Based on presented data, it can be said that small differences in NO_x production were expected, and experimental findings confirmed that, however it was duly noted that this analysis was only limited to λ study and was not extended to temperature distribution or flame propagation.

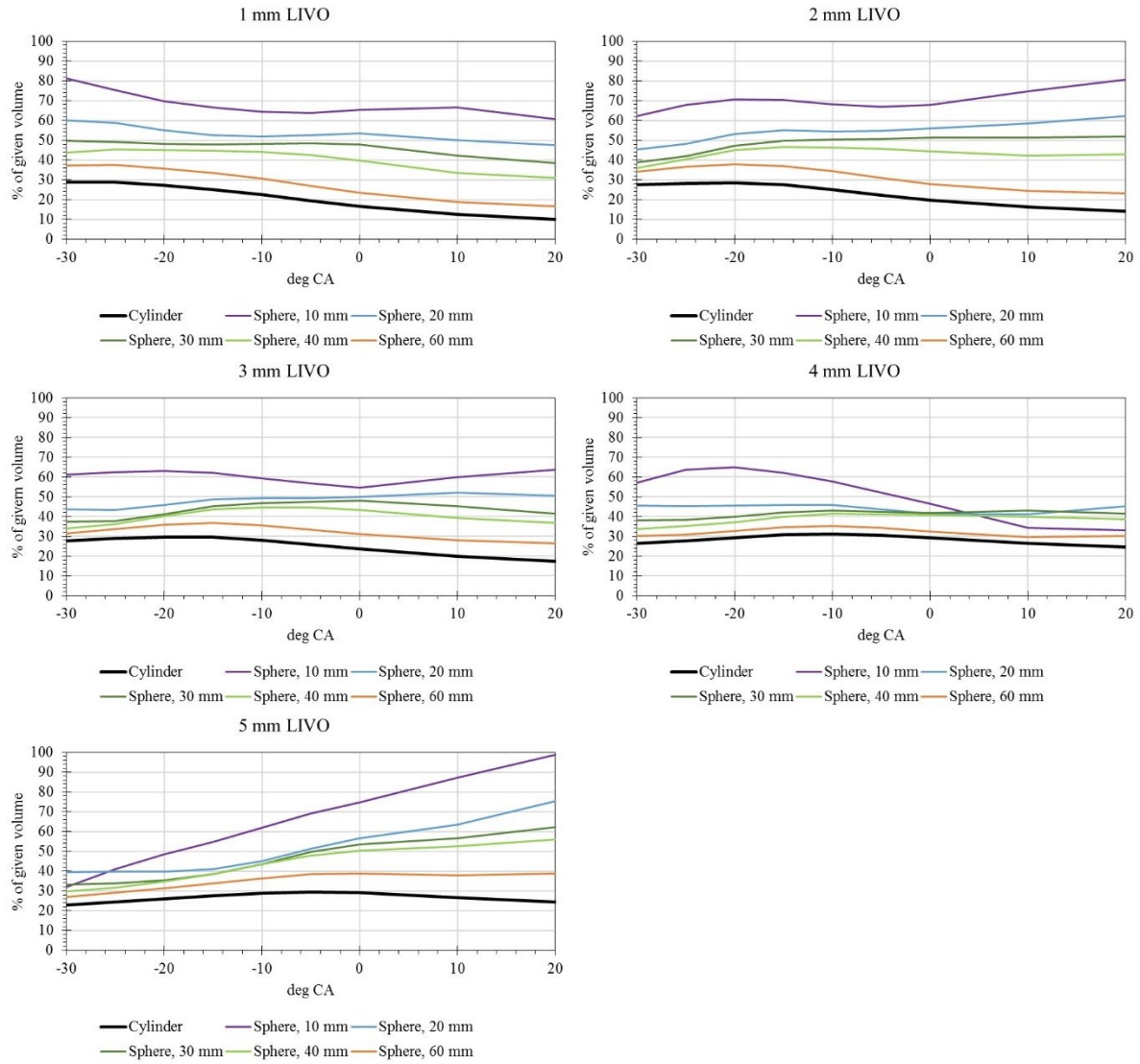


Figure 4.31 – Percentage of the cylinder and specific spherical volumes occupied by a mixture in the λ range of $[1.05, 1.15]$ for various max lift LIVO cases at global $\lambda=1.0$, 1500 RPM, 7.9 bar GMEP

4.2.2.2 EIVC Operation, Lambda Study

As per 3.6 bar GMEP condition, LIVO staggered lift study at 7.9 bar was complemented with EIVC lambda sweep. It was found that combustion stability was significantly better for all 1-5 mm max lift EIVC conditions when compared to DVA, and allowed for achieving greater air-fuel equivalence ratios. It was almost identical for that of SVA up to $\lambda=1.4$ (at which point DVA already exceeded 3% CoV of NMEP limit) and kept at around 1% CoV of NMEP. Increasing dilution to $\lambda=1.5$, started showing deteriorated stability for the highest (5 mm) EIVC lift, whilst additional 0.1 increase in lambda saw 4 mm max lift separated from the main cluster of lower lifts and SVA presenting best stability (presented in Figure 4.32).

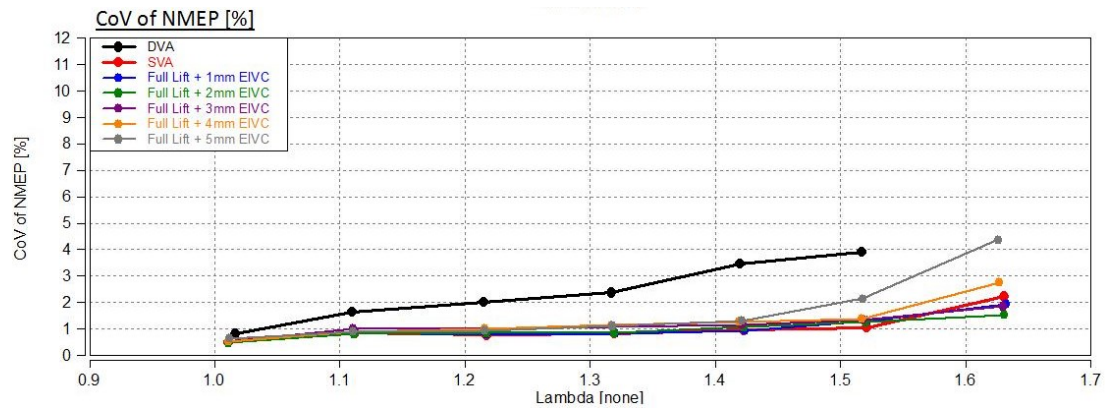


Figure 4.32 - Combustion stability of EIVC cases for lambda sweep at 1500 RPM, 7.9 bar GMEP

Combustion durations (Figure 4.33) were closely matched to results shown in section 4.2.1.2 for lower load scenario. Distinguishable cluster of SVA and 1 mm and 2 mm EIVC lifts for both initial and later stages of combustion was followed by separated 3 mm EIVC case that still outperformed DVA and realised faster combustion. Once the lift of second valve was increased to 4 mm and 5 mm, combustion durations were almost identical as the ones observed for DVA. Again, majority of the difference in combustion durations was attributed to its first 10% MFB, ranging from 7 °CA at stoichiometric conditions (SVA and 1-2 mm EIVC vs DVA and 4-5 mm EIVC) to 12°CA at $\lambda=1.5$. At the points, 10 to 90% MFB offset remained constant at 2-3°CA regardless of increasing air-fuel equivalence ratio.

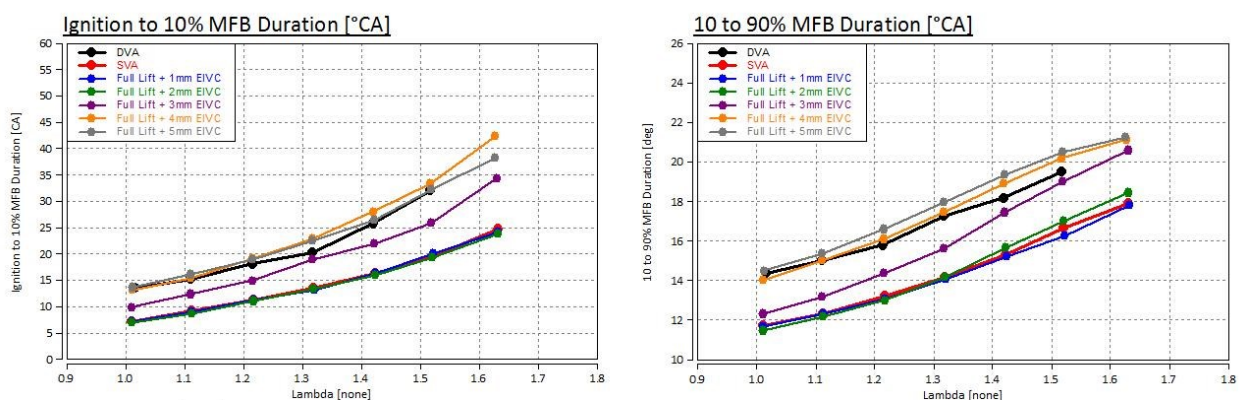


Figure 4.33 - Burn duration analysis of EIVC cases for lambda swings at 1500 RPM, 7.9 bar GMEP

TKE of 1500 RPM, 7.9 bar GMEP running with second lift in the EIVC mode showed trends similar to the ones previously observed in Figure 4.14 for 3.6 bar GMEP load, especially in the

ignition region. There were quantitative differences between the two, as increased load (and plenum pressure) led to values being higher by about 15%, however qualitatively same clustering linked to combustion metrics was shown. Fast burning SVA, 1 mm and 2 mm lift EIVC cases represented the highest TKE at ignition, forming a cluster at around $23\text{--}24\text{ m}^2/\text{s}^2$, whilst DVA and 3 mm case recorded about $18\text{--}19\text{ m}^2/\text{s}^2$. 4 mm and 5 mm cases were also very close together at approximately $17\text{ m}^2/\text{s}^2$ at ignition. Interestingly, these were almost identical to the values noted for 3.6 bar GMEP case, due to more advanced ignition for higher load.

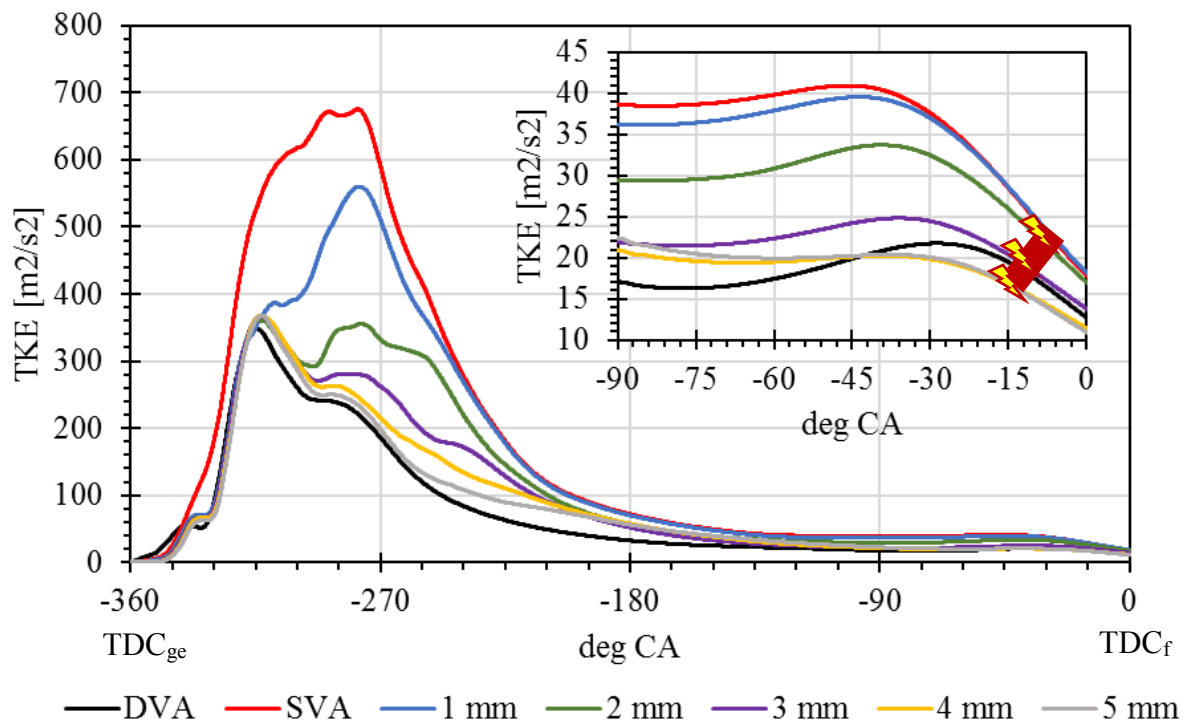


Figure 4.34 - TKE comparison between various max lift EIVC cases at 1500 RPM, 7.9 bar GMEP, $\lambda=1.0$ with a zoom on mid-stroke compression to TDC region

Fuel consumption measured using the EIVC staggered lift strategy, revealed that 2 mm, 3 mm and 4 mm max lift cases performed better than others, whilst all tested conditions brought noticeable improvement over standard DVA operation. Maximum reduction of 2.3% was achieved by the three aforementioned, most optimised conditions at $\lambda=1.5$ when compared to DVA of equivalent dilution, and nearly 10% when compared to stoichiometric, baseline DVA point.

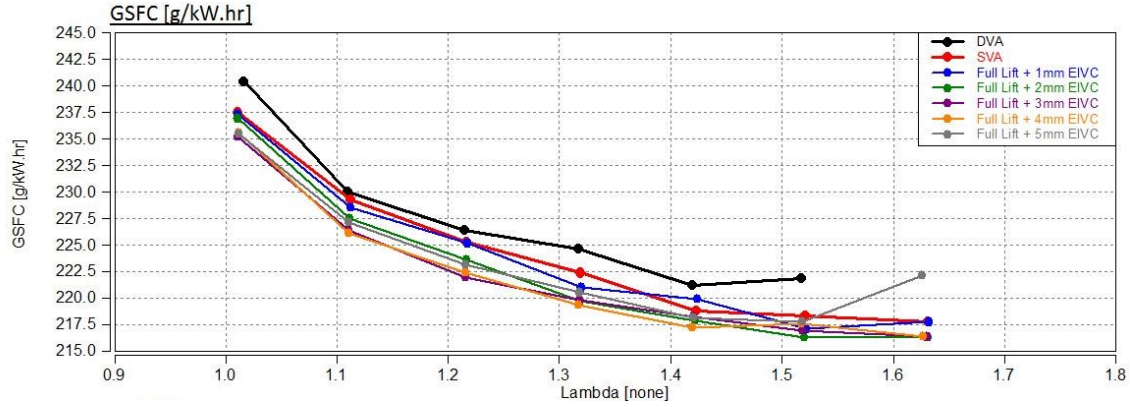


Figure 4.35 - GSFC of EIVC cases for lambda sweep at 1500 RPM, 7.9 bar GMEP

Having already extensively established in the previous analysis that applying any valve lift strategies other than the standard DVA carries penalty in heat transfer, it was quantified for 1500 RPM, 7.9 bar GMEP. Heat transfer profiles presented in Figure 4.36 were very similar in their nature to the ones shown for lower load in Figure 4.16. As the lift on the second valve was increased, gradual (step-like) increase in heat transfer took place. The in-cylinder temperatures varied by 25K between DVA and SVA, whilst 1 mm EIVC was only 4K lower than SVA, and opening second valve up to 5 mm resulted in the maximum difference of 8K (still relatively better matched to SVA than DVA). These reductions were higher than the ones observed for LIVO cases and should be attributed to higher fresh charge admission and hence more effective cooling in the initial stages of induction process (before second valve closed).

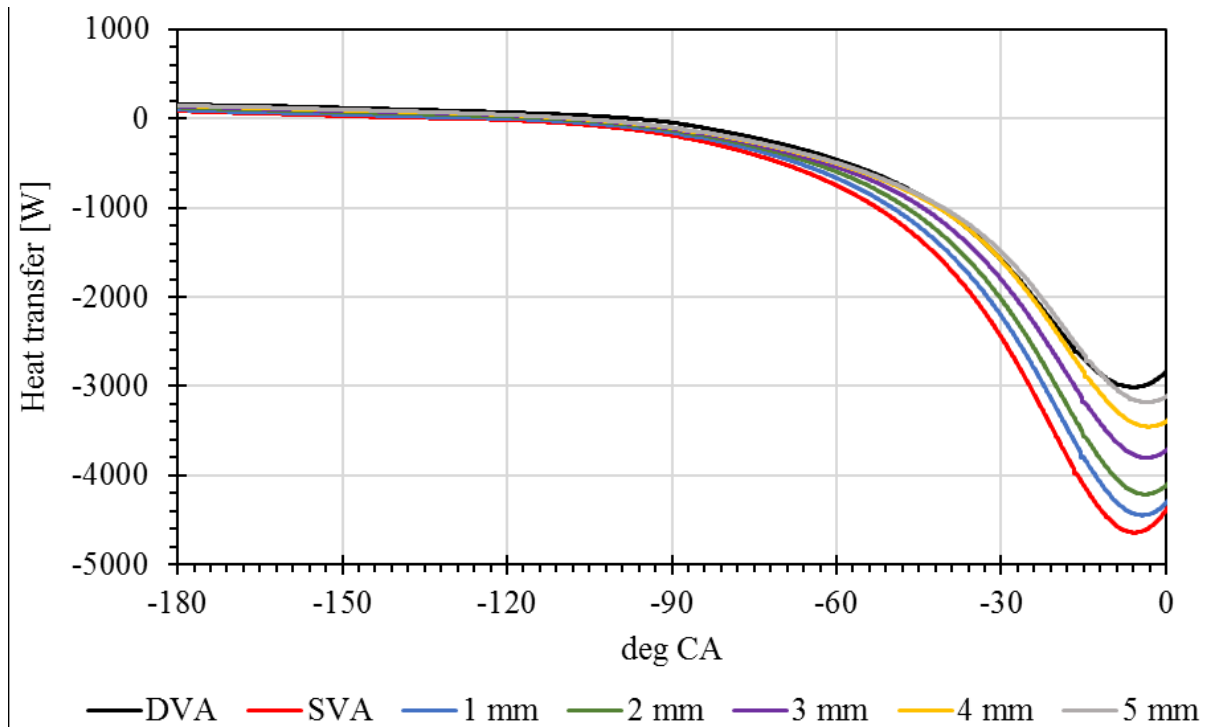


Figure 4.36 - Heat transfer comparison between various max lift EIVC cases at 1500 RPM, 7.9 bar GMEP, $\lambda=1.0$

Heat transfer differences between LIVO and EIVC were presented in Table 4.4. It should be noted that increasing the second valve lift past 2 mm exhibited large differences between two strategies. LIVO was better matched to SVA even at the maximum 5 mm tested case (43%), whilst EIVC for the same lift achieved reduction of 90%, being almost on par with DVA.

Table 4.4 - Heat transfer reduction vs SVA baseline comparison of LIVO and EIVC cases at various maximum lifts at 1500 RPM, 7.9 bar GMEP, $\lambda=1.0$

Maximum valve lift on second valve	Percentage decrease of heat transfer compared to SVA [%]	
	LIVO	EIVC
1 mm	9	12
2 mm	15	26
3 mm	22	52
4 mm	31	73
5 mm	43	90

Emissions formation of EIVC was mainly focused on THC and NO_x, as other species (CO and CO₂) did not present any noticeable differences when compared to LIVO cases presented in Figure 4.28.

THC production remained unchanged for 1 mm EIVC lift when compared to its LIVO counterpart, however any further increase in EIVC lifts were accompanied with noticeable increase in THC concentration in exhaust gases. Compared to LIVO, the increase of approximately 10%, 12%, 23% and 27% was recorded for 2 mm, 3 mm, 4 mm and 5 mm respectively. The big jump could be seen in moving to 4 mm and 5 mm lifts that were also characterised by slower burn durations and lower TKE. It is believed that altered charge motion and mixture preparation were responsible for such drastic increase in THC production.

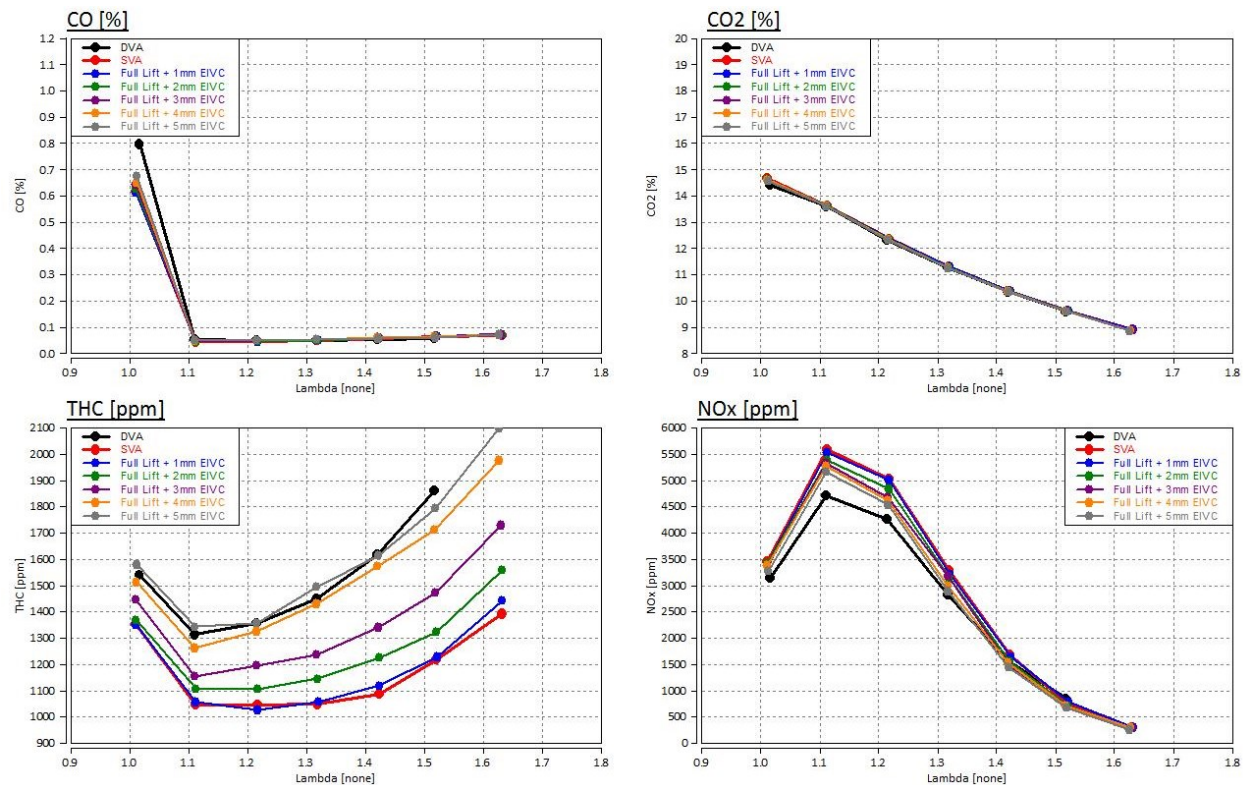


Figure 4.37 - Emissions formation comparison for lambda sweep between various max lift EIVC cases at 1500 RPM, 7.9 bar GMEP

Mixture preparation effects were again quantified using the Homogeneity Index (Figure 4.38). HI values and lack of clear trends suggested that THC emissions were more sensitive to TKE at ignition and speed of combustion rather than charge homogeneity. It was noted that differences in TKE and burn durations were relatively larger than differences in homogeneity

index (0.90 to 0.96 for all cases at ignition) and all conditions exhibited highly homogenous mixture.

Having analysed the HI, it was expected that fuel impingement on the piston might have changed due to introducing new induction method, where 1 mm and 2 mm EIVC staggered lift valvetrain strategies closed second valve prior to SOI. Fuel film mass on the piston surface was therefore plotted for all EIVC lifts (Figure 4.39).

Interestingly, there was a trend showing increased accumulation of fuel on the piston when lifts were increased from 2 mm to 5 mm, however 1 mm lift also presented high impingement which was not linked to increased THC formation in experimental data presented earlier. To further understand what the underlying cause of the THC emission trends might be (seen in Figure 4.37), spatial distribution of fuel film thickness on the piston was presented in Figure 4.40.

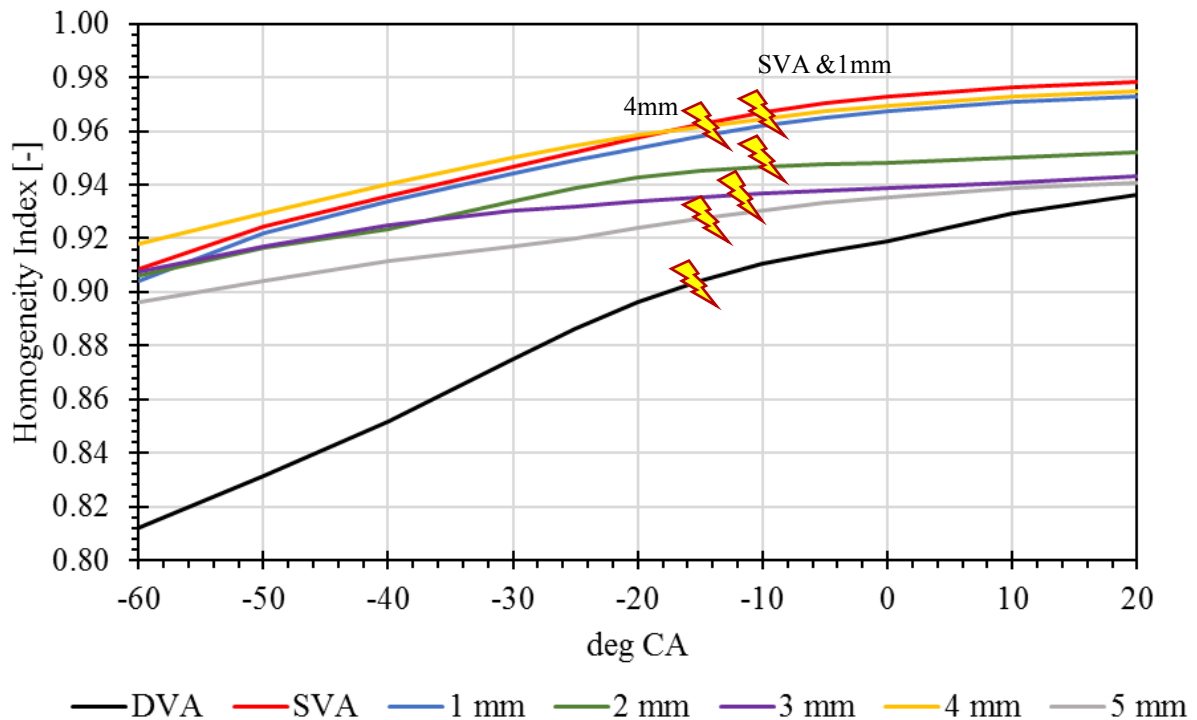


Figure 4.38 - Homogeneity Index comparison for lambda sweep between various max lift EIVC cases for $\lambda=1.0$ at 1500 RPM, 7.9 bar GMEP

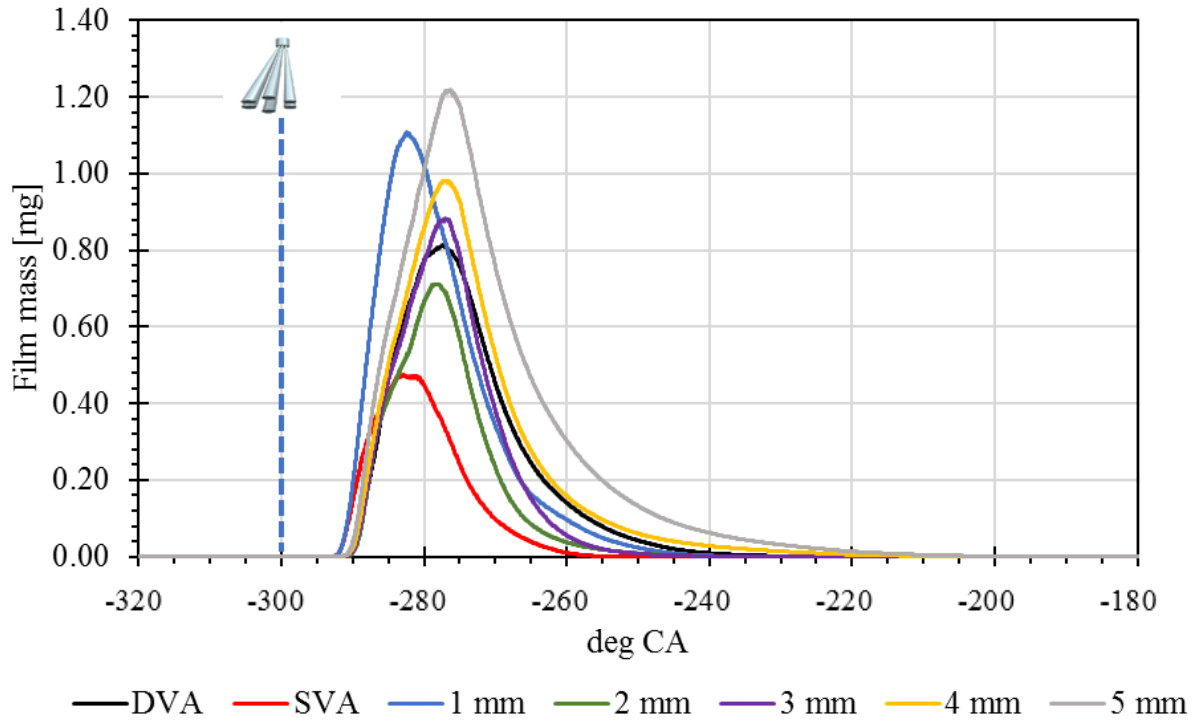


Figure 4.39 - Accumulated liquid fuel film mass on the piston surface comparison between various max lift EIVC cases for $\lambda=1.0$ at 1500 RPM, 7.9 bar GMEP

It can be seen that locations of fuel deposition points vary between 1 mm EIVC case and all the others. Whilst 1 mm lift resulted in hitting the piston on the intake side, for 4 mm and 5 mm maximum valve lifts, fuel impingement footprint showed fuel deposited mainly in the exhaust valve cut outs. These results can also be linked to much lower in-cylinder flow field for higher lifts, hence reducing evaporation rate of the fuel. It is believed that high initial impingement for 1 mm EIVC lift case was offset by much higher in-cylinder motion, whilst 4 mm and 5 mm lift cases exhibited much slower evaporation rate along the significantly weakened in-cylinder flow. It was also noted that (as per Chapter 3) simulations used single component fuel that evaporates much faster and were not fully representative of evaporation rates seen in experiments run on SCRE. It was only used as a tool which provided extra level of insight when comparing one valve lift to the other.

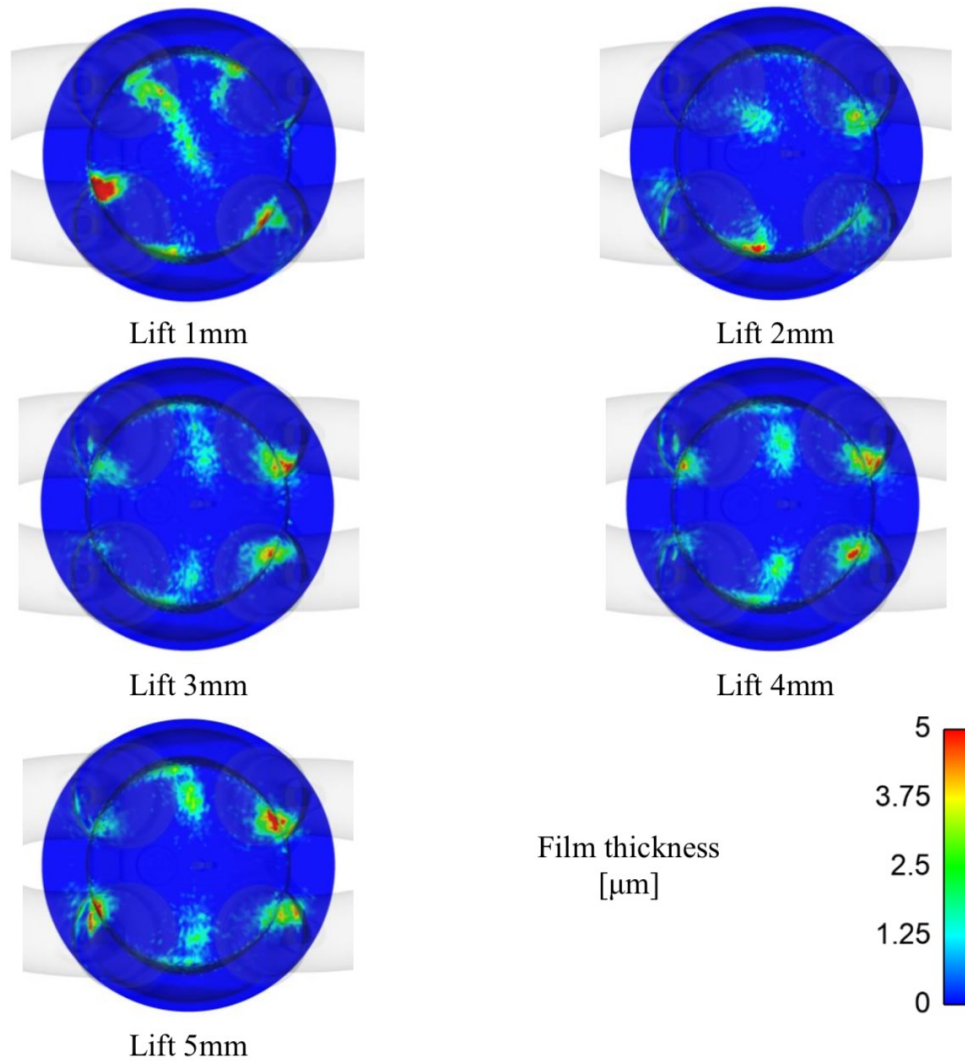


Figure 4.40 – Fuel impingement areas on the piston surface at 280°CA $b\text{TDC}_f$ comparison between various max lift EIVC cases for $\lambda=1.0$ at 1500 RPM, 7.9 bar GMEP (Kountouriotis, 2019)

NO_x concentration in the exhaust changed only by small margin for maximum lifts on the second valve of 3 mm, 4 mm and 5 mm when compared to LIVO strategy – it diverged further from SVA towards DVA for air-fuel equivalence ratios up to 1.2. Maximum and minimum production rates were attributed to SVA and DVA respectively for λ 1.0 to 1.3. After passing the threshold value of $\lambda=1.3$, all strategies converged onto a single line with negligible differences (shown in Figure 4.37). Analysis of the spheres being occupied by the λ range of [1.05, 1.15], assumed to be most favourable for NO_x production, for the five EIVC staggered lift cases did not follow any easily identifiable trend, especially when compared with Figure 4.31 (representing LIVO cases). It is believed that majority of the differences can be attributed to temperatures, as presented in Chapter 3. Earlier heat transfer discussion proved that in-

cylinder temperatures were more closely matched to SVA, especially for LIVO, and these trends agree with NO_x formation shown in Figures 4.28 and 4.37.

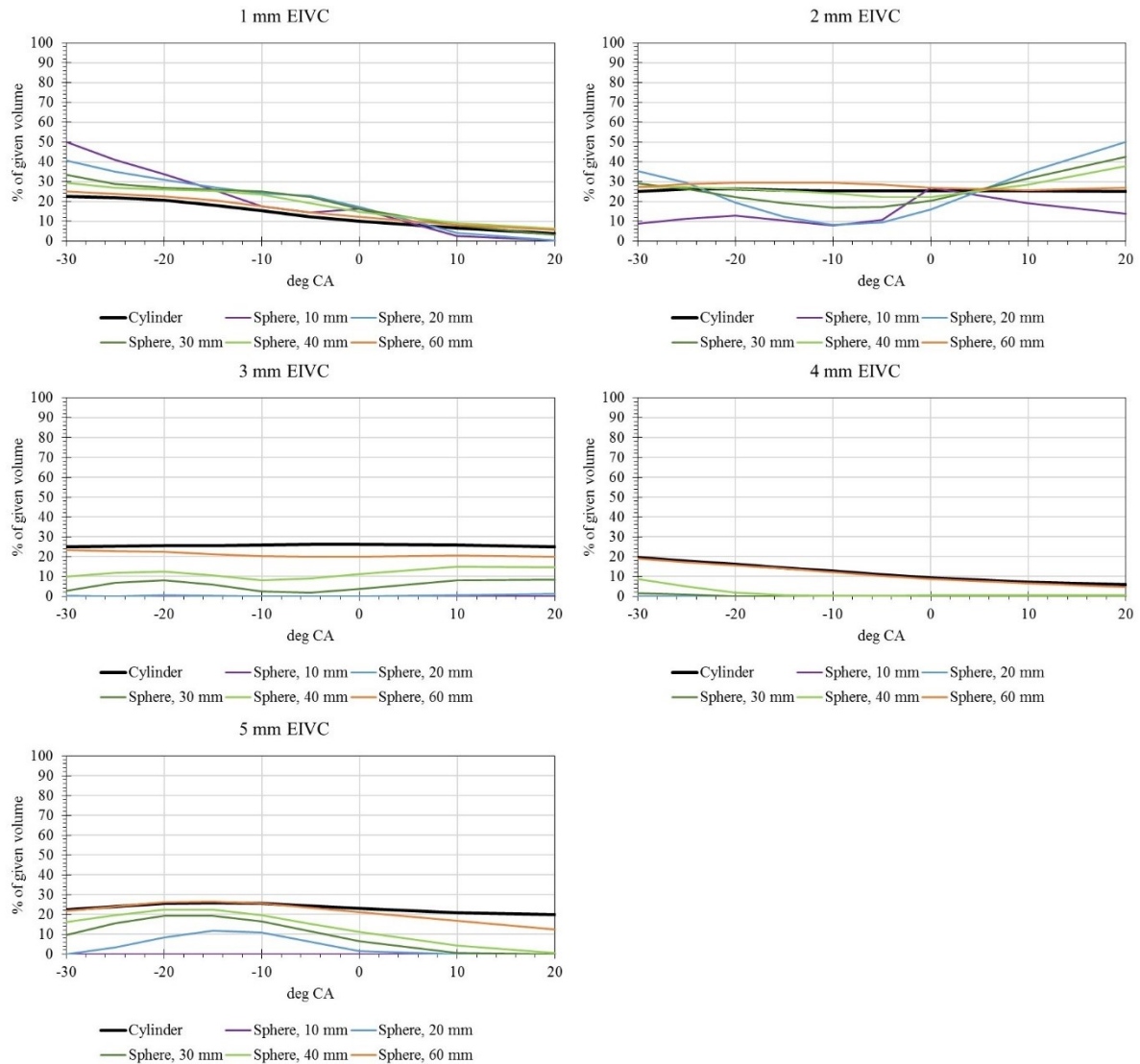


Figure 4.41 - Percentage of the cylinder and specific spherical volumes occupied by a mixture in the λ range of [1.05, 1.15] for various max lift EIVC cases at global $\lambda=1.0$, 1500 RPM, 7.9 bar GMEP

4.2.2.3 Optimum

Once again, having completed analysis of the lambda sweeps, points showing highest GSFC benefit and good combustion stability were chosen for more detailed analysis under excess air and EGR dilution. 3 mm max lift on the second valve was selected for both LIVO and EIVC cases. As per 3.6 bar GMEP experiments, collected data should only be analysed for points which did not breach the 3% CoV of NMEP criteria, hence in Figures 4.40 - 4.43 it can be seen

that two points of the LIVO dataset were marked with red and excluded from analysis (CoV of NMEP was measured at 3.5% and 3.9% for 5% EGR and 10% EGR points respectively).

First thing that should be noted is the difference in size of the contour plots presented in this section. Three more points (adhering to CoV of NMEP criteria) were recorded for LIVO, showing extended capabilities for tolerating more dilution when operated with this strategy, compared to EIVC.

Ignition advance was shorter for 3 mm LIVO case (vs 3 mm EIVC), and presented sensitivity to dilution very similar to the one already discussed for lower load scenario i.e. as the dilution increased, the delta in ignition angle and ignition to 10% MFB between LIVO and EIVC increased. Also, both strategies have shown that as the charge dilution was being increased, the initial burn duration (ignition to 10% MFB) and ignition advance started deteriorating quicker. It was shown graphically by decreased spacing between isolines in the contour plots.

Later stages of combustion (10-90% MFB duration) remained a constant, small offset between the two strategies for equivalent dilution levels at approximately 1.0-2.5°C.A. Rate of change with increasing dilution also remained almost constant, shown by equally spaced isolines in the last two contour plots in Figure 4.42.

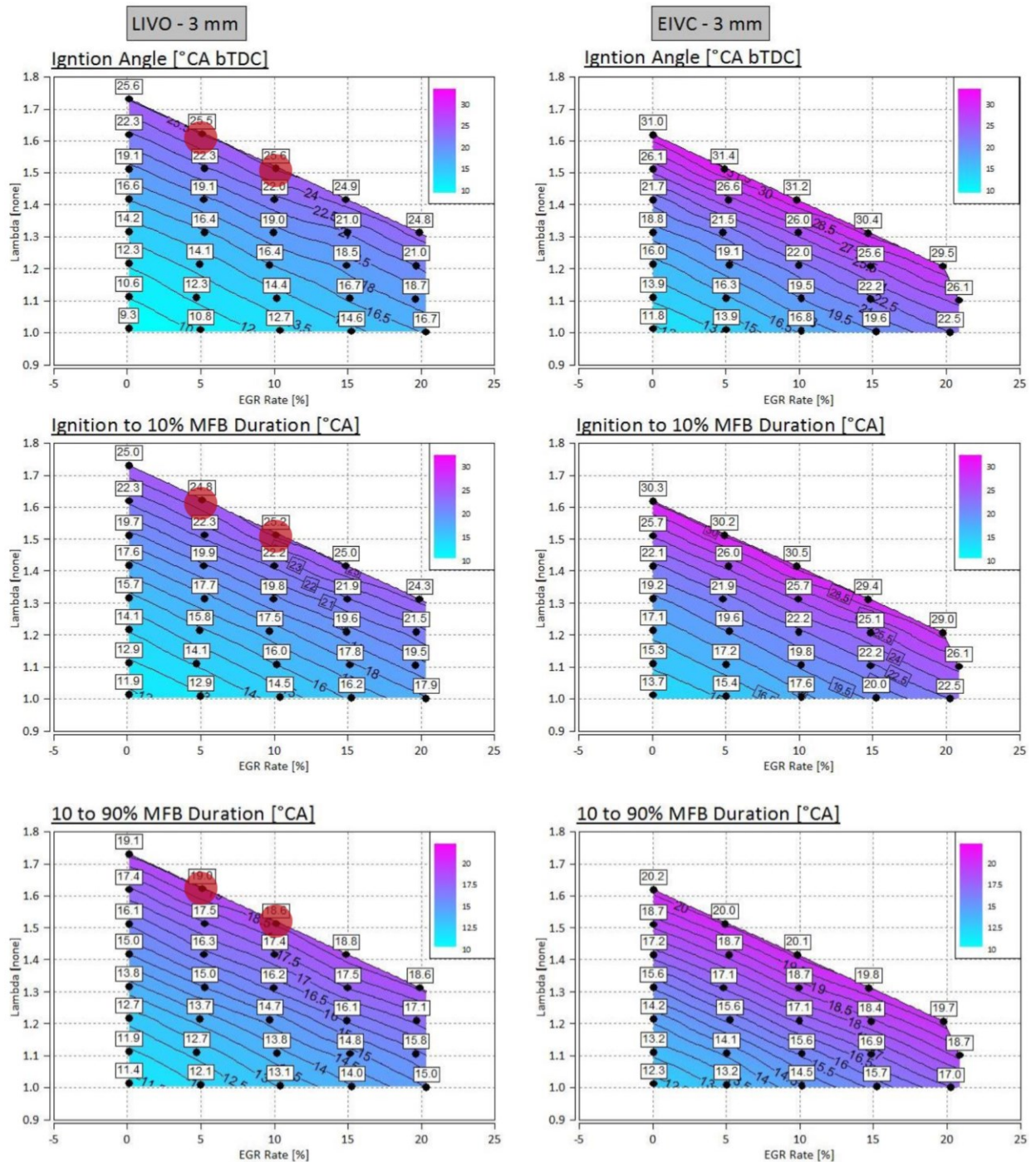


Figure 4.42 - Ignition angle, Ignition to 10% MFB and 10 to 90% MFB comparison between 3mm LIVO and 3mm EIVC at 1500 RPM, 7.9 bar GMEP

Increasing the load had an impact on fuel consumption characteristic and how the two staggered lift valve operating strategies stood against each other. They became much more closely matched for equivalent dilution (vs 1500 RPM, 3.6 bar GMEP). Colouring of the maps (Figure 4.43) representing GSFC indicated that the most optimised regions formed at low EGR

rates (0-5%), and high air dilution ($\lambda=1.4$ or higher). Minimum GSFC recorded for 3 mm LIVO case was 212.6 g/kWhr ($\lambda=1.6$, 0% EGR) vs 215.0 for 3 mm EIVC ($\lambda=1.4$, 5% EGR), showing the difference of just 1.1% in favour of LIVO.

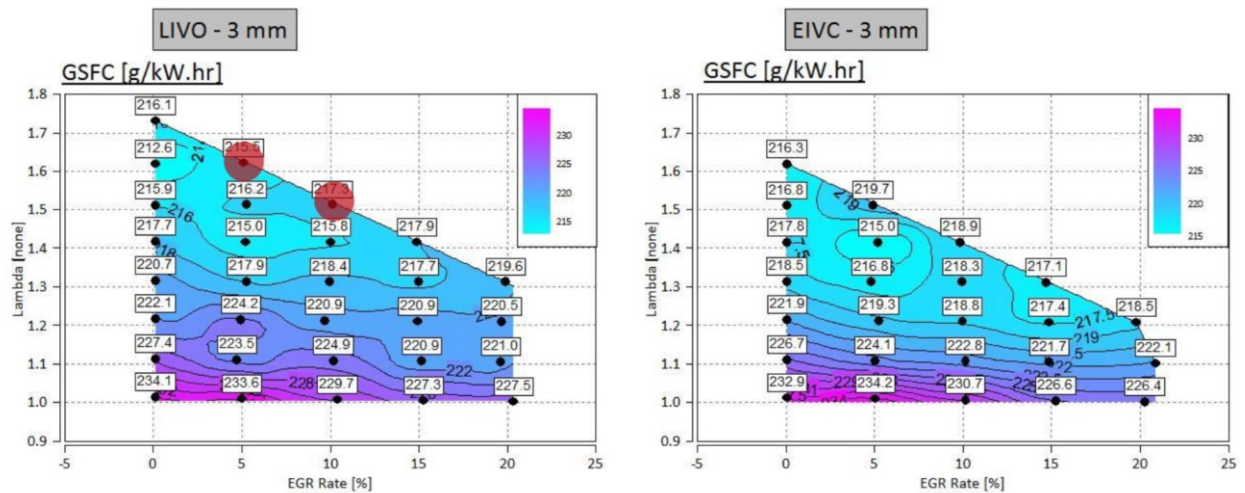


Figure 4.43 - GSFC comparison between 3mm LIVO and 3mm EIVC cases at 1500 RPM, 7.9 bar GMEP

Emissions formation of both THC and NO_x were lower for 3 mm maximum lift LIVO condition for points of equivalent dilution. General trends discussed for lower load case remained unchanged, however the absolute values of THC production lowered significantly, whilst NO_x raised throughout. Once points previously chosen on the basis of their GSFC benefit were confronted, LIVO presented itself with nearly 5% lower THC (1324 ppm vs 1392 ppm) and more importantly 49% lower NO_x (415 ppm vs 806 ppm).

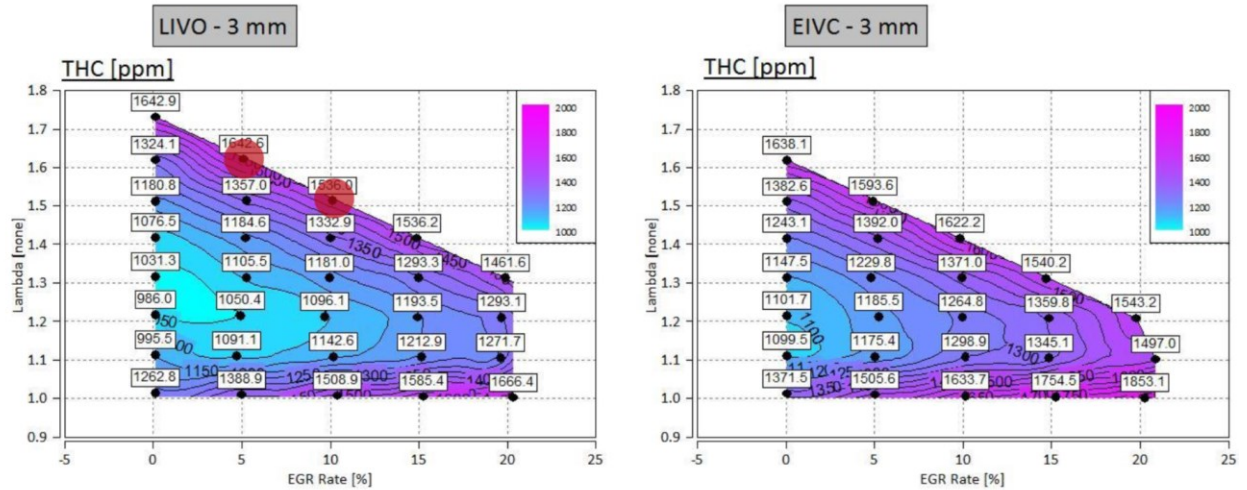


Figure 4.44 - Unburned hydrocarbons emissions comparison between 3mm LIVO and 3mm EIVC at 1500 RPM, 7.9 bar GMEP

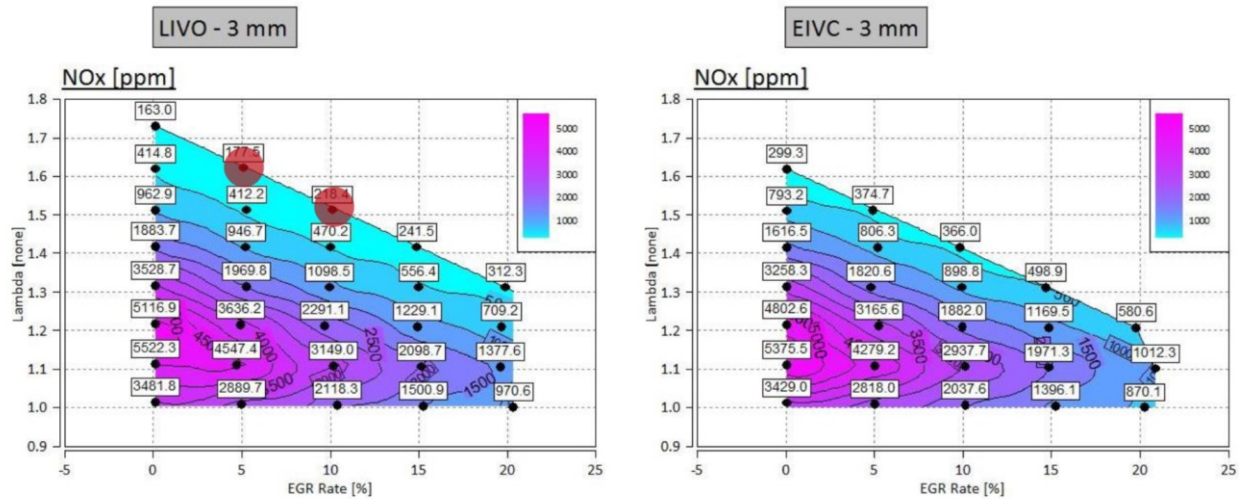


Figure 4.45 - NO_x emissions comparison between 3mm LIVO and 3mm EIVC at 1500 RPM, 7.9 bar GMEP

Summing up the 1500 RPM, 7.9 bar GMEP point subjected to staggered lift study on the second valve, maximum GSFC benefit relative to baseline, stoichiometric DVA operation at each EGR level was presented in Figure 4.46. As already seen in Figure 4.22, optimum points were achieved for the lowest overall dilution under DVA operation, however at increased load three other strategies (SVA, LIVO and EIVC) reached their optimum fuel consumption for almost the same lambda level, only varying by maximum of 0.1.

Optimum values were selected for each valvetrain operating mode and compared in Table 4.5. Increasing the load to 7.9 bar GMEP, resulted in the optimum strategy switching from 3 mm maximum lift in EIVC mode on the second valve at 3.6 bar GMEP to 3 mm LIVO.

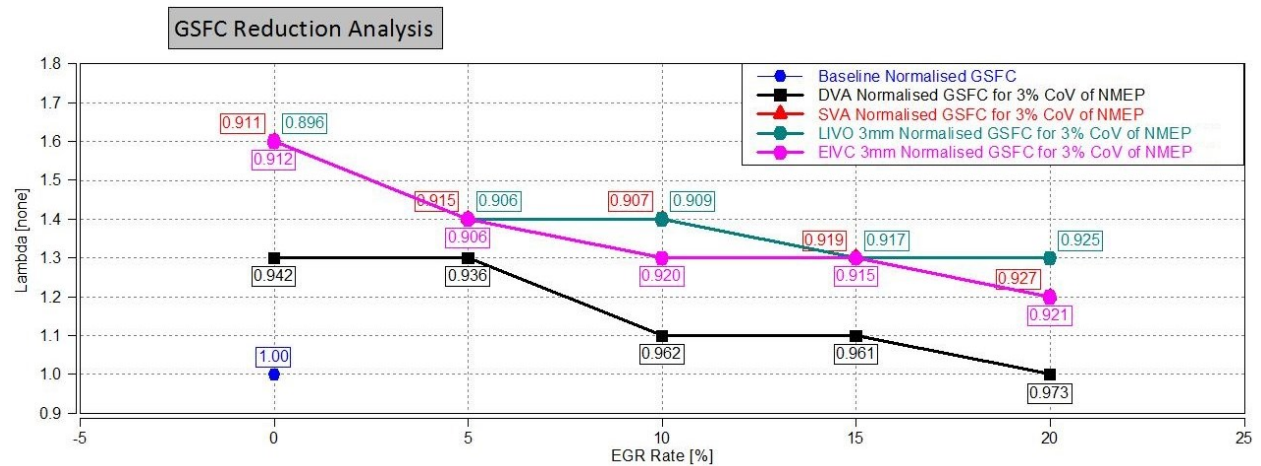


Figure 4.46 - Maximum GSFC reduction achieved for DVA, SVA, 3mm LIVO and 3mm EIVC normalised to baseline stoichiometric, 0% EGR DVA GSFC at 1500 RPM, 7.9 bar GMEP

Table 4.5 - Summary of the highest GSFC benefit achieved for all tested valve strategies at 1500 RPM, 7.9 bar GMEP

Valve Strategy	Lambda [-]	EGR Rate [%]	Benefit achieved over stoichiometric DVA [%]	Penalty over the best condition overall [%]
DVA	1.3	5	6.4	4.5
SVA	1.4	10	9.3	1.2
3mm LIVO	1.6	0	10.4	-
3mm EIVC	1.4	5	9.4	1.1

Similarly, to 1500 RPM, 3.6 bar GMEP case, overall indicated fuel conversion efficiency was calculated. It should be noted that once the pumping loop was included in the analysis, LIVO and EIVC switched places, however by a margin of less than 0.5%, and both of them broke the 40% efficiency barrier. All the optimised points were run at plenum pressures above the atmospheric. It is also worth stressing that compared to baseline, stoichiometric DVA operation

at 1500 RPM, 7.9 bar GMEP, an impressive 12.7% increase in thermal efficiency was achieved by the means of dilution and running advanced valvetrain strategy – 3 mm EIVC staggered lift.

Table 4.6 - Thermal efficiency results achieved for the GSFC optimised conditions at 1500 RPM, 7.9 bar GMEP

Valve Strategy	Lambda [-]	EGR Rate [%]	η_f [%]	$\Delta\eta_f$ vs the best η_f achieved[%]
DVA	1.3	5	39.20	3.0
SVA	1.4	10	39.76	1.6
3mm LIVO	1.6	0	40.31	0.2
3mm EIVC	1.4	5	40.40	-
DVA	1.0	0	35.85	11.3

4.2.3. 2000 RPM, 9.8 bar GMEP

4.2.3.1 LIVO Operation, Lambda Study

Experimental study was extended to higher speed and load condition, however it was not supplemented by numerical simulations. Discussion of experimental results is therefore based on the observations made earlier for two 1500 RPM studies of staggered lift and Chapter 3 results of valve deactivation at this speed and load condition.

Combustion stability was again significantly improved over the standard DVA operation up to a threshold value of $\lambda=1.6$ for all LIVO maximum lifts and they matched stability observed for SVA (Figure 4.47). Beyond this point a step decrease in combustion stability with no apparent trend took place. It can be said with high level of confidence that $\lambda=1.7$ condition did not meet stability criteria for any valve lifts. It is based on the readings of CoV of NMEP listed best to worst, being: 4 mm, 2 mm, 3 mm, 1 mm, 5 mm, SVA, hence the results prove that there was high chance of partial burns and the actual recorded values most likely differed only on the basis of how many of these randomly occurring cycles happened within the 300 cycles that were recorded during measurement.

Combustion durations under LIVO staggered operation at 2000 RPM followed the same qualitative trends described for both 1500 RPM cases.

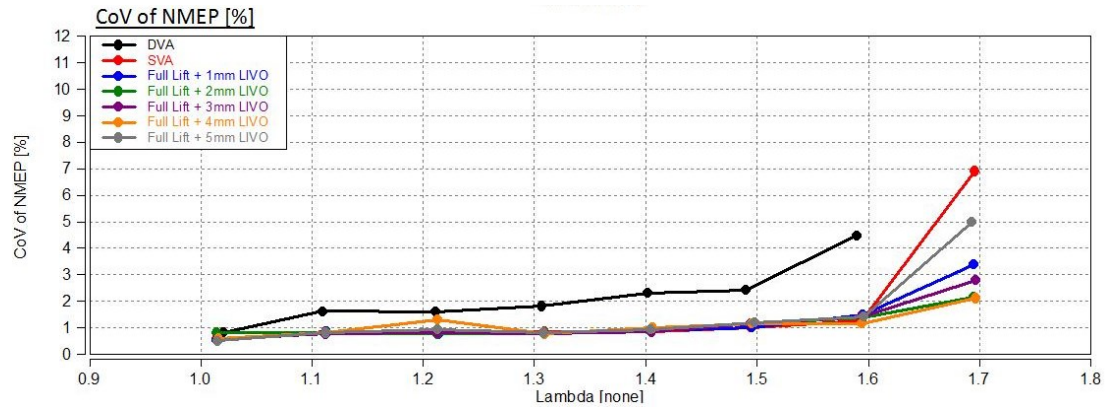


Figure 4.47 - Combustion stability of LIVO cases for lambda sweep at 2000 RPM, 9.8 bar GMEP

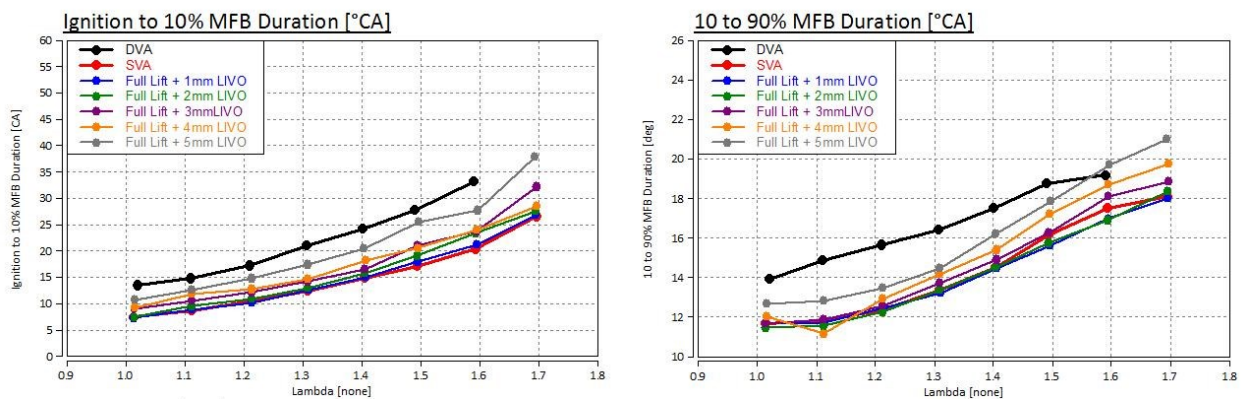


Figure 4.48 - Burn duration analysis of LIVO cases for lambda swings at 2000 RPM, 9.8 bar GMEP

Fuel consumption benefit delta between various lift conditions was much smaller compared to lower speed and load cases, however in-line with what has already been presented in Chapter 3. Adding to the extensively described small differences in fuel consumption between DVA, and SVA, it can be said that opening second valve in the LIVO mode did not bring any significant benefits, and even though 4 mm maximum lift LIVO case achieved the lowest GSFC value of 207.3 g/kWhr (at $\lambda=1.6$), it was within 0.5% of the value recorded for SVA, and only 2.2% better than DVA at $\lambda=1.5$ (for which it did not breach the 3% CoV of NMEP stability limit).

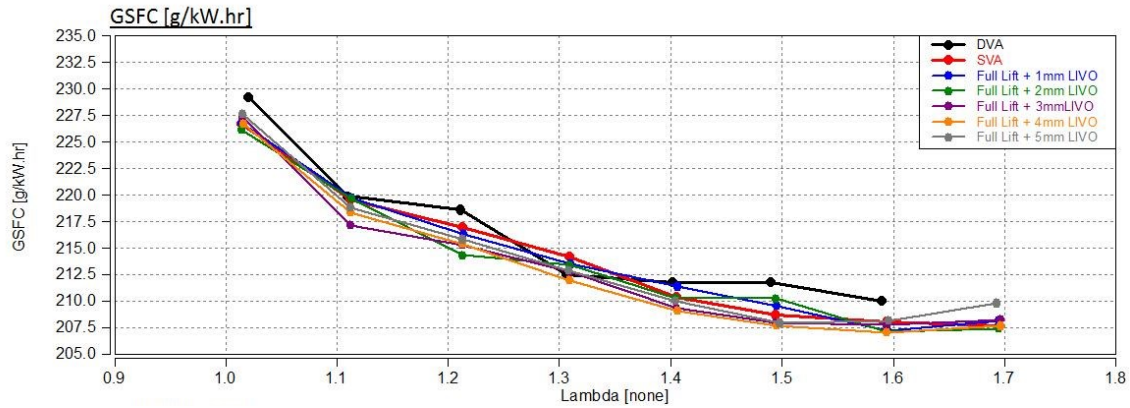


Figure 4.49 - GSFC of LIVO cases for lambda sweep at 2000 RPM, 9.8 bar GMEP

Emissions formation trends were also very similar to the ones already seen for the lower speed and load cases. The only exception observed for THC values, was that all LIVO cases were more closely matched to SVA, and values were kept below 1000 ppm up to the last point that could be potentially used for engine calibration ($\lambda=1.6$) for all strategies other than DVA.

NO_x trends presented in Figure 4.50 formed only 2 lines. Lower formation was represented by DVA strategy, whilst SVA and 1-5 mm LIVO were consistently higher and non-distinguishable from each other (values were almost identical).

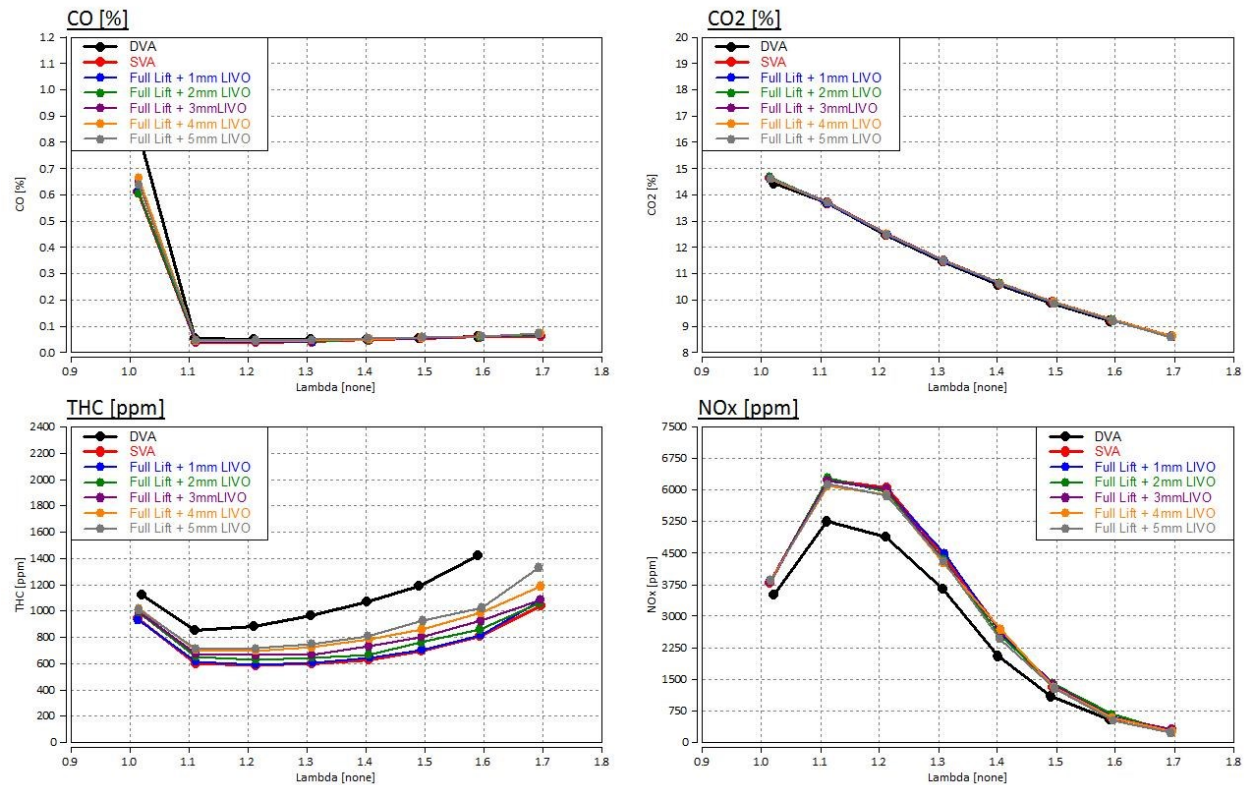


Figure 4.50 - Emissions formation comparison for lambda sweep between various max lift LIVO cases at 2000 RPM, 9.8 bar GMEP

4.2.3.2 EIVC Operation, Lambda Study

Subjecting SCRE to running with EIVC mode on the second lift at 2000 RPM, 9.8 bar GMEP presented itself with slightly different characteristics of its combustion stability (Figure 4.49), however it was well matched to trends seen earlier for 1500 RPM, 7.9 bar GMEP scenario (Figure 4.32) i.e. 1 – 4 mm EIVC lifts were well matched to SVA, whilst 5 mm EIVC lift started showing deteriorated stability much earlier. It was shown by trendline representing 5 mm EIVC lift separating towards DVA at $\lambda=1.3$ in Figure 4.51. Combustion durations were also noticeably extended with increasing EIVC lifts, to the point where 4 mm and 5 mm cases burned slower than DVA. It can be seen in Figure 4.52 that both initial (ignition to 10% MFB) and later (10% to 90% MFB) stages of combustion were affected. Such behaviour was most likely a result of significantly decreased TKE during ignition (proven for all other cases analysed earlier).

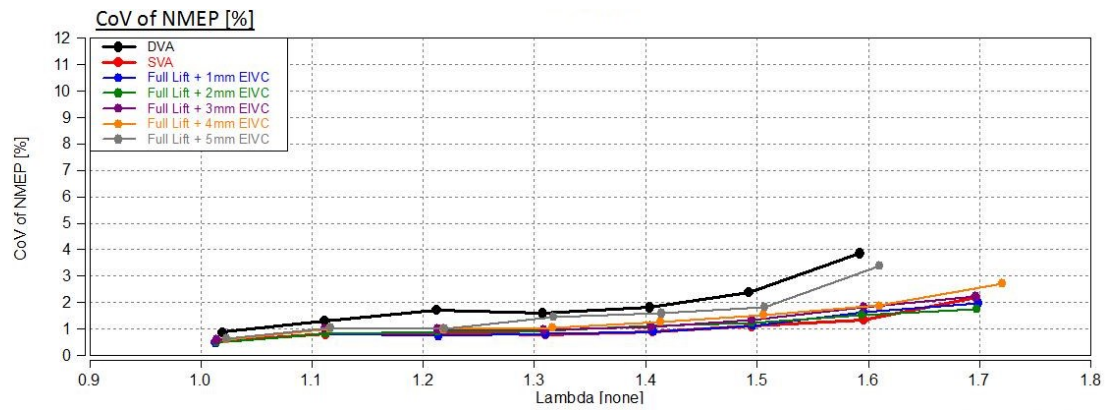


Figure 4.51 - Combustion stability of EIVC cases for lambda sweep at 2000 RPM, 9.8 bar GMEP

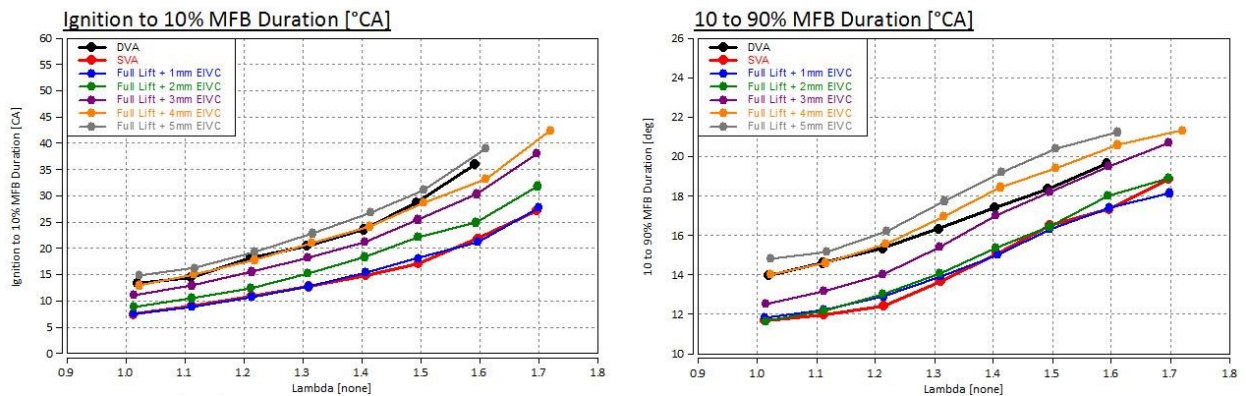


Figure 4.52 - Burn duration analysis of EIVC cases for lambda swings at 2000 RPM, 9.8 bar GMEP

As a result, fuel consumption metric of GSFC did not present any significant benefits of introducing staggered lift of either LIVO or EIVC. GSFC of 4 mm EIVC lift presented in Figure 4.53 achieved value lower than other lifts by approximately 1.2%, however its trendline appeared to have high and irregular differences point-to-point, hence it was deemed negligible and most likely affected by measurement error.

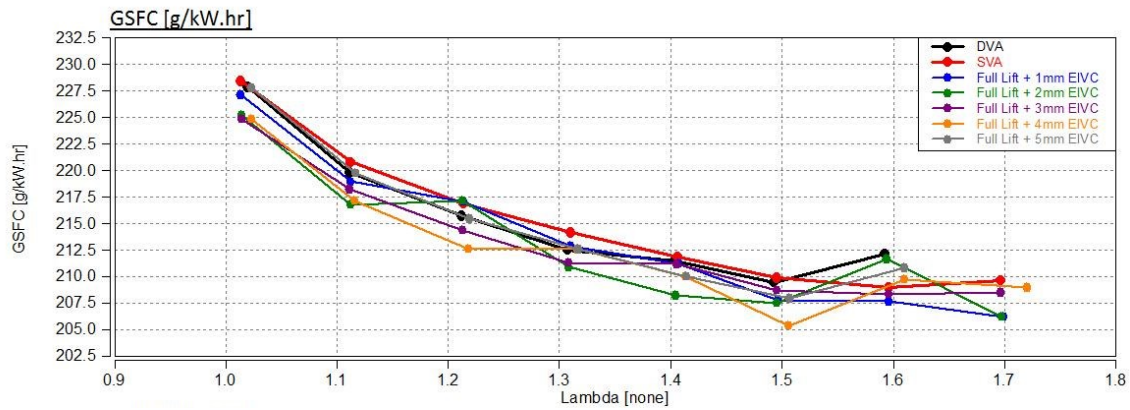


Figure 4.53 - GSFC of EIVC cases for lambda sweep at 2000 RPM, 9.8 bar GMEP

Pollutants formation was also not significantly altered by introducing EIVC staggered lift mode at 2000 RPM, 9.8 bar GMEP. There was only slightly higher spread in NO_x emissions (Figure 4.54), showing that introducing EIVC had more effect than LIVO in shifting general trend lines away from the line representing SVA and towards the DVA. Similar increase in data spread when applying EIVC was observed earlier for 1500 RPM cases.

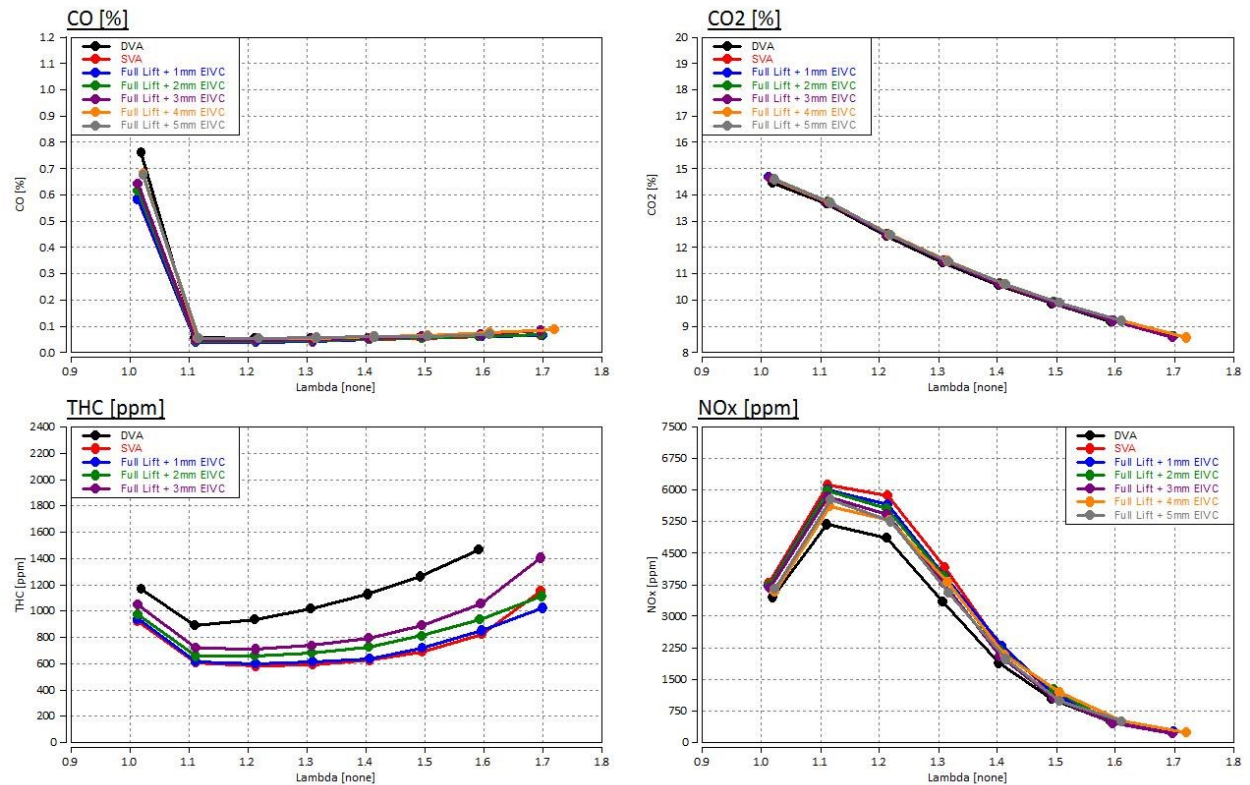


Figure 4.54 - Emissions formation comparison for lambda sweep between various max lift EIVC cases at 2000 RPM, 9.8 bar GMEP

Due to mechanical failure of the engine, it was not possible to run supplementing study of EGR and excess air dilution (as presented for 1500 RPM cases), however 4 mm LIVO and 2 mm EIVC conditions would have been analysed further. However, based on the findings already presented in Chapter 3 and lambda sweeps performed as part of staggered lift study, very moderate improvements in fuel consumption were expected.

4.3. Concluding Remarks

Use of a CVVL system installed on the intake of SCRE enabled study of two differential intake valve lift strategies – Late Intake Valve Opening (LIVO) and Early Intake Valve Closing (EIVC). Both strategies were examined with maximum valve lifts on the second valve ranging from 1 mm to 5 mm (in 1 mm increments). The study was initiated by the conclusions reached upon examining Single Valve Actuation vs Dual Valve Actuation, stating that there must be an optimum strategy where the second valve is opened only slightly to increase in-cylinder

flow (compared to DVA), but at the same it would not suffer from significant heat transfer losses attributed to running in SVA mode. Experimental results confirmed the theory and the following conclusions were reached:

- Swirl motion created inside the cylinder was governing the TKE levels reached for each condition. Introducing swirl which would match or exceed tumble motion (swirl/tumble ratio ≥ 1 achieved for all LIVO cases and EIVC with second lifts < 3 mm) ensured significant increase in TKE at ignition.
- LIVO (regardless of the maximum lift) resembled SVA running condition studied in Chapter 3. It was caused by the similarities in flow forming during the induction stroke. Second valve always remained shut in the beginning, and was only opened halfway through induction stroke at the earliest for 5 mm scenario.
- All LIVO lifts on the second valve were characterised by significantly increased TKE during ignition, leading to faster burn and improved combustion stability for diluted mixtures when compared against DVA.
- EIVC strategy was more closely matched to DVA for maximum second valve lifts of 4 mm and 5 mm. It was caused by the initial part of the induction stroke being subjected to both valves opening, hence creating in-cylinder flow field matched to that of DVA. TKE at ignition was actually lower than that of DVA which resulted in slower combustion.
- Lower TKE at ignition for higher lifts was attributed to reduced swirl motion. Second valve opened late during induction stroke when jet velocities already decreased and strong swirl motion was unable to be generated.
- When EIVC was subjected to lifts of 3 mm or lower, second valve closed early enough to generate swirl motion that increased TKE levels above that of DVA at ignition and improved burn durations.
- Heat transfer penalties (compared to DVA) were significantly different when employing LIVO or EIVC. For lifts of 3 mm and higher relative heat transfer penalty was more than twice as high for LIVO compared to EIVC. It was quantified at 1500 RPM to be 22% vs 52%, 31% vs 73% and 43% vs 90% for 7.9 bar GMEP and 17% vs 56%, 34% vs 74% and 43% vs 88% for 3.6 bar GMEP at valve lifts of 3 mm, 4 mm and 5 mm respectively.

- Maximum heat transfer reduction of approximately 55% (compared to SVA) could be achieved without negatively affecting combustion metrics and making them inferior to standard DVA condition.
- Overall indicated engine efficiency was significantly increased compared to baseline, stoichiometric DVA operation as well as optimised DVA and SVA diluted conditions. It was found that 3 mm maximum lift EIVC strategy was optimum for both load cases (3.6 and 7.9 bar GMEP) at 1500 RPM. Values reached:
 - 32.56% vs 30.41% at 3.6 bar GMEP. An improvement of 7.1% over stoichiometric DVA.
 - 40.40% vs 35.85% at 7.9 bar GMEP. An improvement of 12.7% over stoichiometric DVA.
- Based on the GSFC metric alone, 3 mm EIVC scenario was found to be optimum at 1500 RPM, 3.6 bar GMEP. It achieved improvement over stoichiometric DVA condition of 7.2%, compared to 5.4% reached for both 3 mm LIVO and SVA.
- At 1500 RPM, 7.9 bar GMEP optimum GSFC values were reached for 3 mm LIVO strategy, and achieved reduction of 10.4% over stoichiometric DVA condition. It was followed by 3 mm EIVC at 9.4% and SVA at 9.3%.
- Best overall efficiency reached for EIVC vs LIVO (in contrary to GSFC benefit) indicated that EIVC strategy had less pumping losses, however the difference was marginal (>0.5%).
- Staggered lift cases optimised for fuel consumption were located at much higher lambda values for a given EGR rate (vs DVA), and exhibited overall dilution levels similar to that of SVA.
- Emissions formation was found to be more sensitive to EIVC strategy rather than LIVO, especially at increased loads. Significantly altered charge motion within the cylinder resulted in higher piston wetting and THC formation compared to LIVO. Fuel impingement on the piston was also found to be increasing as second valve lift increased for EIVC with lifts of 2 mm and higher.
- NO_x production was found to be maximum for SVA and minimum for DVA regardless of condition tested, especially for lower lambda values (sensitive to load). Introducing staggered lift resulted in production rates lower than SVA, but higher than DVA. Generally, as the maximum lifts were increased, EIVC started approaching lower limit set by than DVA faster than LIVO of equivalent lift.

Chapter 5

High Energy Ignition

Systems Evaluation

5.1. Preliminary Remarks

An investigation was launched to address the needs of AJ-20 combustion system requirements for an ignition system that would be most effective in increasing its tolerance to charge dilution. It was also motivated by potential synergies that could be achieved by combining a higher energy ignition system with increased in-cylinder velocities identified by Bosch in their assessment of requirements for modern ignition systems (Brandt *et al.*, 2016).

Based on the data presented in Chapter 3, 1500 RPM, 3.6 bar GMEP condition was selected as the most appropriate for this study based on its combustion stability behaviour. It has shown the most gradual deterioration of combustion stability and highest potential of introducing SVA operation. Other points (1500 RPM, 7.9 bar GMEP and 2000 RPM, 9.8 bar GMEP) were characterised by more sudden drops in combustion stability and relatively lower benefits of valve deactivation strategy. Limiting the study to a single point of interest was dictated by time constraints and significant time expense associated with both running the tests and performing

data post-processing. The new system was designed to measure the secondary voltage and current delivered to the spark plug during ignition (described in Chapter 2).

Measurements were completed under both motored and fired conditions for DVA and SVA. Introducing motored points analysis was dictated by the desire to fully understand effects of in-cylinder flow field on the ignition and decouple them from combustion related variables. Need for comprehensive assessment of three ignitions systems was identified based on limited publications which attempted to identify underlying causes of extended engine operating envelope for modern DISI engines.

The test matrix used for this study was reduced (compared to the contour plots presented in Chapter 3 and 4) due to significantly extended measurement time for each point and need for gathering both firing and motoring data, effectively doubling the number of measurements. Lambda was incremented by 0.2 (instead of 0.1 used previously) up to $\lambda=1.4$, and then by 0.1 to gain extra resolution in the most unstable regions. Similarly, EGR rates were only increased in 10% steps.

It should also be noted that due to mechanical failure engine was rebuild prior to taking spark measurements and this study was taken in Build 4 phase (as outlined in Chapter 2).

The amount of data acquired, considering the very high sampling rate needed for current and voltage, the numerous ignition systems and two valve operating strategies, was very large. Therefore, measurements presented in this chapter serve mainly as a database of spark energy and duration behaviour that warrants analysis beyond what was possible for fundamental understanding without parallel visual observations in an optical engine. Nevertheless an effort was made to link the spark data at all conditions via the rate of heat release both in the early stages of combustion identified by 0-5% MFB and the later stages identified by 50% MFB timing. Clearly there is room for further work in providing holistic understanding of these systems that would require input from suppliers of the systems that was not made possible within the time constraints of this thesis and the fact that the AJ-20 optical engine in the lab was not available for this type of work due to other research engagements in projects that were running in parallel.

5.2. Test Procedure

Having established, that 1500 RPM, 3.6 bar GMEP condition was the most promising from the dilution extension via use of high energy ignition systems point of view, a new test procedure was designed. At first, engine was run with stock CoP system, targeting the nominal 3.6 bar GMEP load condition and 50% MFB at 8°C_A aTDC_f. Ignition angle and corresponding plenum pressure was recorded for each condition. Other ignition systems were then run targeting the same intake plenum pressure and ignition advance. It was done to ensure that all systems were compared with the same in-cylinder flow field (being a function of plenum pressure) and the same in-cylinder pressure at ignition (realised by fixing the ignition angle).

Detailed investigation included taking measurements for both DVA and SVA, as well as fired and motored point of every single test point. It was done to gain extra insight into the secondary voltage and current traces, and how they behave without introducing an extra level of complexity associated with combustion. After completing two separate datasets for DVA and SVA, limited number of points run using the DVA strategy were matched using SVA. Matching was done by using two new methods. First methods targeted the same in-cylinder pressure at ignition and the same ignition angle. To achieve that SVA plenum pressure had to be lowered. Second method also targeted the same in-cylinder pressure at ignition, but instead of matching the ignition angle, intake plenum pressure was matched and ignition angle was advanced.

5.3. Results

5.3.1. System Characteristics, Typical Traces

Firstly, typical secondary voltage and current traces (extracted for $\lambda=1.0$, 0% EGR cases) were presented for three different ignition systems. They were run in total of five different configurations:

- Stock Coil
- Bosch CEI – 5 ms duration, maximum power
- Bosch CEI – 3 ms duration, maximum power
- BW DCI – Dual Coil Continuous (DCC) mode, 12 A primary current, 5 ms duration

- BW DCI – Multistrike mode, 12 A primary current, 5 ms duration

The above settings were applied to capture maximum theoretical benefit available when running any of the systems. 5 ms duration Bosch CEI system was set to 100% power level, 0.0 crossover, dwell time of 1.2 ms (advised by system supplier) and maximum 5 ms boost time. Second configuration of Bosch system only changed boost time setting to 3 ms. It was done to investigate ignition duration sensitivity.

BW DCI system was also set to run with the maximum attainable primary current of 12 A. Similarly to Bosch, BW system was also run in two modes, however it was done to test sensitivity to mode of spark delivery, rather than its duration. Both continuous discharge (DCI/DDC) mode and multistrike/ multispark modes were set to 5 ms duration.

It was duly noted that other researches presented diminishing returns of running overly long ignition timing (Brandt *et al.*, 2016), however systems were powered by external power source and could be used with maximum setting without any expense from the engine output point of view. It was acknowledged that such operation would not be possible in real applications where increased power drain would affect alternator work, and should be optimised.

Stock Coil

Figure 5.1 presented typical current and voltage traces of stock, single CoP system. Maximum current recorded after the breakdown was measured at approximately 100 mA. It should be noted at this point that the actual breakdown has not been captured due to its extremely short timescales expressed in nanoseconds (as per details presented in Chapter 1). Immediately after reaching its peak, current started dropping at a fairly constant rate. It resulted in triangle shape of the current trace which is characteristic for conventional Transistor Coil Ignition (TCI) systems. Typical voltage traces were characterised by the peak voltage reached immediately after spark breakdown in the range of 1.0-1.5 kV, whilst the voltage did not drop below 500 V throughout the duration of the spark event. When current dropped to 0 and no further power was transferred to the spark plug (represented by bottom green line in Figure 5.1), potential difference was no longer sustained and voltage trace decayed towards 0.

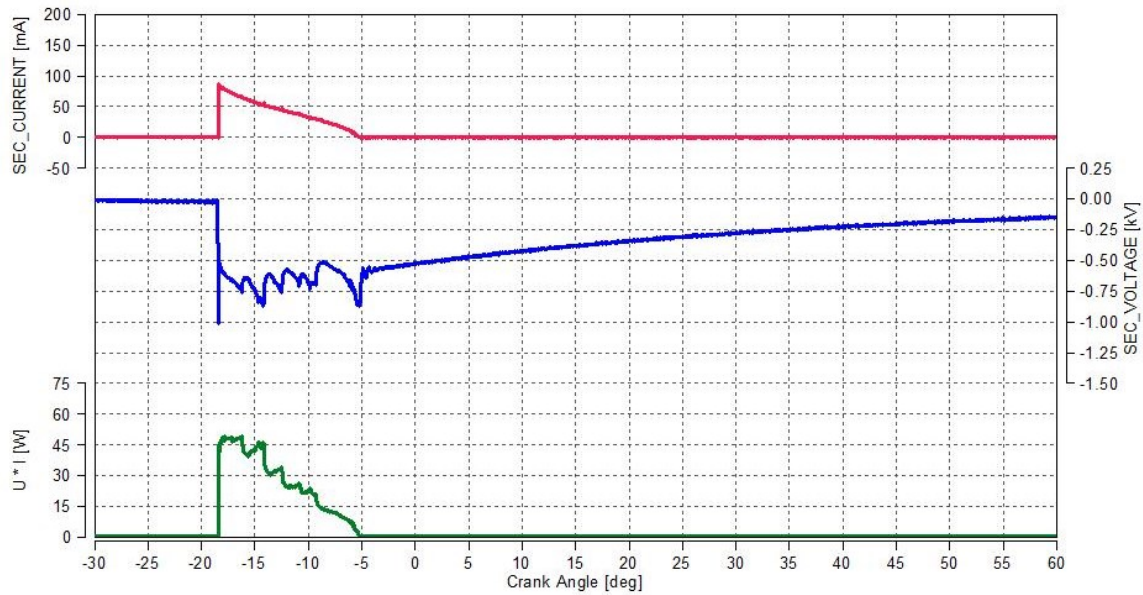


Figure 5.1 – Representation of secondary current and voltage for Stock Coil ignition system at 1500 RPM, 3.6 bar GMEP, stoichiometric condition

A series of plots representing spark duration [ms] vs energy [mJ] were produced. Values were computed by the MATLAB script described in Chapter 2 using the 10 MHz data acquired by National Instruments PXI-6115 ADC device.

Data was presented in clusters of 8 plots for ease of comparison. Firstly, DVA motored conditions were plotted for $\lambda=1.0$ (left hand side) and $\lambda=1.6$ (right hand side). These two values were chosen to make comparison between standard, stoichiometric condition and most diluted (lean) condition run on every system for both DVA and SVA. Motoring traces were acquired after firing condition by disabled fuelling. Throttle position and plenum pressure were not adjusted, hence plenum pressure was few kPa higher due to changes in temperature and density.

Then, the same two points were presented under standard firing condition. It was done to present how introducing combustion might have changed the results.

Third set of data was presented under motoring conditions, however one of the intake valves was deactivated, running in SVA mode.

Fourth, last set was comparing the same lambda cases for SVA with normal combustion taking place inside the cylinder.

All plots presented negative correlation between duration and energy, i.e shorter cycles were also characterised by higher energy delivered to spark plug (illustrated by best fit lines in Figures 5.2 - 5.5). Pearson correlation coefficients were calculated to illustrate relationship between duration and energy. Whilst there is some ambiguity as to when it can be said that datasets are correlated, for the purpose of this study it was assumed that if $|r_{xy}| \geq 0.3$, then datasets were correlated. This criteria was met for all tested conditions with Stock Coil.

It was found that stock coil was significantly more prone to spread of ignition energy and its duration under motored conditions for both DVA and SVA. SVA also appeared to have generally higher spread in data compared to DVA across both motored and fired conditions.

Interestingly, there was much less variation (represented by tight clusters) at $\lambda=1.0$ vs $\lambda=1.6$ for fired conditions, whilst the trend was opposite under motoring. Such behaviour was observed for both DVA and SVA.

To get additional insights into the possible reasons behind the aforementioned trends, averaged in-cylinder velocities were investigated.

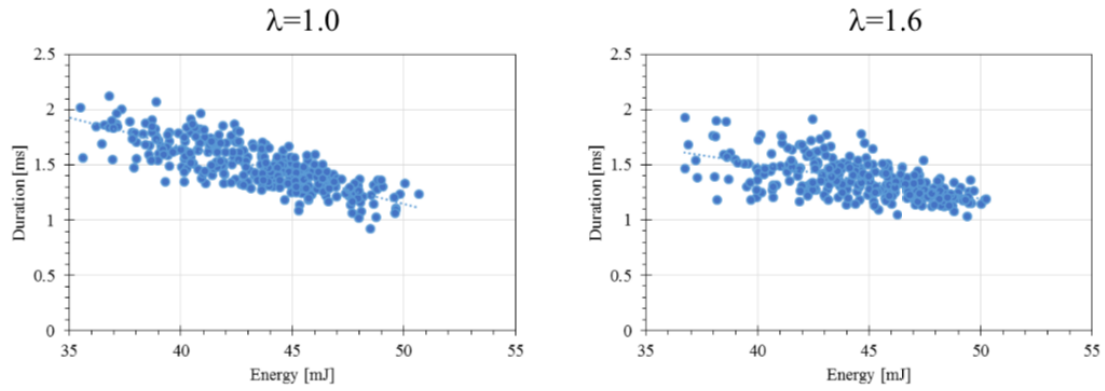


Figure 5.2 – Comparison of DVA Stock Coil Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP equivalent motoring condition

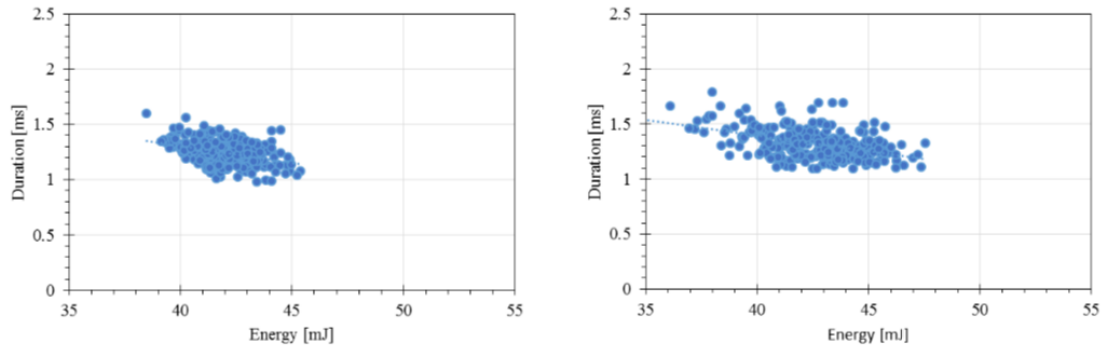


Figure 5.3 - Comparison of DVA Stock Coil Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP

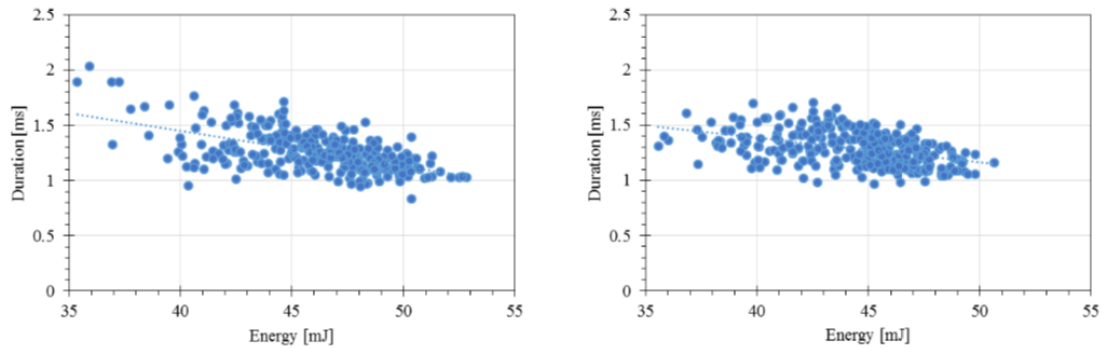


Figure 5.4 - Comparison of SVA Stock Coil Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP equivalent motoring condition

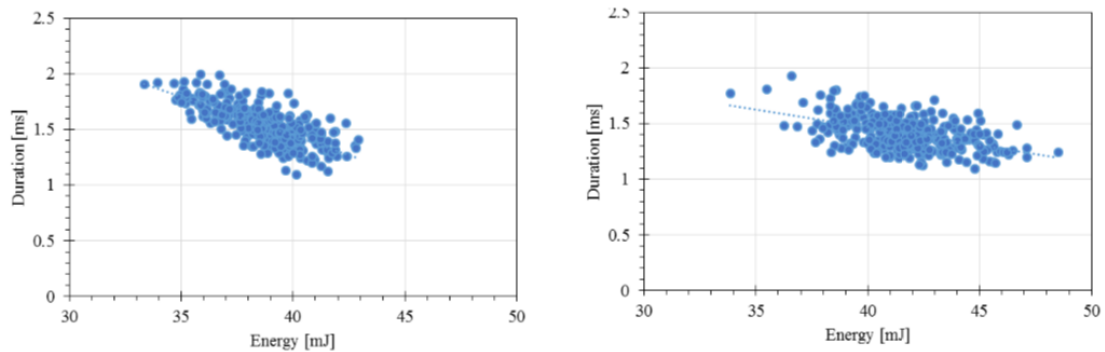


Figure 5.5 - Comparison of SVA Stock Coil Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP

Velocities around the spark plug were identified as key contributors towards long plasma arcs by Schneider *et al.* (2016) and Brandt *et al.* (2016). Method introduced for analysis of their magnitude was very similar to the one introduced in Chapter 3 for NO_x formation analysis, i.e. artificial spheres centred around the spark plug. Spheres of 10 mm, 20 mm, and 30 mm were used in the subsequent analysis. Average velocities were calculated within these spheres in the 40°C**A** bTDC_f to 10°C**A** aTDC_f range and spark timings were marked for any plotted condition.

Firstly, contribution of fuel injection was assessed. Numerical simulations proved that it was negligible for SVA and somewhat more pronounced for DVA, however it did not affect general trends. Comparison of Figure 5.6 showing air only simulation and Figure 5.7 representing same condition including fuel spray led to conclusion that simulating air only was sufficient.

Comparison of DVA vs SVA at $\lambda=1.0$ was then performed for each sphere. Most pronounced was the difference in velocity values between two valve operating strategies within the smallest, 10 mm spheres (arguably the most important for spark behaviour). Even though SVA started at higher value (~15.5 m/s) at 40°C**A** bTDC, it dropped sharply to approximately 3 m/s at 20°C**A** bTDC. DVA started at 12 m/s and was gradually decreasing to ~4 m/s at TDC at fairly constant rate. DVA's average velocity values within the 10 mm sphere were higher for DVA than SVA from 35°C**A** bTDC to TDC, where both strategies plateaued at the same value. Comparing values at ignition, DVA strategy experienced velocities within 6.5 to 7.0 m/s within all three spheres, whilst SVA had much bigger spread and values were noted at 3 m/s, 5.5 m/s and 7.5 m/s for spheres of 10 mm, 20 mm and 30 mm respectively. Therefore, SVA was subjected to higher velocities further away from the spark plug, given that when averaged over entire sphere it surpassed DVA only for the sphere size of 30 mm in diameter.

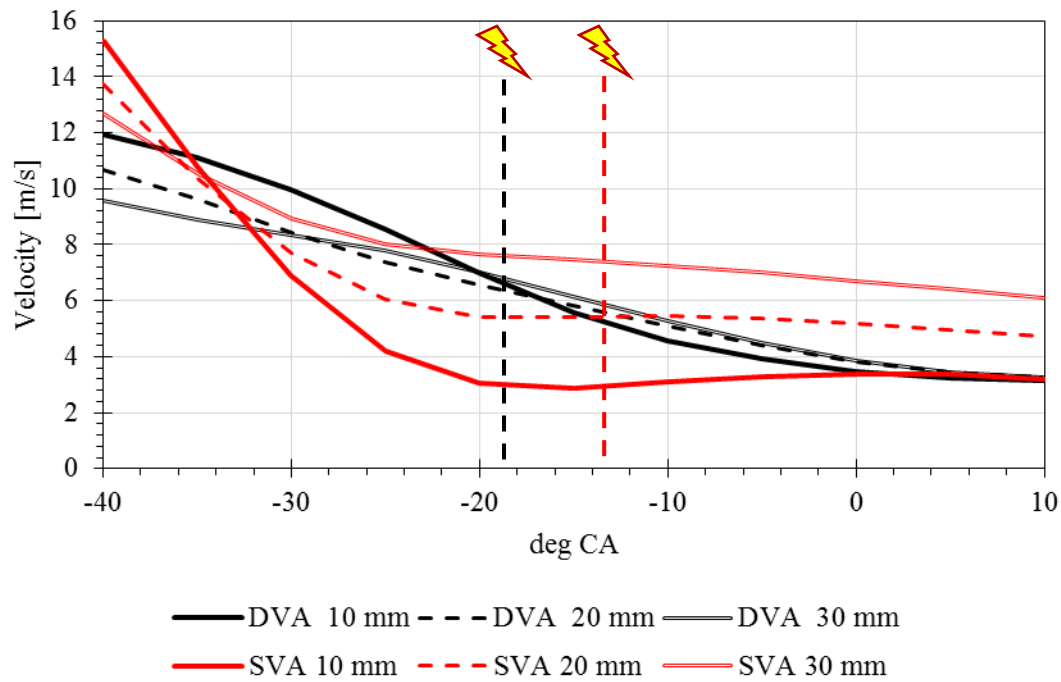


Figure 5.6 – Averaged in-cylinder velocities for 10-30 mm spheres centred around the spark plug for DVA and SVA at 1500 RPM, 3.6 bar GMEP, $\lambda=1.0$; air only

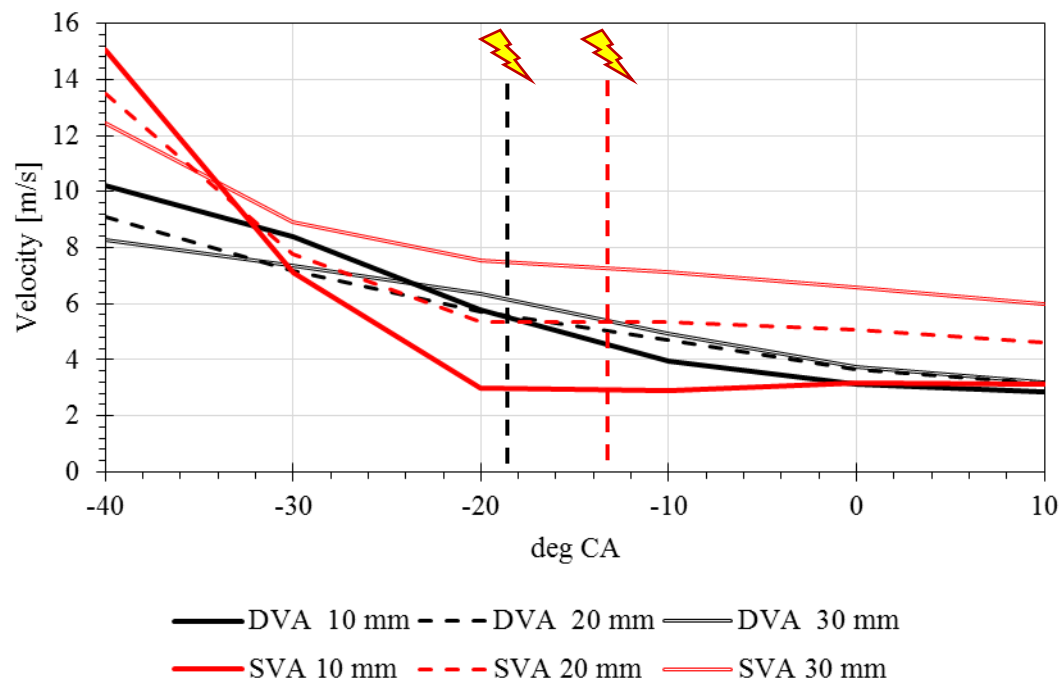


Figure 5.7 - Averaged in-cylinder velocities for 10-30 mm spheres centred around the spark plug for DVA and SVA at 1500 RPM, 3.6 bar GMEP, $\lambda=1.0$; modelled with fuel injection

Figures 5.8 and 5.9 showed how velocities differed between $\lambda=1.0$ and $\lambda=1.5$ for DVA and SVA respectively. It was found that there were no qualitative differences, and quantitative differences were negligible. Velocity values were fractionally higher for $\lambda=1.5$ due to increased plenum pressure and higher jet velocities during air admission process to the cylinder, however in the 50°CA region of interest they were less than 5% higher for DVA and less than 10% higher for SVA (compared to baseline $\lambda=1.0$ condition).

Notable differences were observed for velocity values during spark timing when comparing $\lambda=1.0$ and $\lambda=1.5$ due to significant advance required for more diluted condition (compensating for decreased burn duration). As the advance was increased, velocities were also increased. That was true irrespective of the valve strategy used, as all the velocity profiles presented in Figures 5.8 and 5.9 had negative slopes.

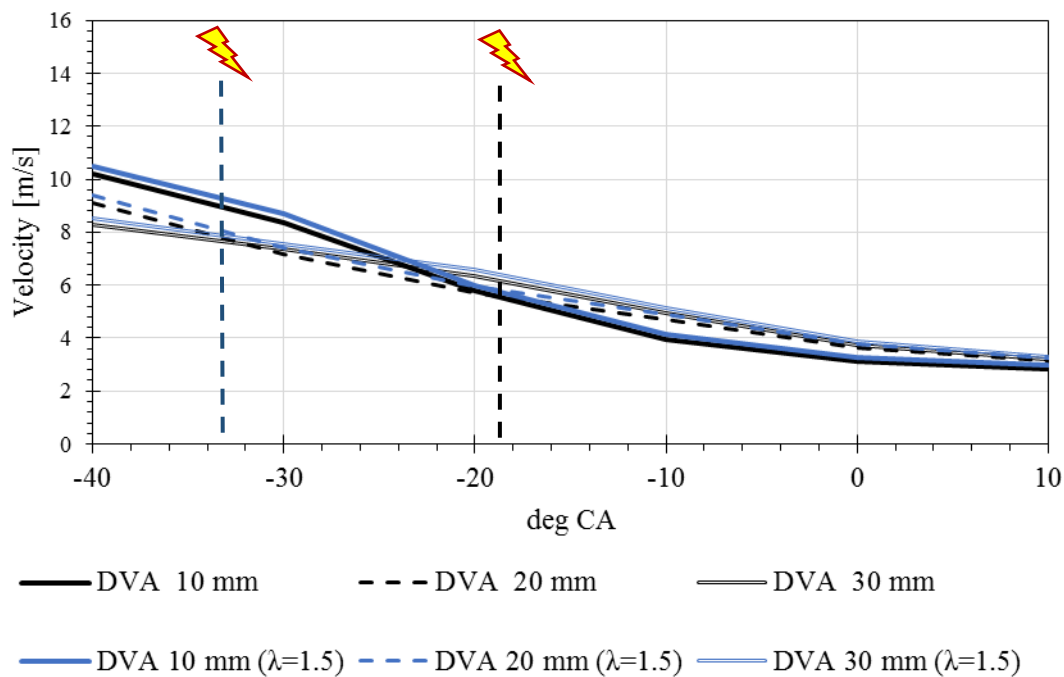


Figure 5.8 - Averaged in-cylinder velocities for 10-30 mm spheres centred around the spark plug for DVA at 1500 RPM, 3.6 bar GMEP; $\lambda=1.0$ and $\lambda=1.5$

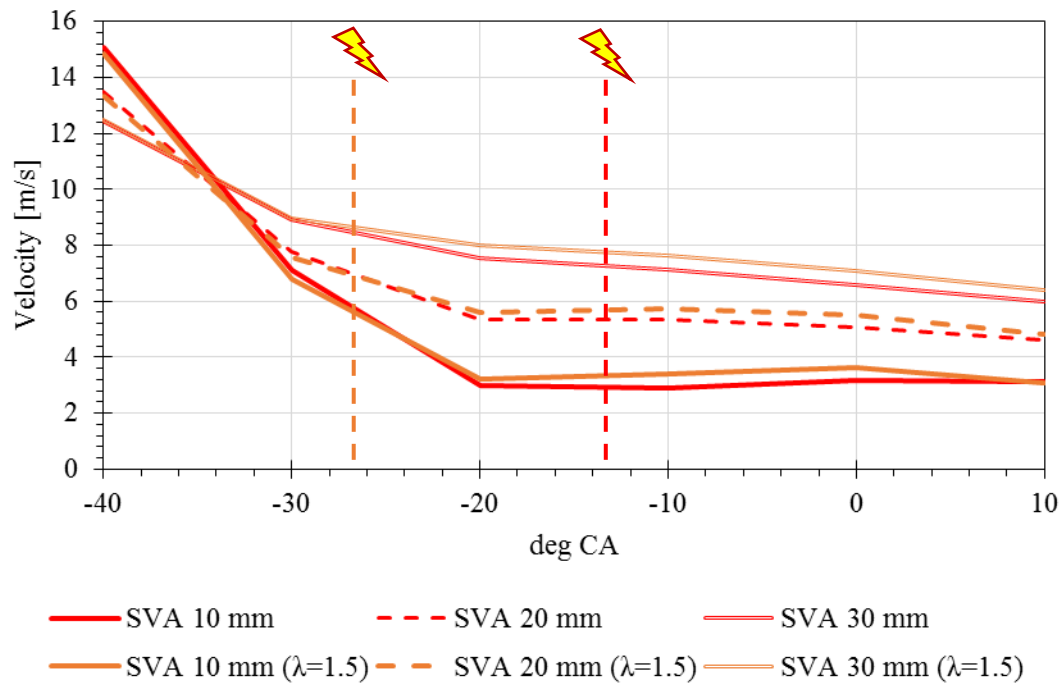


Figure 5.9 - Averaged in-cylinder velocities for 10-30 mm spheres centred around the spark plug for SVA at 1500 RPM, 3.6 bar GMEP; $\lambda=1.0$ and $\lambda=1.5$

Bosch CEI

5 ms Boost Time

Whilst 99% of the ignition events presented earlier for stock coil had duration within the 1-2 ms range characterised by the constant drop in current, prototype CEI system provided by Bosch was able to extend spark duration to more than 5 ms using the step-up converter integrated into the coil design. Typical current trace (presented in Figure 5.10) had a peak between 100 and 150 mA, followed by flat current trace maintained for approximately 5 ms at over 50 mA. Similarly to the current trace, the power delivered to the spark plug over entire duration of ignition event was fairly constant.

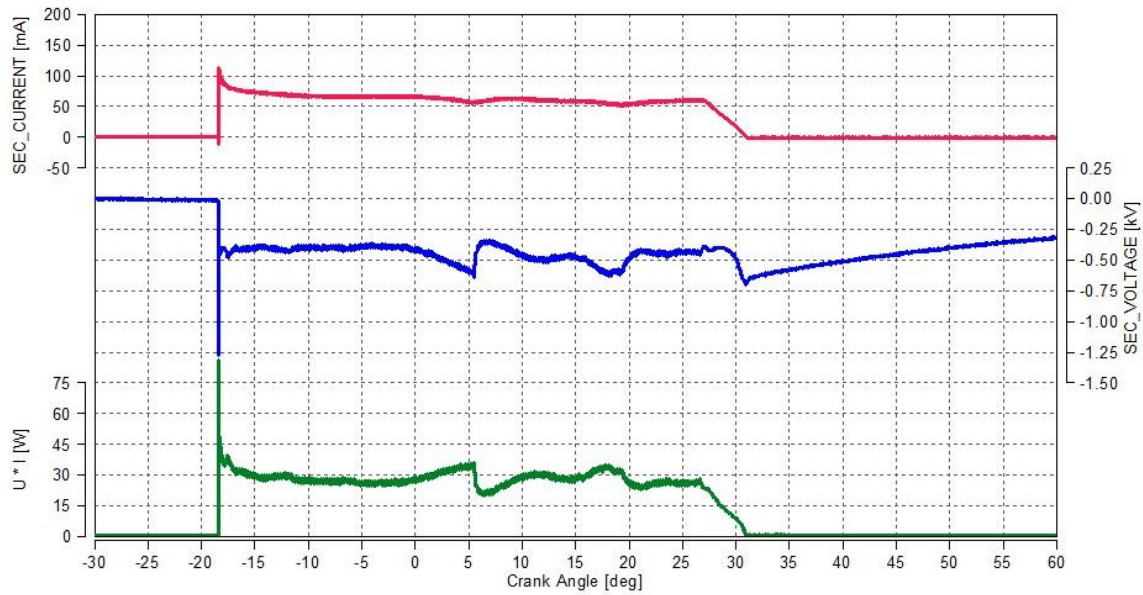


Figure 5.10 - Representation of secondary current and voltage for Bosch CEI ignition system (5ms ignition duration) at 1500 RPM, 3.6 bar GMEP, stoichiometric condition

Analysis of energy vs duration plots for 300 cycles measured by National Instruments based high frequency system showed that $\lambda=1.6$ condition had bigger spread of data compared to $\lambda=1.0$ which was especially well manifested by SVA motored condition (Figure 5.13). Frequent cycles of both lower than average energy and significantly shorter duration were observed. For most cases, however the same correlation was found, as earlier for stock coil, showing that shorter cycles attained higher energy. Also, $\lambda=1.6$ cases (both motored and fired, DVA and SVA) had higher average energy than $\lambda=1.0$. With exception of the motored SVA, $\lambda=1.6$ condition, SVA operated points also reached higher average energy than their counterpart DVA conditions.

Average energy per cycle was about 3-4 times higher than that seen for Stock Coil and it reached 140-170 mJ (depending on condition).

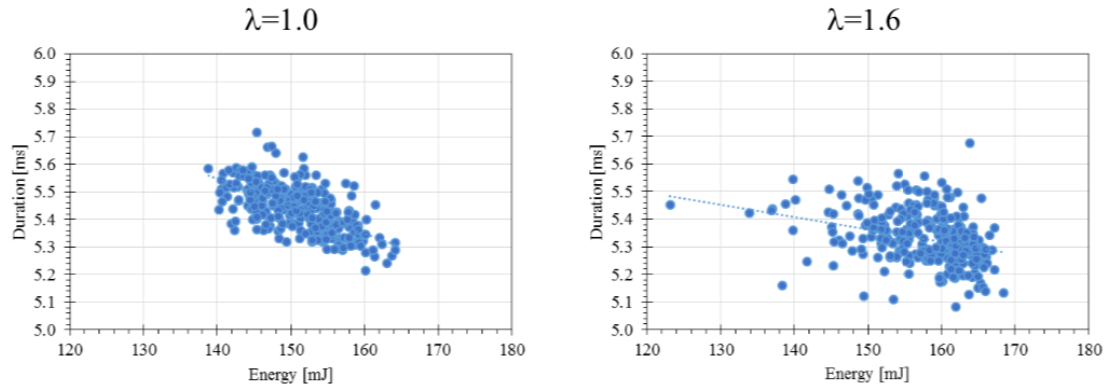


Figure 5.11 - Comparison of DVA Bosch CEI 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP equivalent motoring condition

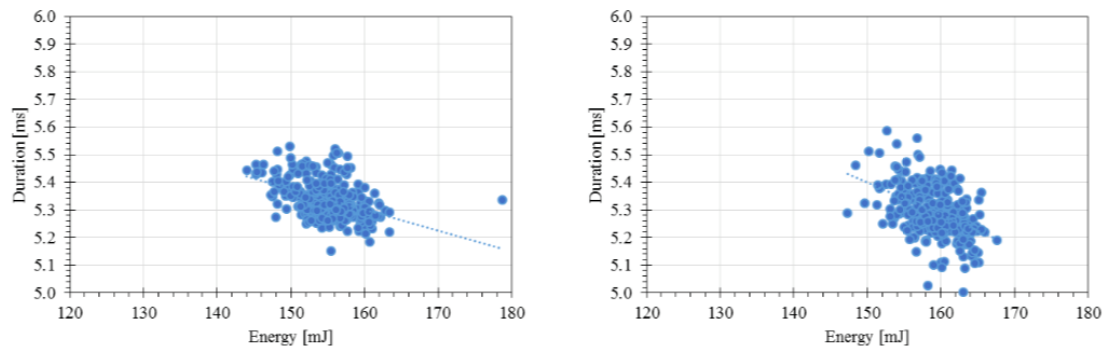


Figure 5.12 - Comparison of DVA Bosch CEI 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP

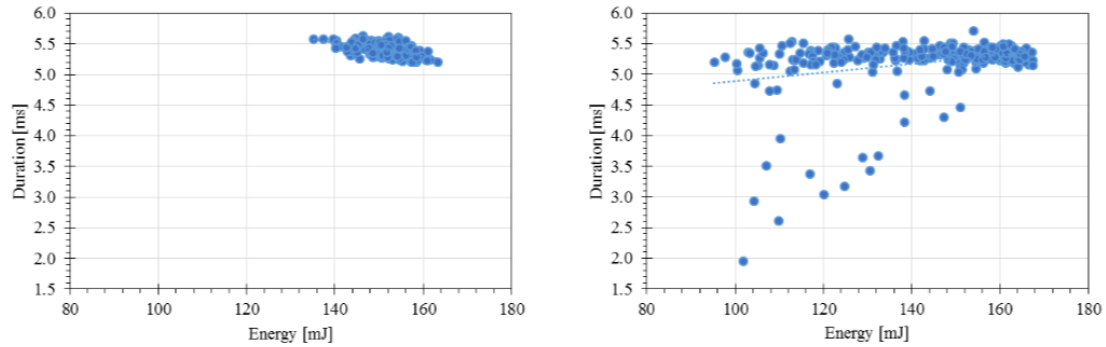


Figure 5.13 - Comparison of SVA Bosch CEI 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP equivalent motoring condition

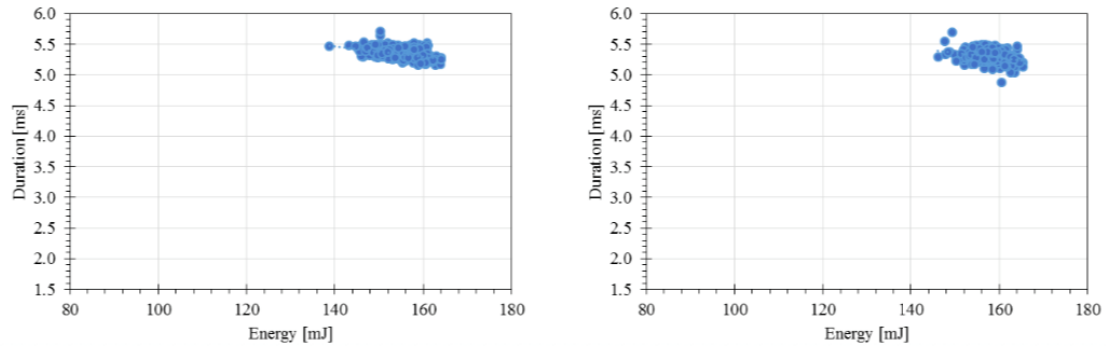


Figure 5.14 - Comparison of SVA Bosch CEI 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP

3 ms Boost Time

When Bosch CEI settings were changed and its Boost Time was lowered to 3 ms, the only difference in the performance was lowered ignition time (as expected). Current and power traces remained qualitatively the same and it can be seen when comparing Figure 5.15 representing 3 ms Boost Time with Figure 5.10 presented earlier for 5 ms.

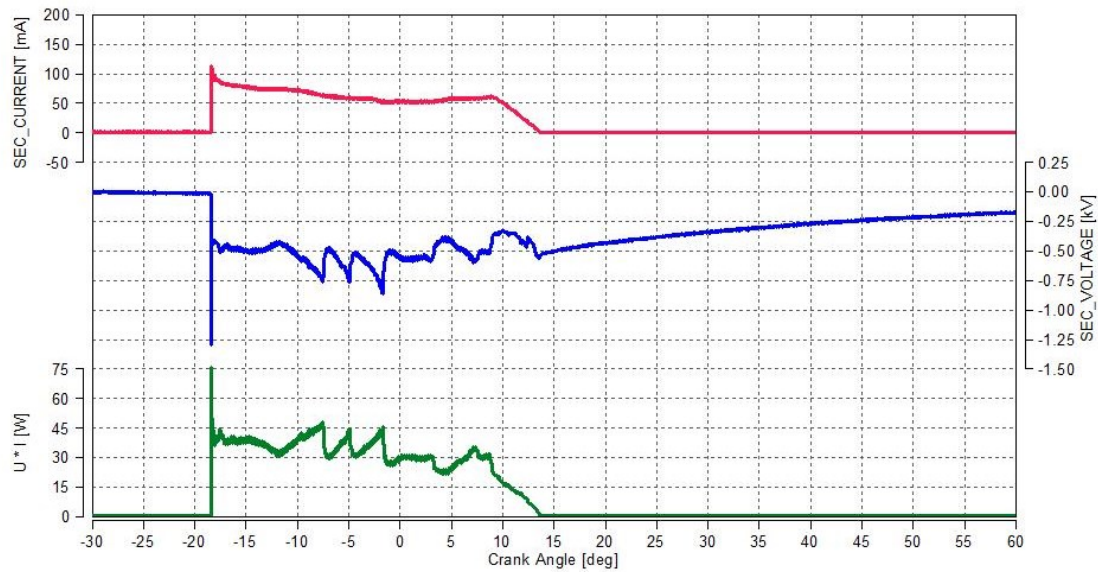


Figure 5.15 - Representation of secondary current and voltage for Bosch CEI ignition system (3ms ignition duration) at 1500 RPM, 3.6 bar GMEP, stoichiometric condition

Again, trends for duration vs energy did change significantly. Interestingly, motored SVA condition at $\lambda=1.6$ presented similar trend to that discussed earlier. It serves as a clear indicator that spread in data and occurrence of significantly shorter ignition events discussed for 5 ms settings earlier was not an outlier. Similarly to 5 ms settings, it only occurred under motoring conditions.

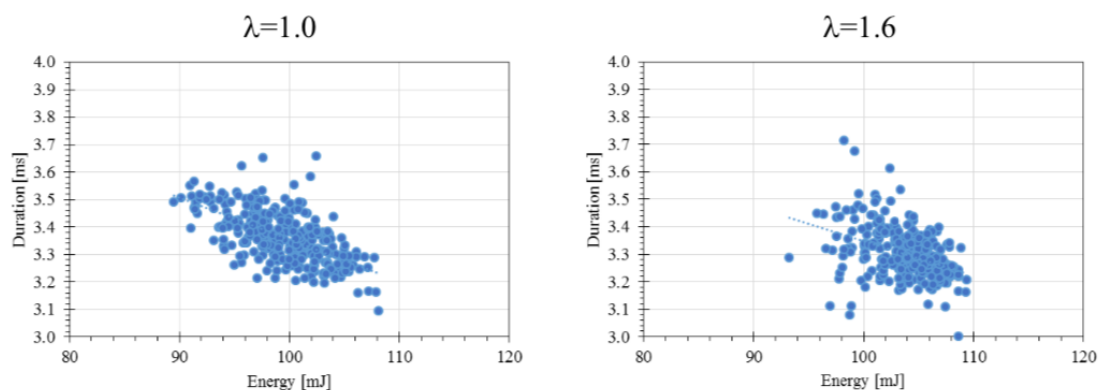


Figure 5.16 - Comparison of DVA Bosch CEI 3ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP equivalent motoring condition

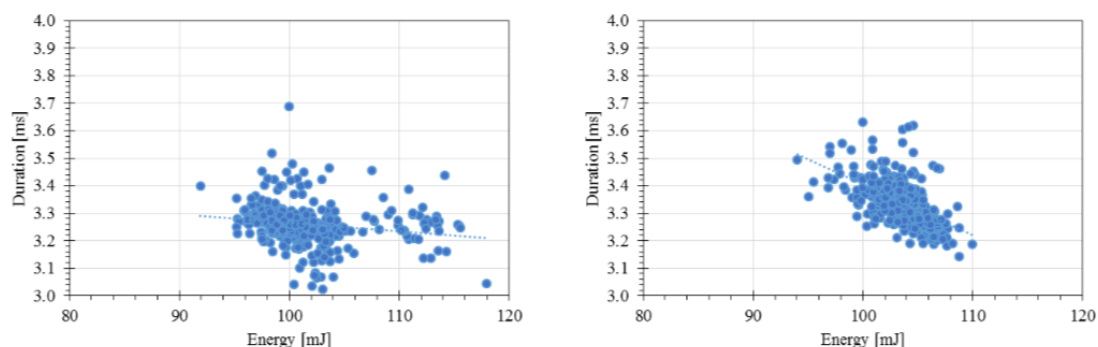


Figure 5.17 - Comparison of DVA Bosch CEI 3ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP

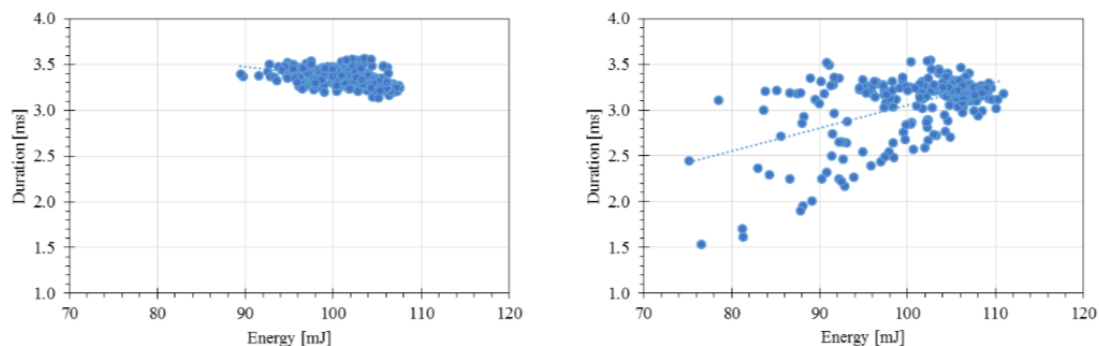


Figure 5.18 - Comparison of SVA Bosch CEI 3ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP equivalent motoring condition

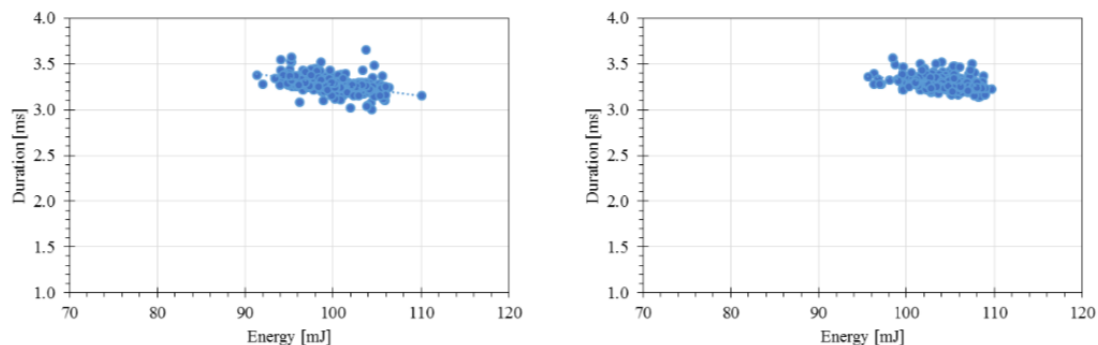


Figure 5.19 - Comparison of SVA Bosch CEI 3ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP

Borg Warner DCI

Dual Coil Continuous (DCC) mode

Borg Warner supplied system had a design which was fundamentally different to that of Stock Coil and Bosch system presented earlier. Its coil pack consisted of two separate coils which could be charged and discharged simultaneously. In this mode coils were discharged alternately in a manner that was overlapping and did not allow for spark to extinguish between discharges. It is shown by the current trace presented in Figure 5.20. Each discharge can be clearly seen as a spike in current trace and was characterised by very high current of 200 mA (vs 100-150 for Bosch system and ~100 mA for Stock Coil). Such behaviour had an impact on power delivered to the spark plug, as it was also exhibiting similar spikes.

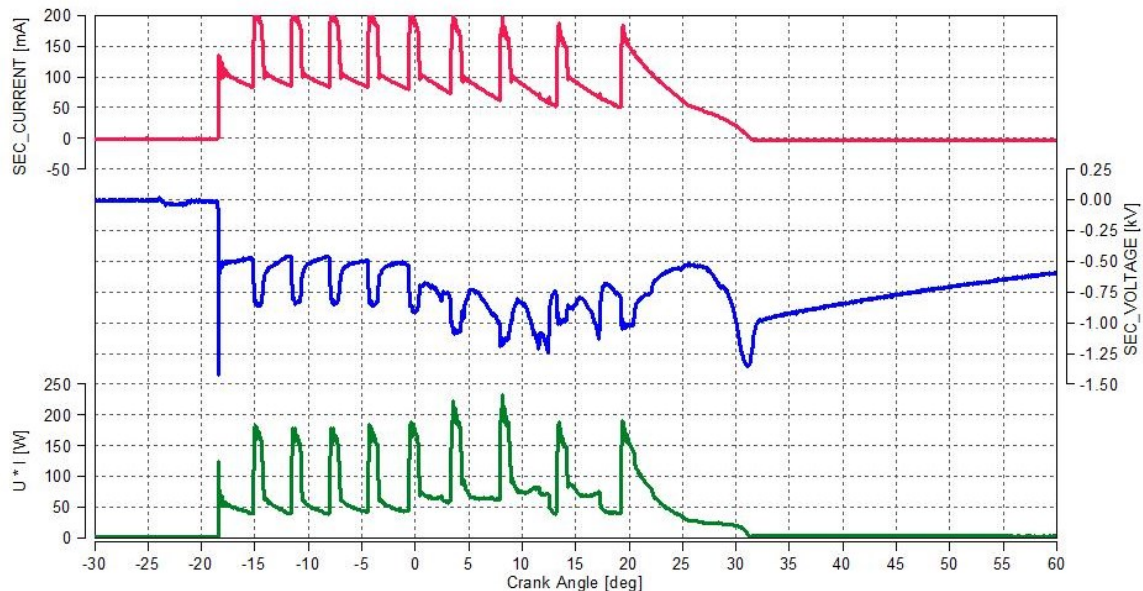


Figure 5.20 - Representation of secondary current and voltage for BW DCI ignition system (5ms ignition duration) running in DCC mode at 1500 RPM, 3.6 bar GMEP, stoichiometric condition

When plots of duration vs energy were plotted (Figure 5.22 – Figure 5.25), it was noted that two clusters of data points appeared. There was always a bigger “cloud” with cycles characterised by high energy per cycle and long duration accompanied by smaller “cloud” with cycles that had significantly shorter duration and only fraction of the energy. Such cycle was presented in Figure 5.21. It appeared that it only had a single restrike from second coil, and failed to deliver subsequent strikes from that coil. It is unknown whether it was due to prototype

nature of the coil pack or could be occasionally expected when running DCI system with maximum demand set on it. It was also noted that such cycles were never consecutive and they occurred randomly without any pattern.

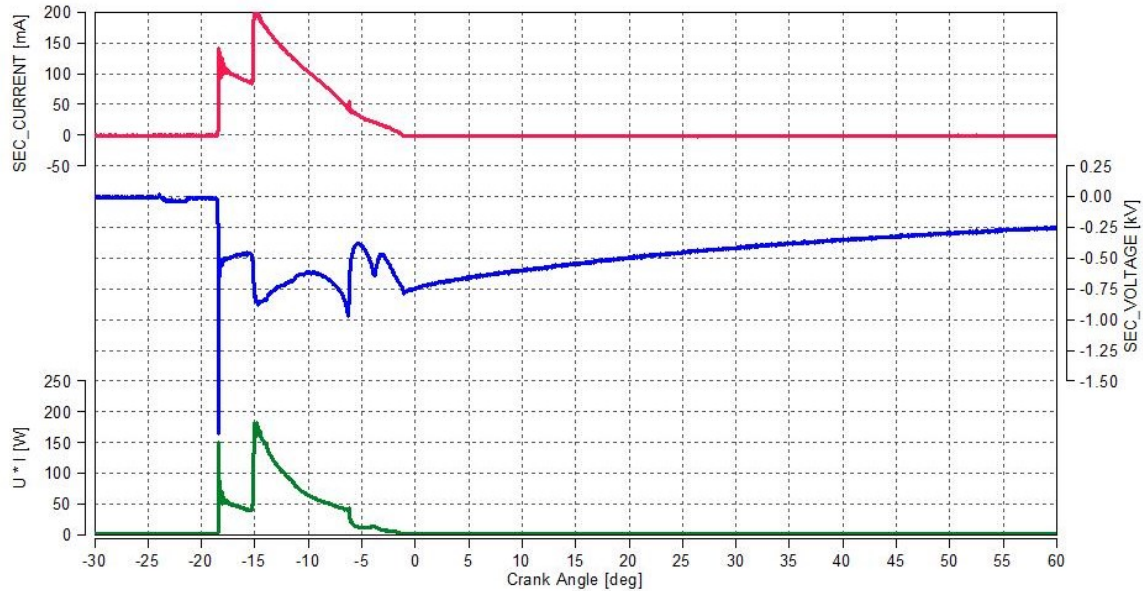


Figure 5.21 - Representation of secondary current and voltage for low energy cycle of BW DCI ignition system (5ms ignition duration) running in DCC mode at 1500 RPM, 3.6 bar GMEP, stoichiometric condition

Once the low energy cycles were considered outliers, new trend emerged that was in opposition to what has been noted earlier for Stock Coil and Bosch CEI. Durations were positively correlated with energy per cycle i.e. higher energy cycles were also longer across all presented cases. Motored SVA condition at $\lambda=1.6$ was again the point that had the biggest spread in data from all eight plots.

Interestingly, under SVA operation (mainly motored) points of medium duration started to appear. They were characterised by lower number of restrikes compared to the high energy cycles, however they achieved more than two presented in Figure 5.21 for low energy cycles clustered in the bottom left-hand corner of all the plots.

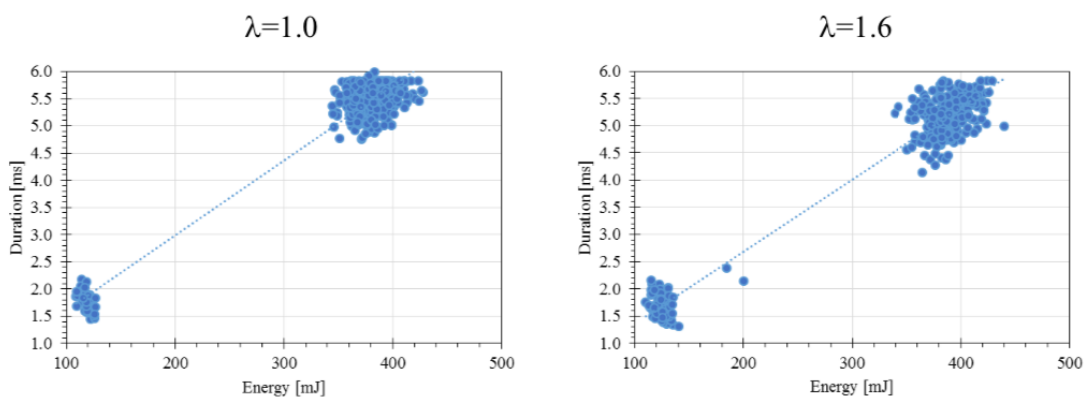


Figure 5.22 - Comparison of DVA BW DCI 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP equivalent motoring condition

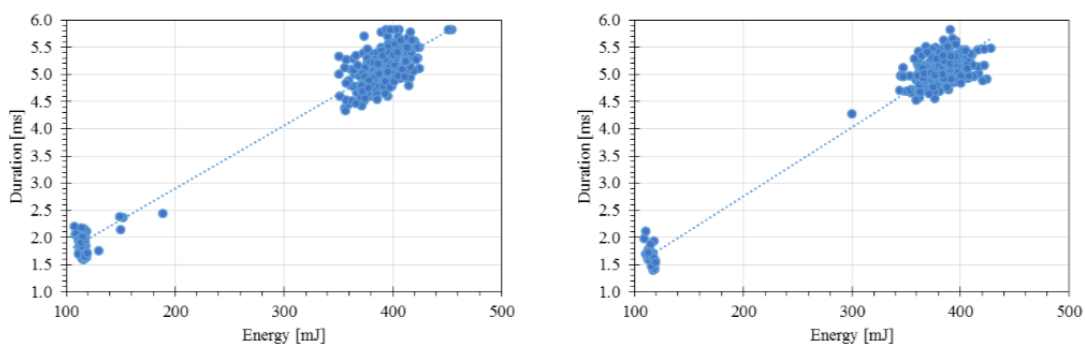


Figure 5.23 - Comparison of DVA BW DCI 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP

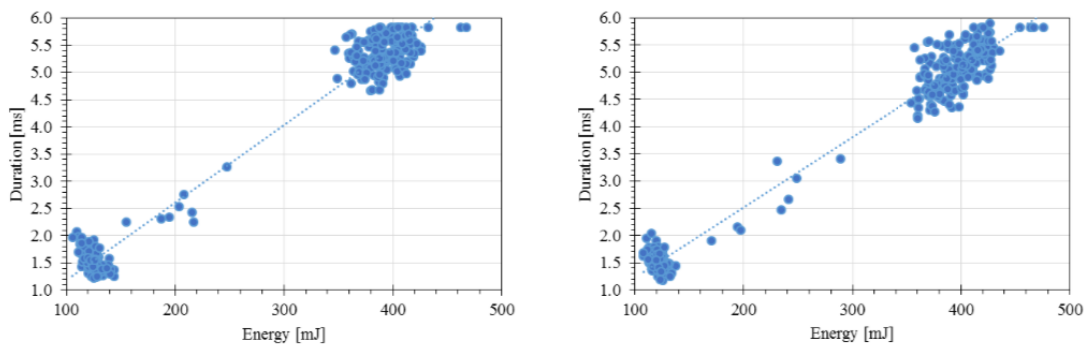


Figure 5.24 - Comparison of SVA BW DCI 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP equivalent motoring condition

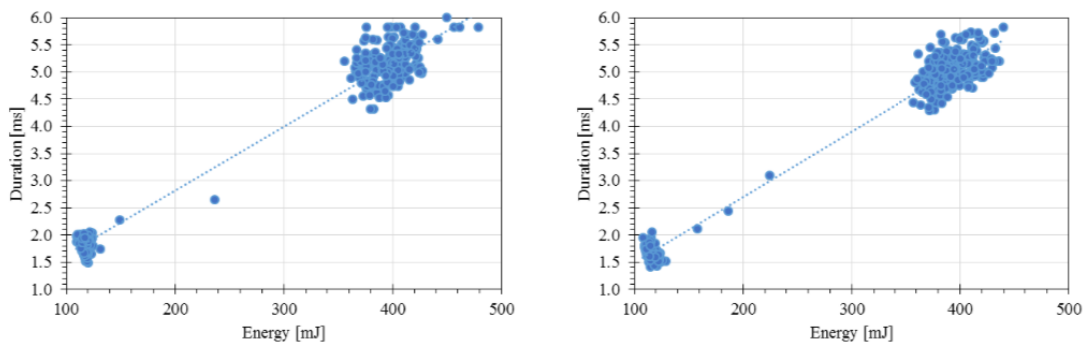


Figure 5.25 - Comparison of SVA BW DCI 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP

Multistrike/ Multispark mode

When set to Multistrike mode, BW system changed its operation and it effectively provided a number of very short ignition events. Figure 5.26 illustrates that with current trace which drops down to zero before next signal is applied and new spike occurs. It is also manifested in power delivered to the spark plug which is being interrupted between each individual restrike event.

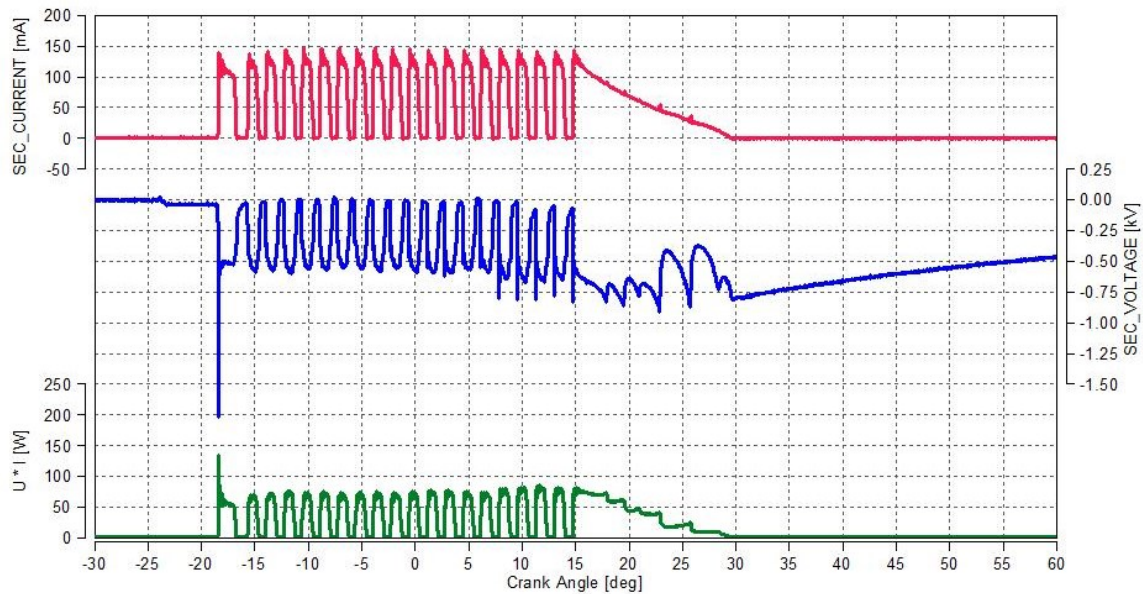


Figure 5.26 - Representation of secondary current and voltage for BW DCI ignition system (5ms ignition duration) running in Multistrike mode at 1500 RPM, 3.6 bar GMEP, stoichiometric condition

Similarly to what has been observed for continuous operation of BW system, a number of low energy cycles were seen for multistrike operation. Average energy delivered in this mode was about 50% lower than that of DDC.

It was observed that when running in SVA mode, average energy per cycle was higher than DVA counterpart.

Also, it was noted that energy vs duration correlation switched again and was in-line with Stock Coil and Bosch CEI characteristics, i.e. clusters of high energy cycles had negative values of Pearson correlation coefficient, showing that shorter ignition events had higher energy.

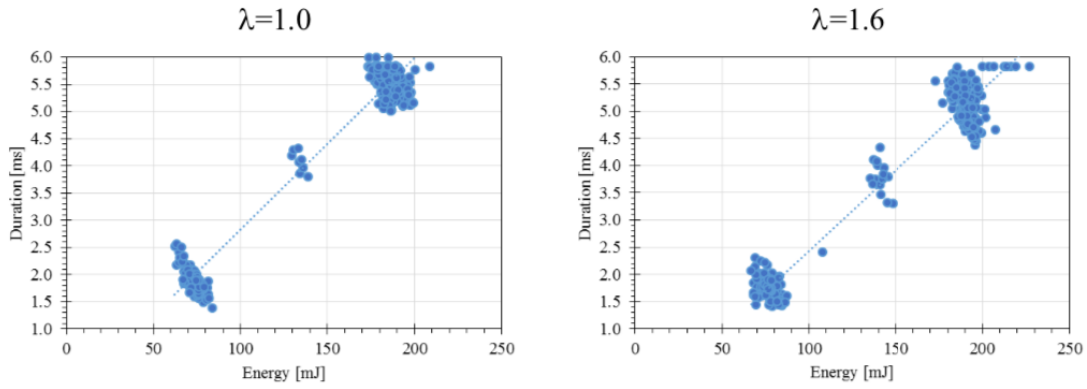


Figure 5.27 - Comparison of DVA BW Multistrike 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP equivalent motoring condition

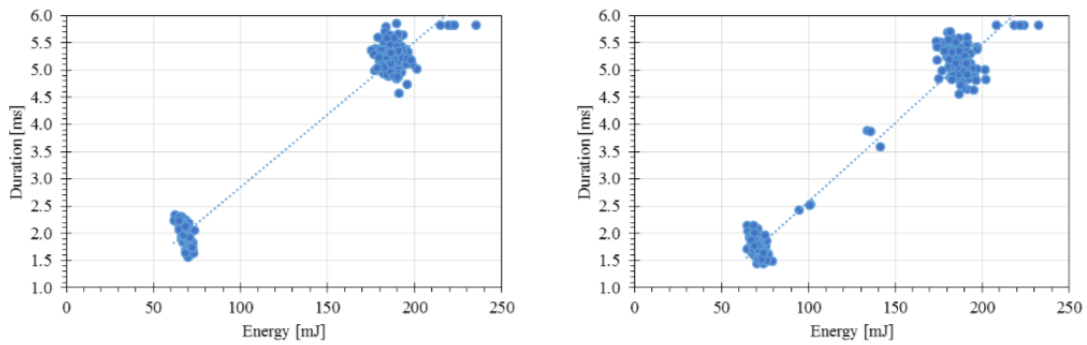


Figure 5.28 - Comparison of DVA BW Multistrike 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP

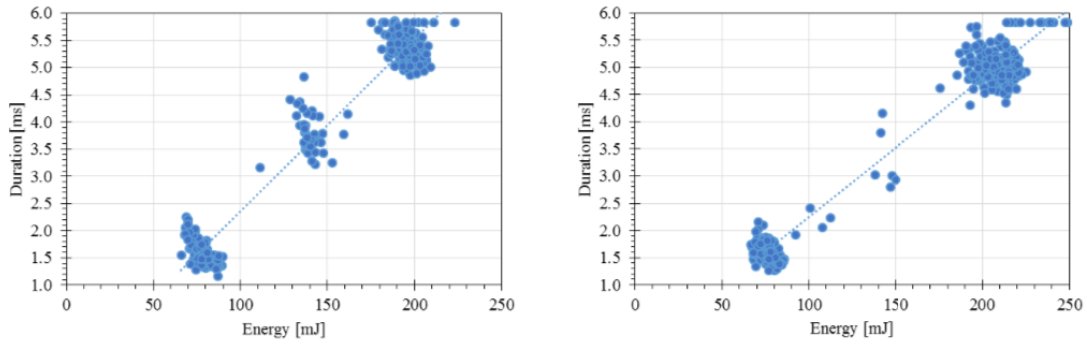


Figure 5.29 - Comparison of SVA BW Multistrike 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP equivalent motoring condition

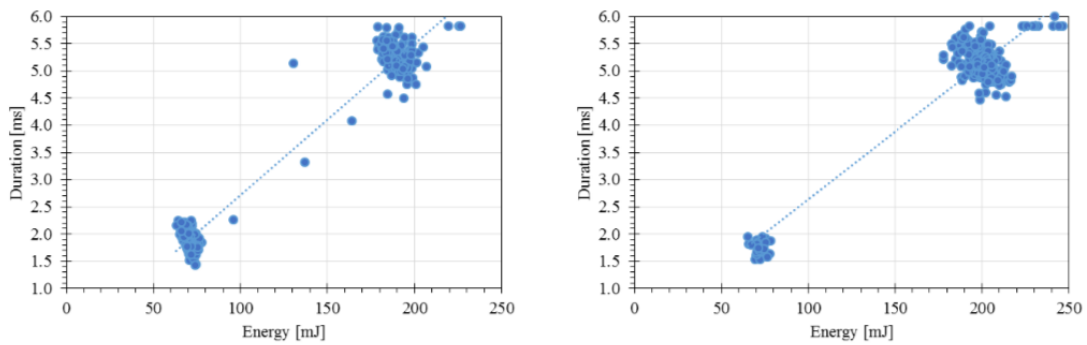


Figure 5.30 - Comparison of SVA BW Multistrike 5ms Energy vs Duration plots for $\lambda=1.0$ and $\lambda=1.6$ at 1500 RPM, 3.6 bar GMEP

5.3.2. Combustion Analysis

DVA

Combustion stability comparison between all five tested systems run with standard DVA operation showed that differences between the systems were moderate across the entire map. Combustion stability deteriorated with introduction of diluents, however there were no major benefits to be seen. Only BW DCI system achieved consistent improvement over stock coil in the most dilute conditions, whilst all the other systems performed similarly (if not worse) to baseline Stock Coil system. This study was more of a fundamental investigation, therefore it was not limited to NVH criteria of 3% CoV of NMEP used earlier in Chapters 3 and 4.

Such results did not meet expectations and they would not warrant advocating that any significant improvements can be made to dilution tolerance of AJ-20 combustion system by using high energy ignition systems.

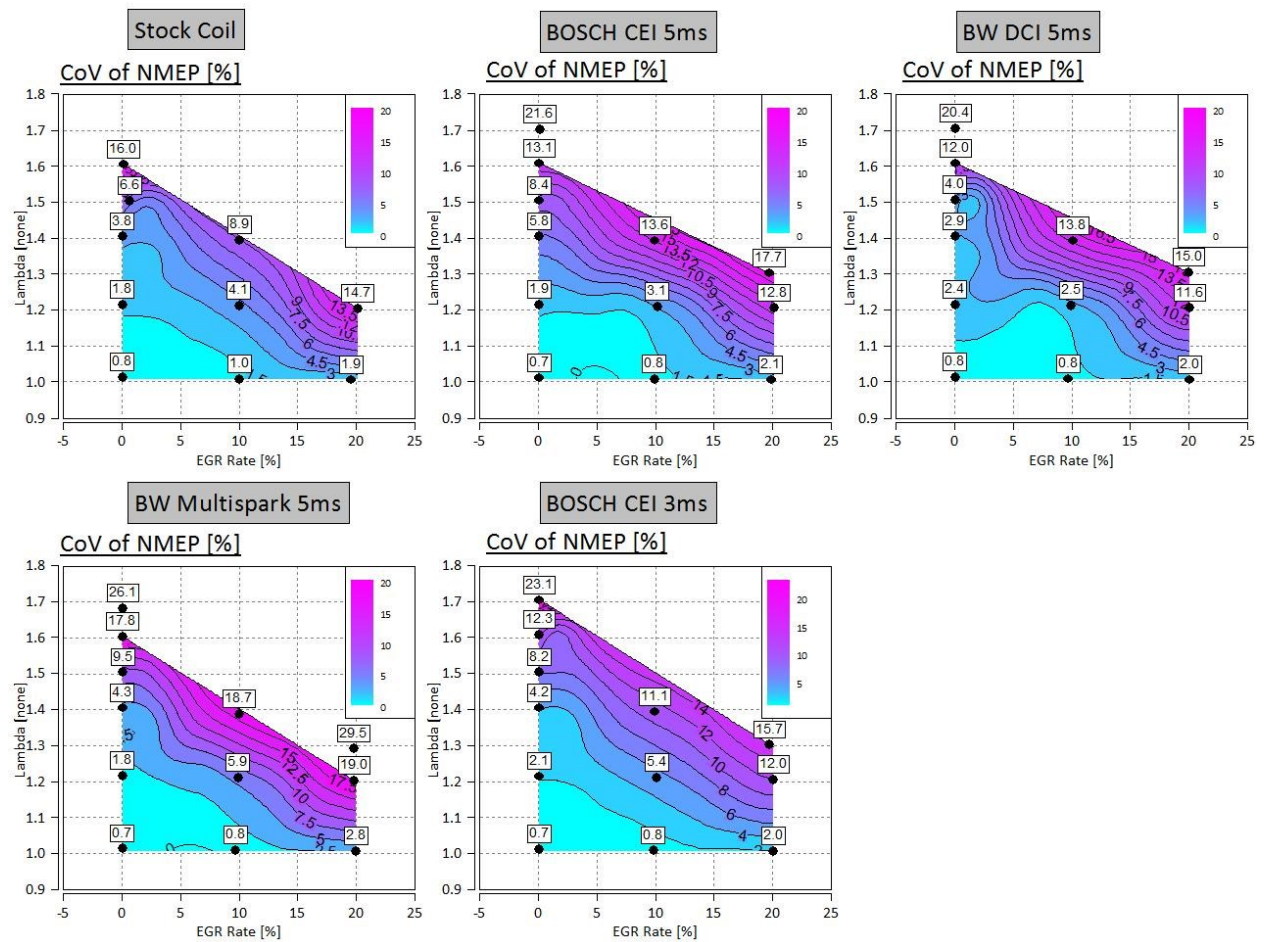


Figure 5.31 – CoV of NMEP comparison for various ignition systems at 1500 RPM, 3.6 bar GMEP condition run in DVA valvetrain mode

Similarly to lack of significant improvements in combustion stability, initial phase of combustion (ignition to 5% MFB) remained largely unchanged between various ignition systems.

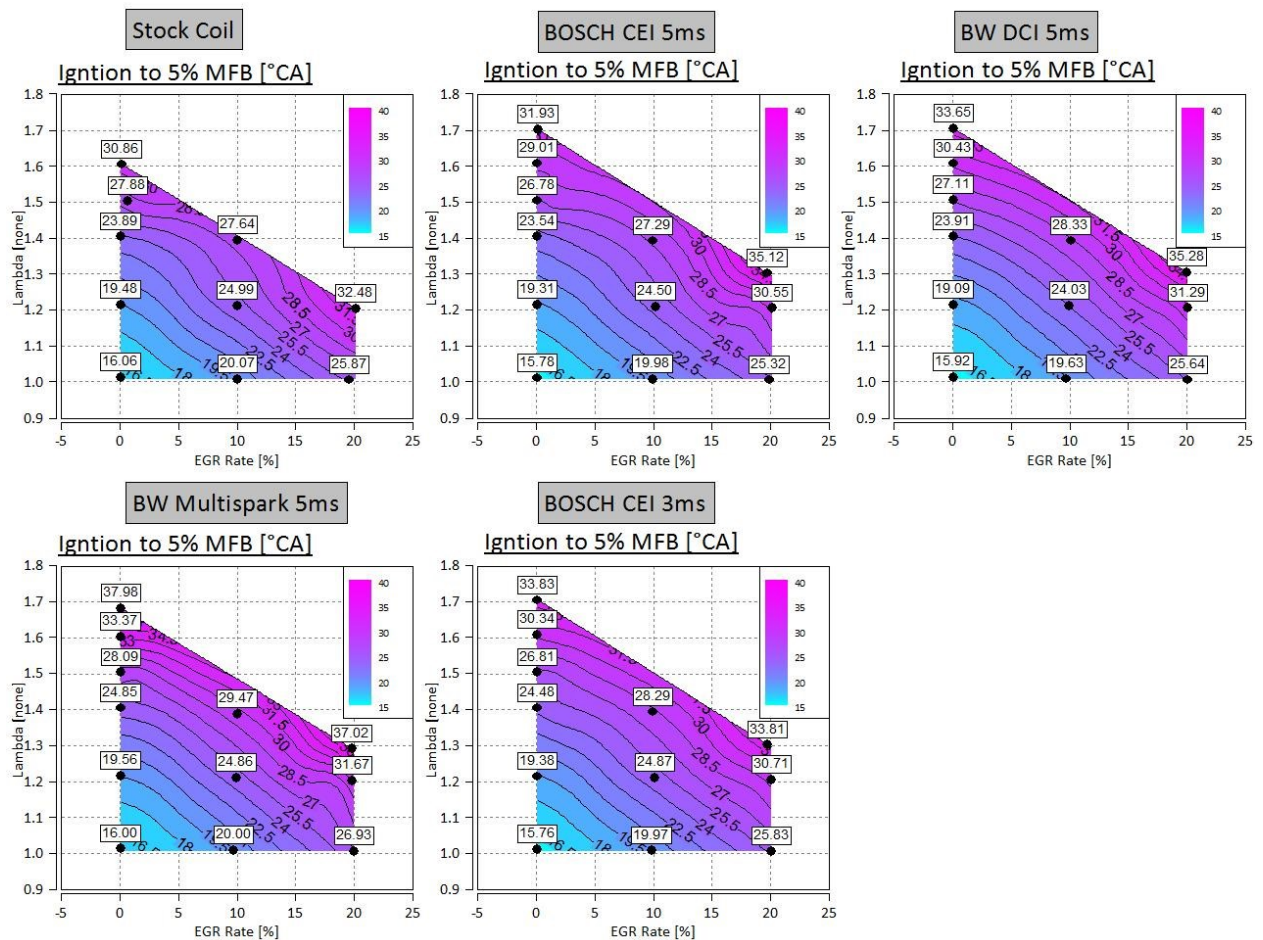


Figure 5.32 – Ignition to 5% MFB comparison for various ignition systems at 1500 RPM, 3.6 bar GMEP condition run in DVA valvetrain mode

Given that test procedure kept the same ignition angle for all tested systems, location of 50% MFB was selected as an indicator of shift within later stages of combustion. If advanced, it would indicate faster burn, however it also remained almost constant for all investigated systems. It should be noted that conditions with severely deteriorated combustion stability cannot be reliably compared due to heat release calculations being affected by partial burn cycles or misfires.

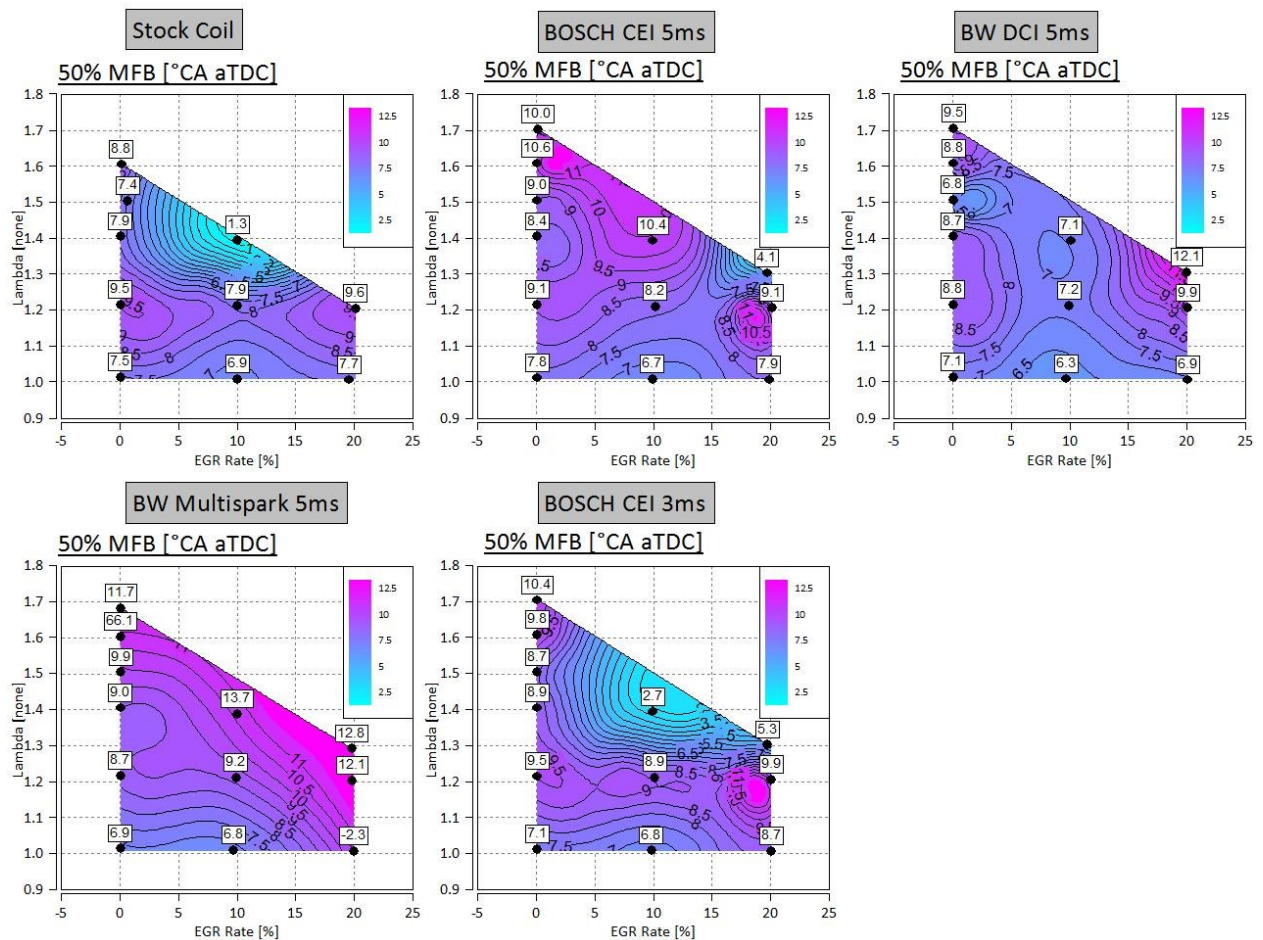


Figure 5.33 - 50% MFB location comparison for various ignition systems at 1500 RPM, 3.6 bar GMEP condition run in DVA valvetrain mode

SVA Results

After the somewhat disappointing results of DVA, same analysis was done for SVA study. It was shown that dramatic improvements were achieved. Big turquoise area in Figure 5.34 for BW DCI 5ms represented conditions that could be utilised for engine calibration exercise (CoV of NMEP $\leq 4\%$). It was the best system by a significant margin. It managed to reduce CoV of NMEP higher than 20% observed for Stock Coil to less than 3.5% at $\lambda=1.4$, 10% EGR. Its application was effective across the whole map, improving stability for both lean and EGR diluted mixtures. BW system run in multistrike mode was however most closely matched to Stock Coil system, offering very little improvement in combustion stability.

Interestingly, two Bosch systems run with different ignition durations were very closely matched. Such result would indicate that optimum length of the spark could be shorter than 3

ms, and increasing it further and further did not bring any benefits. Energy levels of ~100 mJ were sufficient for Bosch system to achieve same results as ~150 mJ operation.

Another conclusion which can be reached based on the combustion stability results of SVA tests, is that characteristics of power delivery to the spark plug is of the primary importance. Multispark system delivered almost double the energy of 3 ms Bosch CEI configuration (~200 mJ vs ~100 mJ), however the effect was much worse. Continuous discharge mode used for both BW DCI and Bosch systems appeared to be preferred over a number of very short strikes, where spark is extinguished in-between. It is however not possible to firmly identify whether it was the level of current sustained in DCI system (approximately twice as high as Bosch system) or restrike character of alternating coils that contributed the most to combustion stability improvements.

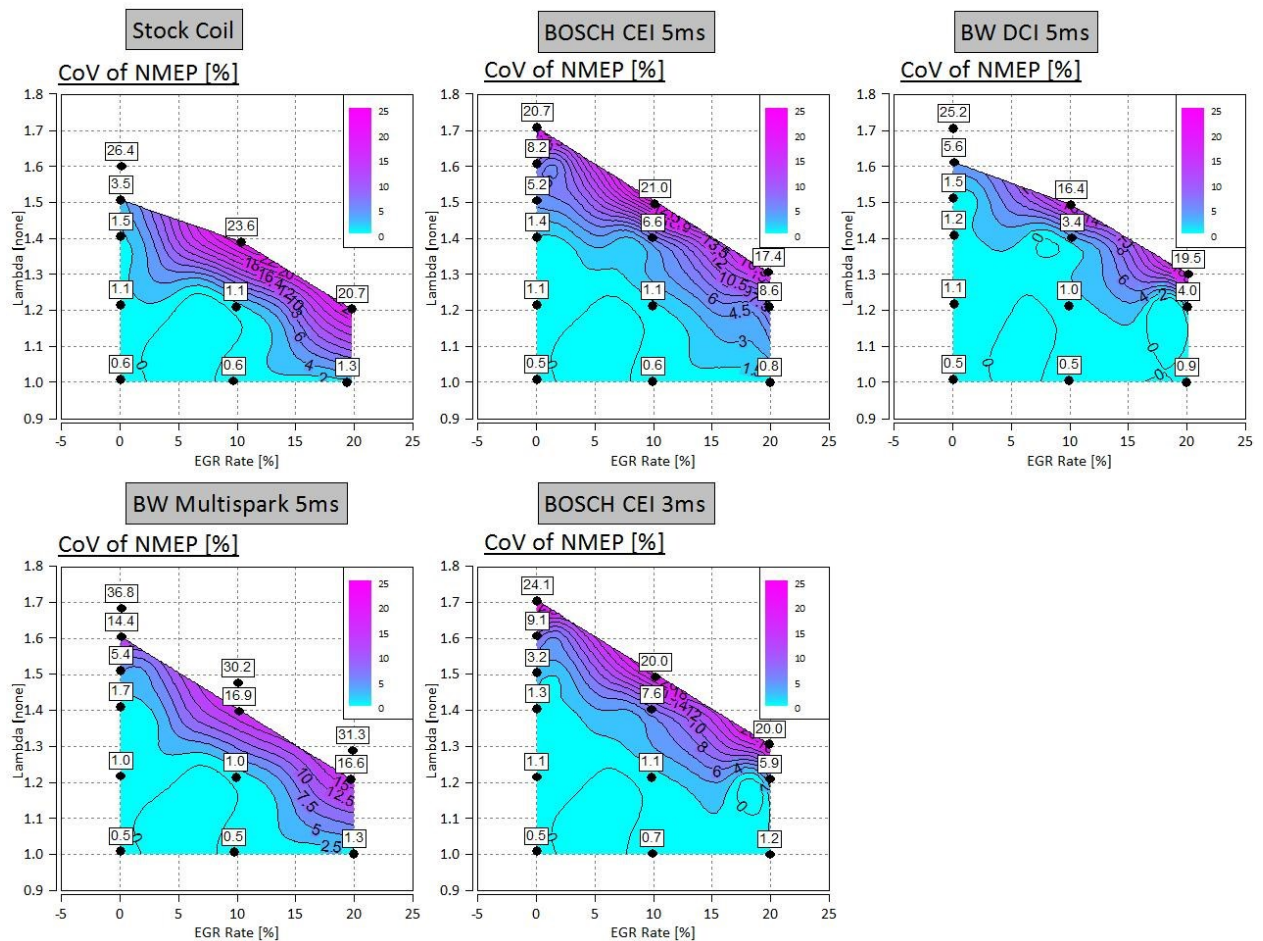


Figure 5.34 - CoV of NMEP comparison for various ignition systems at 1500 RPM, 3.6 bar GMEP condition run in SVA valvetrain mode

Combustion phasing analysis of its initial stages found that ignition to 5% MFB was improved when advanced ignition systems were used. Gains in the initial burn durations were limited to highly diluted conditions, where significant combustion stability benefit was realised. Maximum shortening was limited to approximately 2°CA achieved by the BW DCI system running in continuous discharge mode.

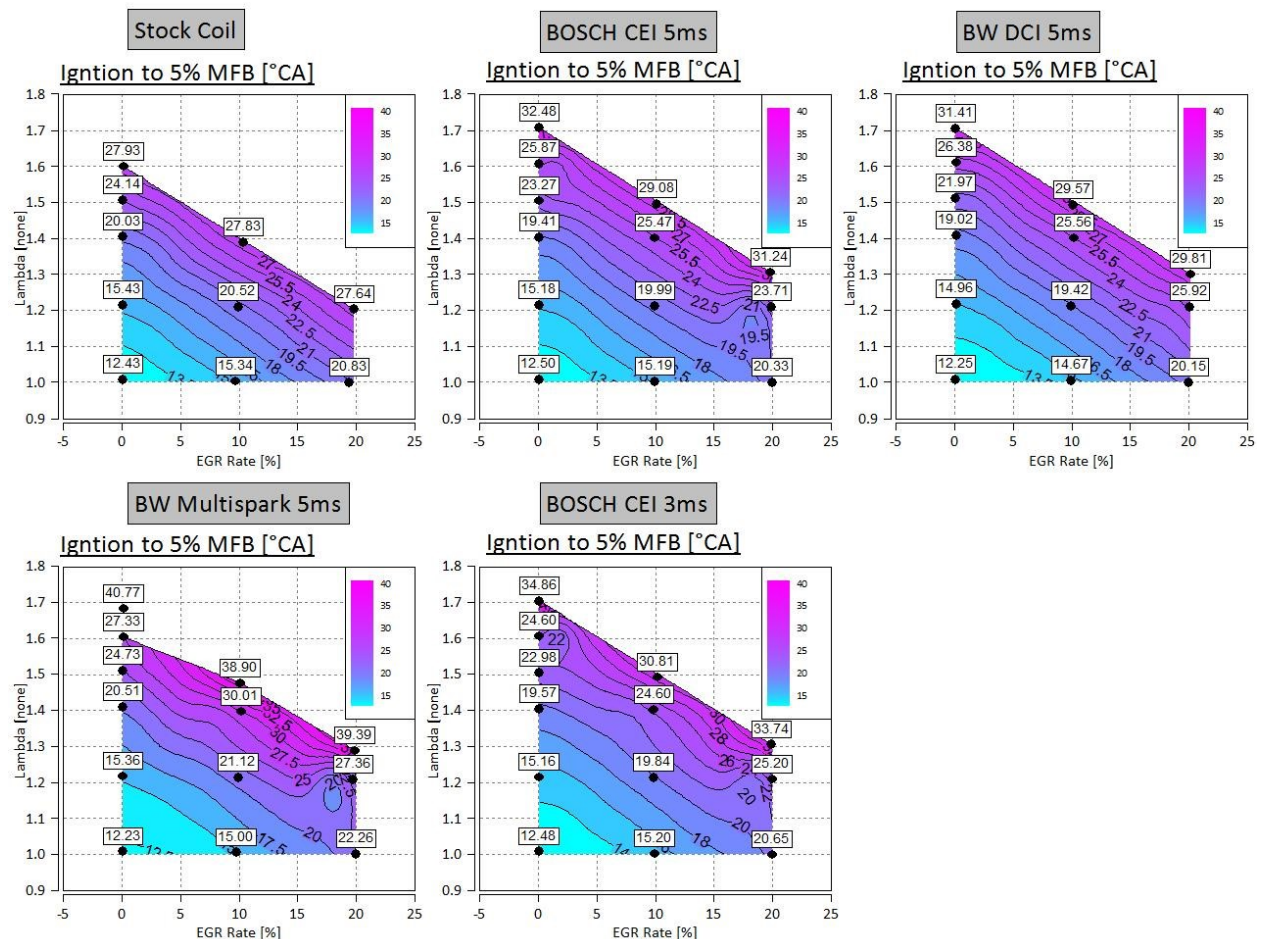


Figure 5.35 - Ignition to 5% MFB comparison for various ignition systems at 1500 RPM, 3.6 bar GMEP condition run in SVA valvetrain mode

Later stages of combustion, represented by location of 50% MFB point (usually used as a target for combustion phasing) showed very little variation and sensitivity to switching ignition systems.

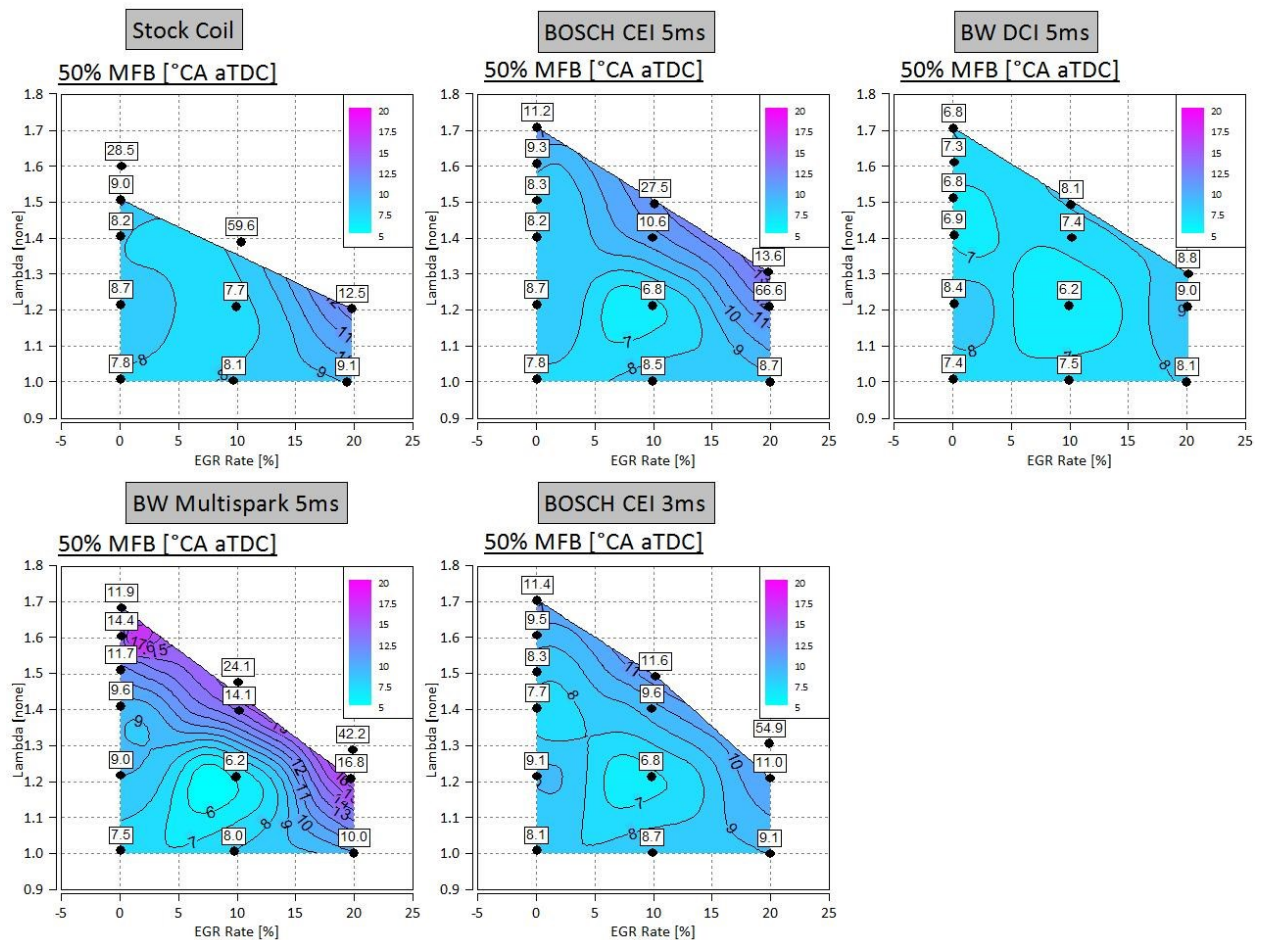


Figure 5.36 - 50% MFB location comparison for various ignition systems at 1500 RPM, 3.6 bar GMEP condition run in SVA valvetrain mode

When comparing results achieved for SVA to these recorded for DVA, it becomes apparent that whilst no recommendation could be made for introducing higher energy systems under standard DVA operation, SVA with its increased in-cylinder motion can realise the synergy effect when pushing the dilution tolerance envelope.

Given that velocity plots for spheres centred around the spark plug gap (Figure 5.6 – Figure 5.9) did not provide results that could be used for any conclusive statements as to why SVA was so much more prone to realising combustion benefits associated with higher energy ignition systems, further studies with the use of optical engine are recommended.

5.3.3. Heat Release Analysis

To further understand how introducing advanced ignition systems influenced combustion and gain insight into why Borg Warner system operated in DCI mode proved to be significantly more stable under diluted running conditions, heat release curves were plotted and analysed for all systems.

First point chosen for analysis was SVA, $\lambda=1.4$ and 10% EGR due to the biggest spread observed in CoV of GMEP (Figure 5.34). At this operating condition Borg Warner system run in DCI 5ms mode exhibited 3.4% CoV of GMEP compared to 6.6% for BOSCH CEI 5ms system, 7.6% for BOSCH CEI 3ms, 16.9% for BW Multispark and 23.6% for Stock Coil.

It can be seen in Figure 5.37 that BW DCI system was characterised by the fastest heat release after spark onset which correlates with the stability mentioned earlier, however the same system operated in Multispark mode was significantly slower than all other ignition modes. Stock Coil and BOSCH system operated in 3ms and 5ms modes had very similar cumulative heat release curves with Stock Coil being only fractionally faster than the other two. The aforementioned heat release results were not well correlated with energy release plotted in the same figure, especially for all systems other than BW DCI.

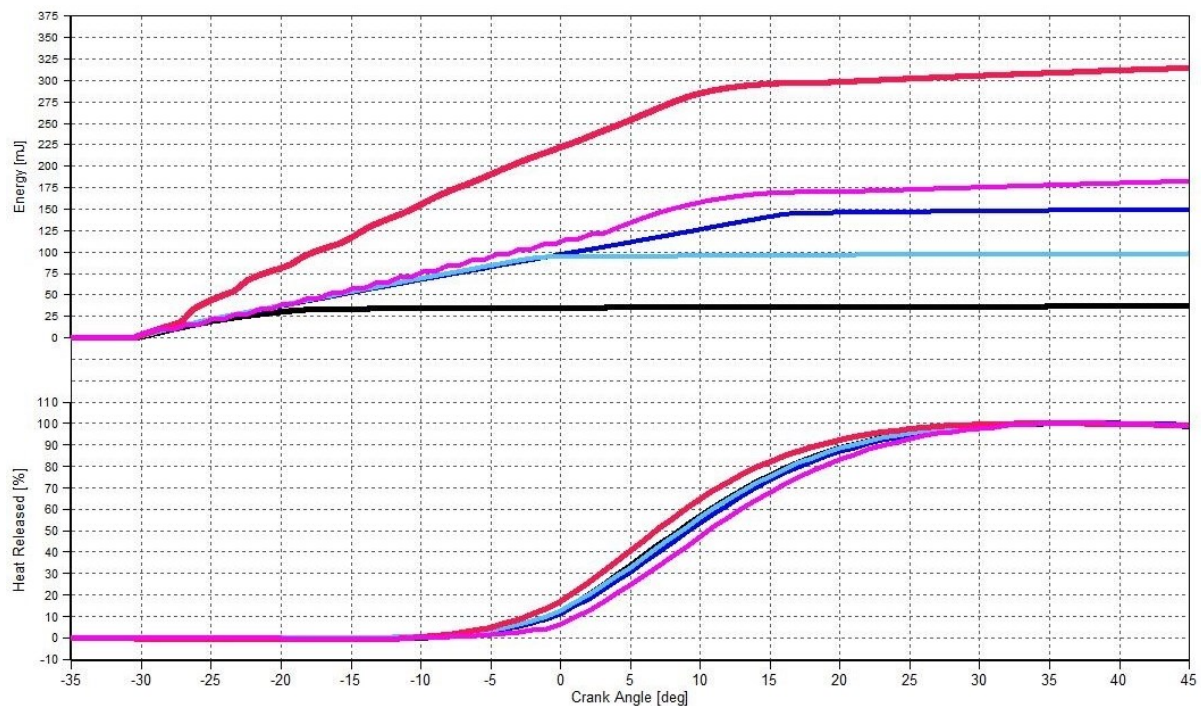


Figure 5.37 – Energy and Heat Release vs Crank Angle comparison for various ignition systems at 1500 RPM, 3.6 bar GMEP condition run in SVA valvetrain mode at $\lambda=1.4$ and 10% EGR dilution

To ensure that results presented earlier were not outliers, further investigation across two more SVA operating conditions was carried out. They were chosen along the excess air only dilution line (0% EGR) - $\lambda=1.2$ and $\lambda=1.5$.

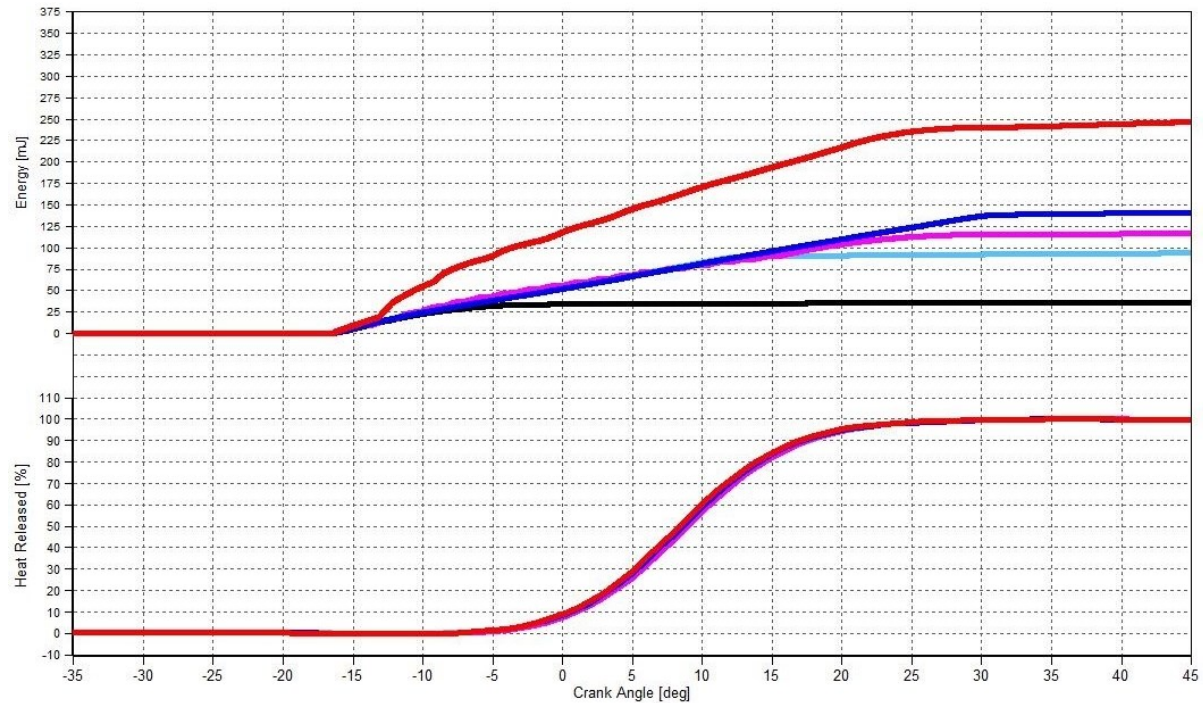


Figure 5.38 - Energy and Heat Release vs Crank Angle comparison for various ignition systems at 1500 RPM, 3.6 bar GMEP condition run in SVA valvetrain mode at $\lambda=1.2$ and 0% EGR dilution

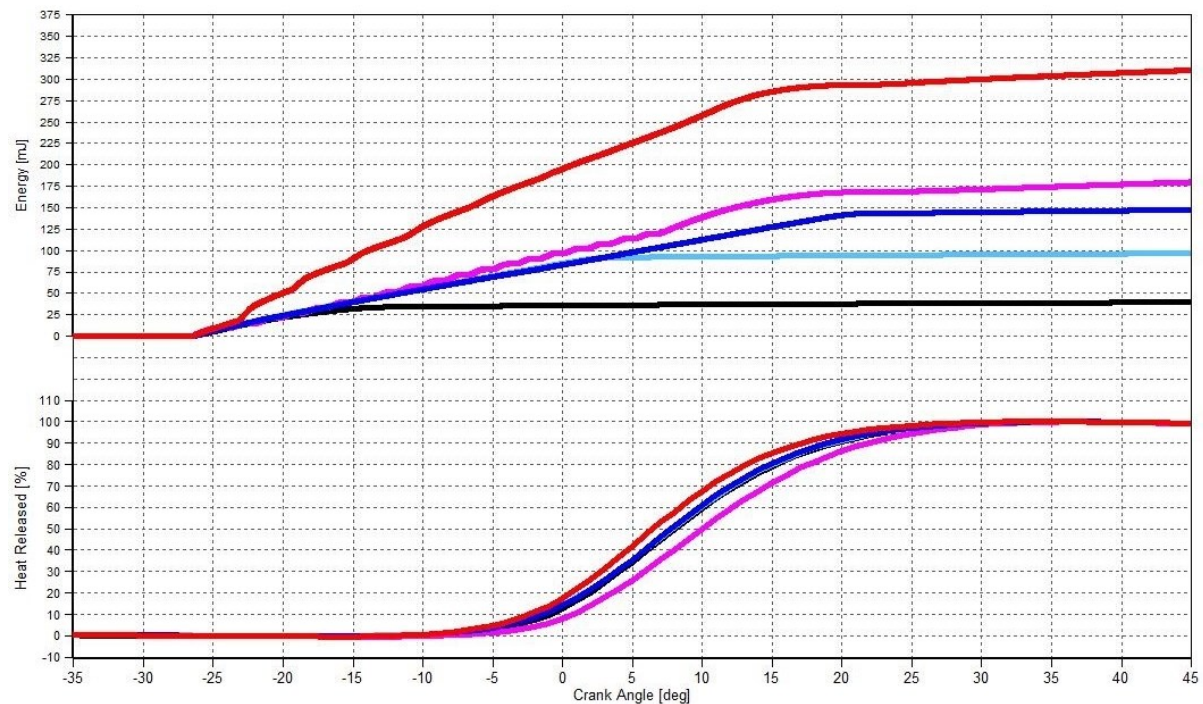


Figure 5.39 - Energy and Heat Release vs Crank Angle comparison for various ignition systems at 1500 RPM, 3.6 bar GMEP condition run in SVA valvetrain mode at $\lambda=1.5$ and 0% EGR dilution

Interestingly, $\lambda=1.5$, 0% EGR point (Figure 5.39) proved to be largely similar in its characteristics to previously analysed $\lambda=1.4$, 10% EGR condition, especially the fastest heat release observed for BW DCI, the slowest for BW Multispark and cluster of both BOSCH CEI operating modes with Stock Coil. This point was also characterised by pronounced differences in combustion stability, ranging from most stable BW DCI with 1.5% CoV of GMEP to least stable BW Multispark and BOSCH CEI 5ms at 5.4% and 5.2% respectively. It confirmed that results shown for $\lambda=1.4$, 10% EGR were not outliers. On the contrary, when dilution was lowered to $\lambda=1.2$ all heat release curves converged almost to a single profile, where BW DCI was only fractionally faster than all the other systems. At this operating condition, all systems were also indistinguishable from the combustion stability perspective – all had stability within 1.0-1.1% CoV of GMEP.

Having observed correlation between stability and heat release profiles for SVA operation, corresponding DVA cases were plotted. It can be seen in Figures 5.40 and 5.41 that differences were less pronounced, with the exception of BW Multispark which had much slower heat release than other systems under $\lambda=1.5$ condition, similarly to previously observed SVA scenario.

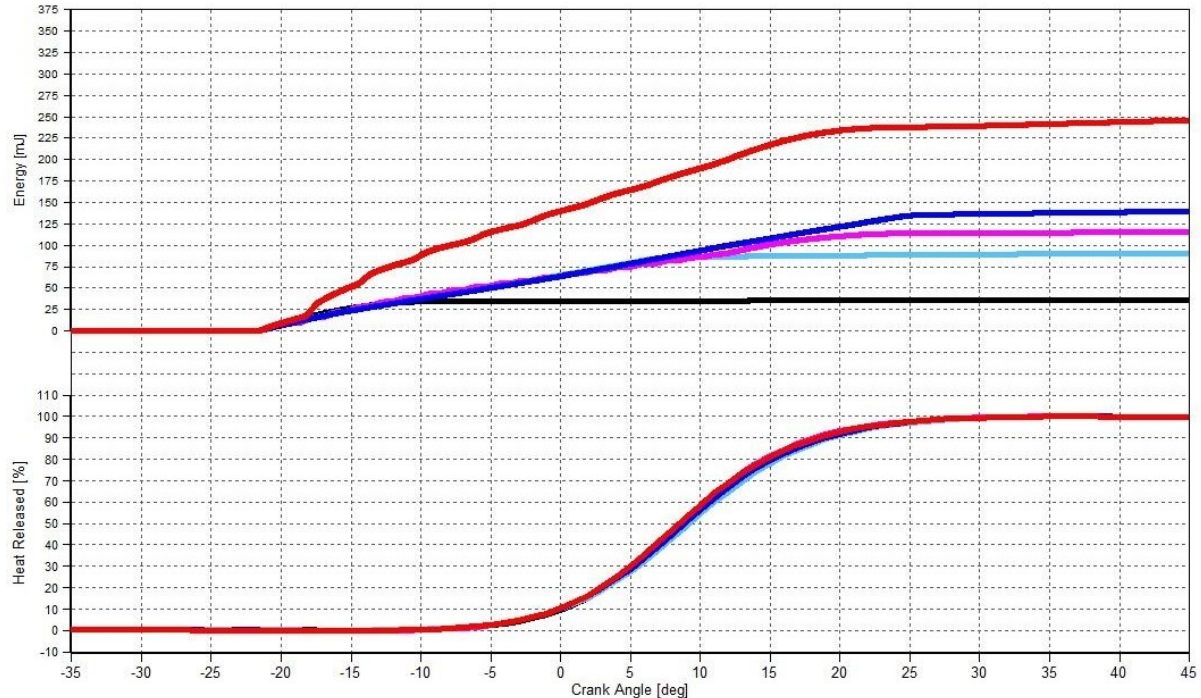


Figure 5.40 - Energy and Heat Release vs Crank Angle comparison for various ignition systems at 1500 RPM, 3.6 bar GMEP condition run in DVA valvetrain mode at $\lambda=1.2$ and 0% EGR dilution

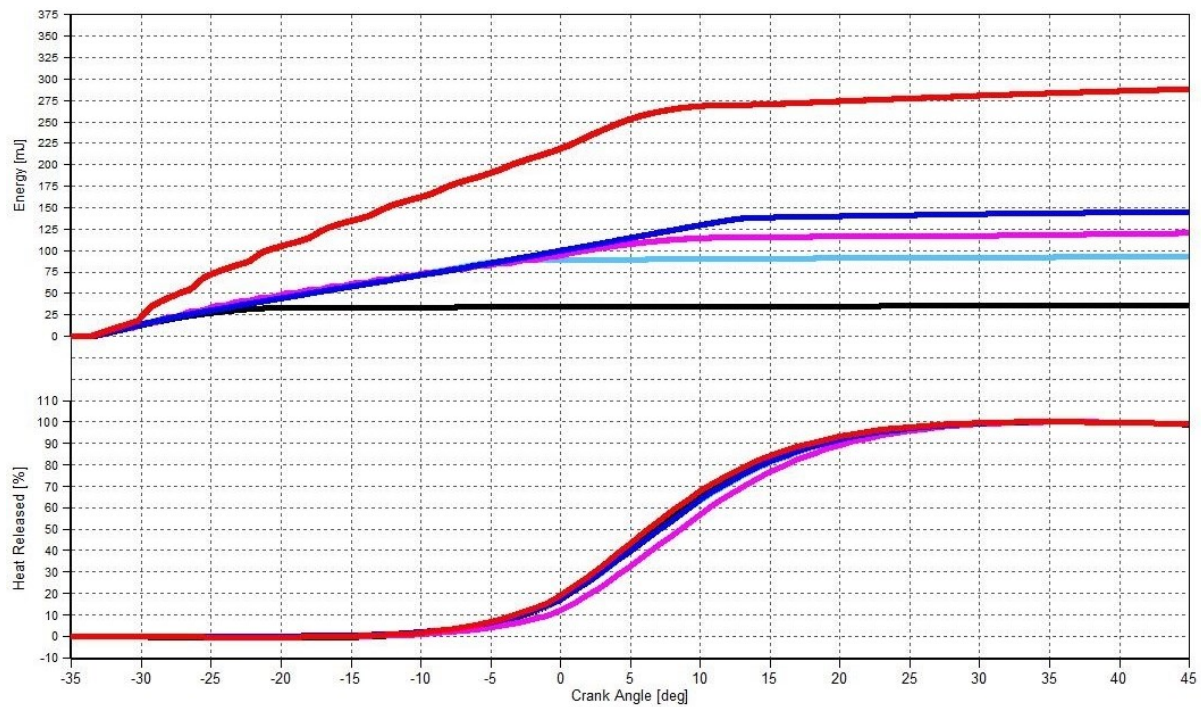


Figure 5.41 - Energy and Heat Release vs Crank Angle comparison for various ignition systems at 1500 RPM, 3.6 bar GMEP condition run in DVA valvetrain mode at $\lambda=1.5$ and 0% EGR dilution

5.3.4. Motored Analysis of DVA vs SVA

As per description of test procedure, an extra dataset was acquired upon completion of both DVA and SVA studies. Few points representative of the DVA strategy were matched using SVA. Matching was completed in two ways outlined below:

- (1) Targeted the same in-cylinder pressure at ignition and the same ignition angle. To achieve that SVA plenum pressure had to be lowered.
- (2) Targeted the same in-cylinder pressure at ignition, but instead of matching the ignition angle, intake plenum pressure was matched and ignition angle was advanced.

Such exercise under motoring conditions enabled generating additional insights into ignition systems behaviour based on the data presented in Figures 5.42 – 5.46.

The same in-cylinder pressure at ignition was reached by either lowering intake plenum pressure for SVA compared to DVA (for the same ignition timing) or advancing spark timing

(for the same plenum pressure). It was attributed to differences in air trapping due to different characteristic of air mass flow through the valves (described in detail in Chapter 3).

Additionally, it was seen that increasing plenum pressure beyond 75 kPa resulted in the most pronounced differences in spark duration. It was always the longest under DVA for all tested systems with exception of BW in Multistrike configuration.

Interestingly, Stock Coil and Bosch systems exhibited negative correlation between spark duration and its energy for DVA conditions, and positive for SVA. Positive correlation was observed for all conditions when using continuous spark BW system irrespective of valvetrain strategy.

BW continuous spark mode was the only system which achieved the highest energy transfer to spark plug for DVA operation. For all the other cases mean energy was the lowest for DVA. Such differences did not have any direct correlation with metrics measured under firing conditions. As described earlier in this chapter, BW system operated in continuous mode outperformed other systems in terms of combustion stability, however the difference was much less pronounced than at SVA valvetrain strategy.

Lastly, the Bosch CEI system set to 3 ms Boost Time presented the lowest delta in spark energy between two different SVA modes (within 2 mJ for all 5 tested conditions).

Stock Coil

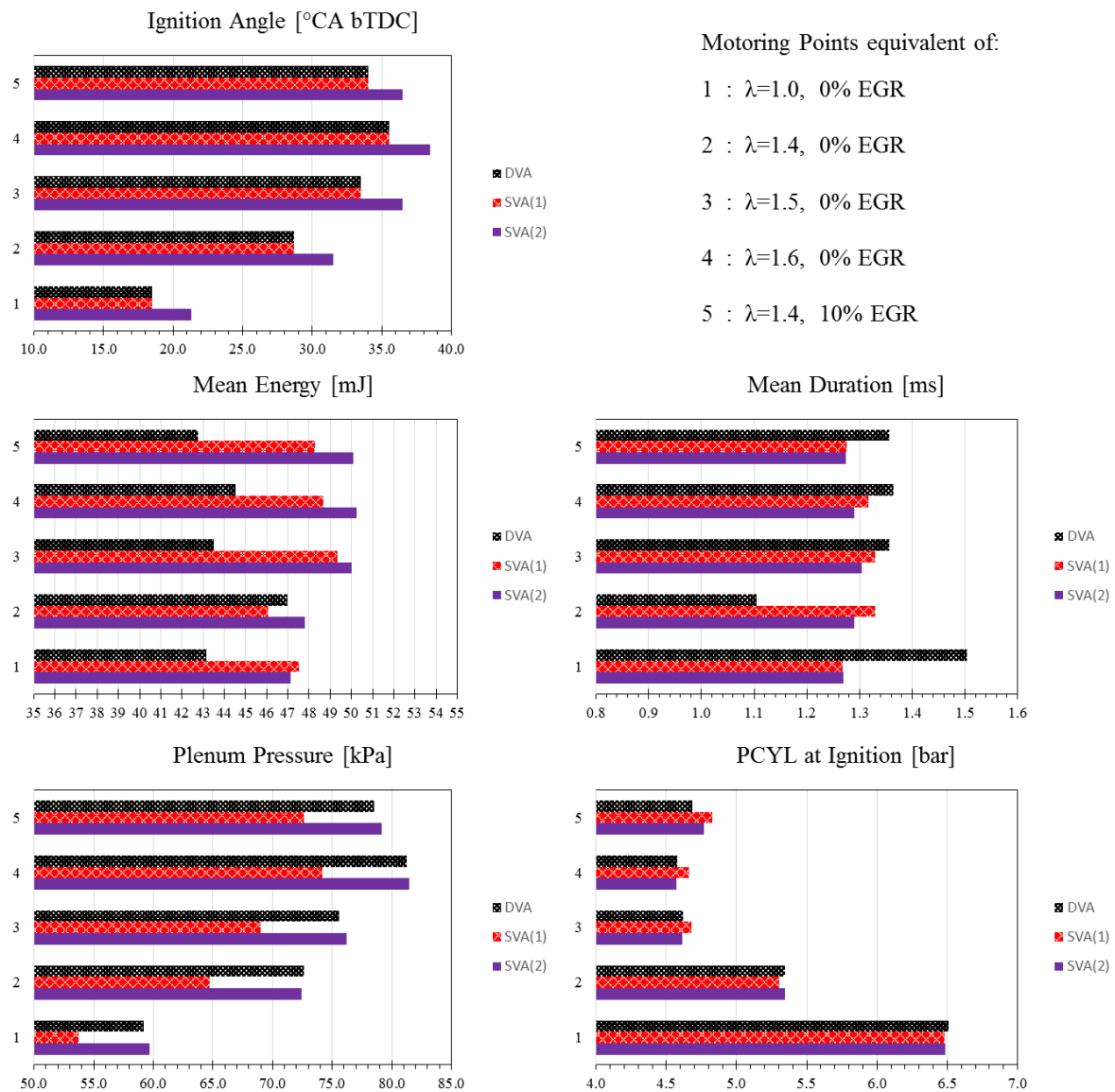


Figure 5.42 – Analysis of Stock Coil DVA vs SVA spark measurements for motored conditions at 1500 RPM

Bosch CEI

5 ms Boost Time duration

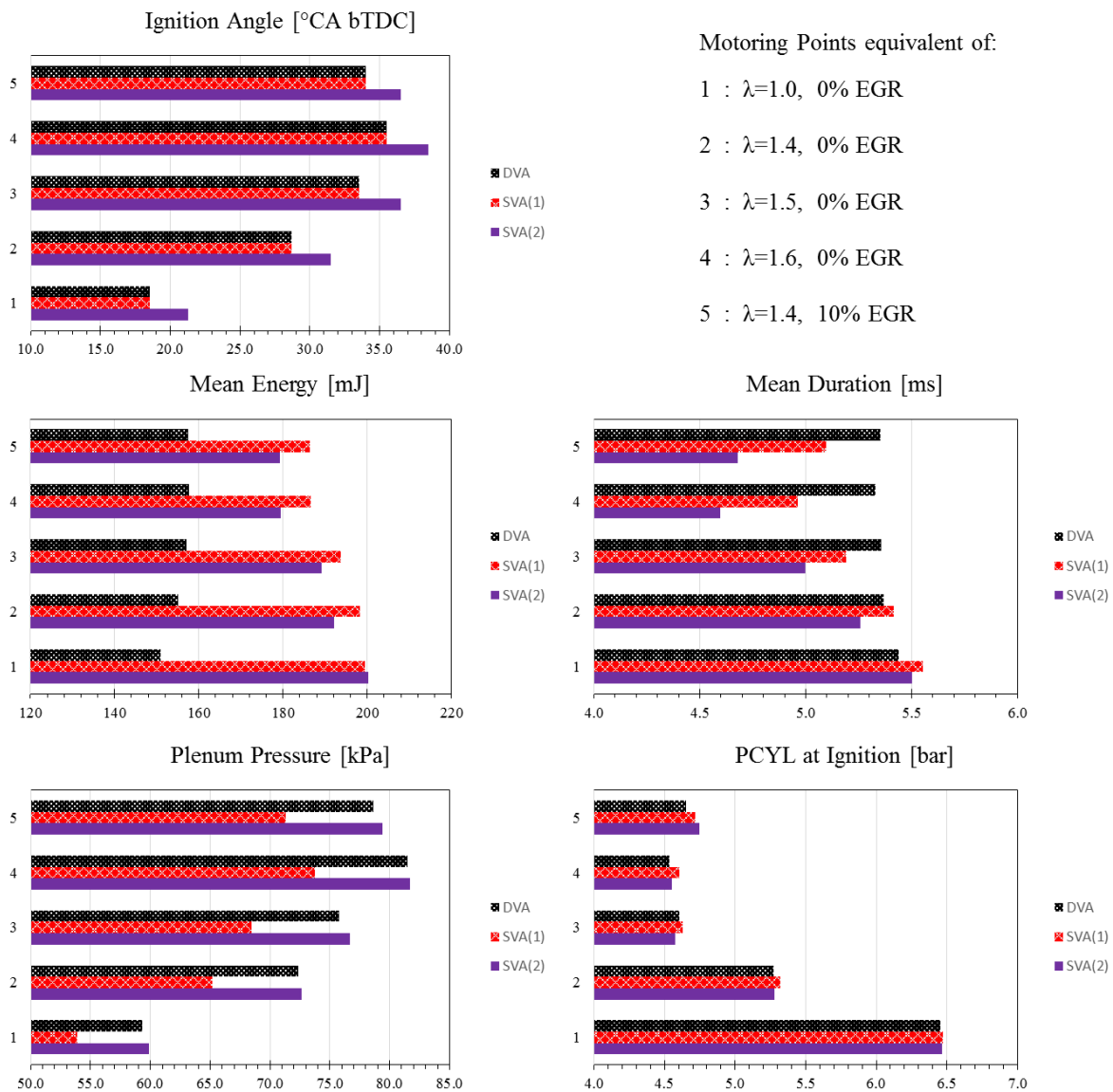


Figure 5.43 - Analysis of 5 ms BOSCH CEI system DVA vs SVA spark measurements for motored conditions at 1500 RPM

3 ms Boost Time duration

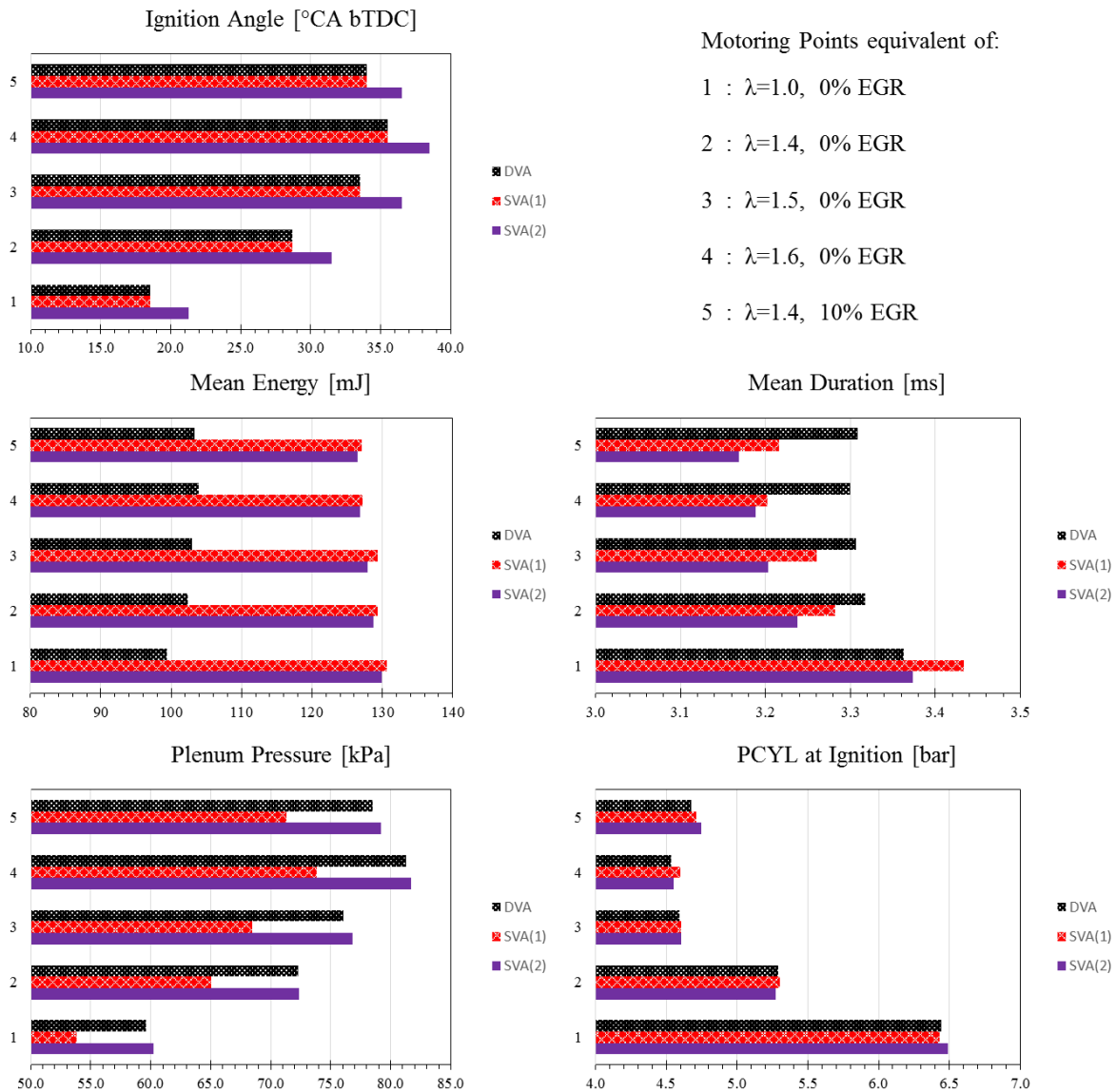


Figure 5.44 - Analysis of 3 ms BOSCH CEI system DVA vs SVA spark measurements for motored conditions at 1500 RPM

Borg Warner DCI

DCI/ DCC Continuous Spark

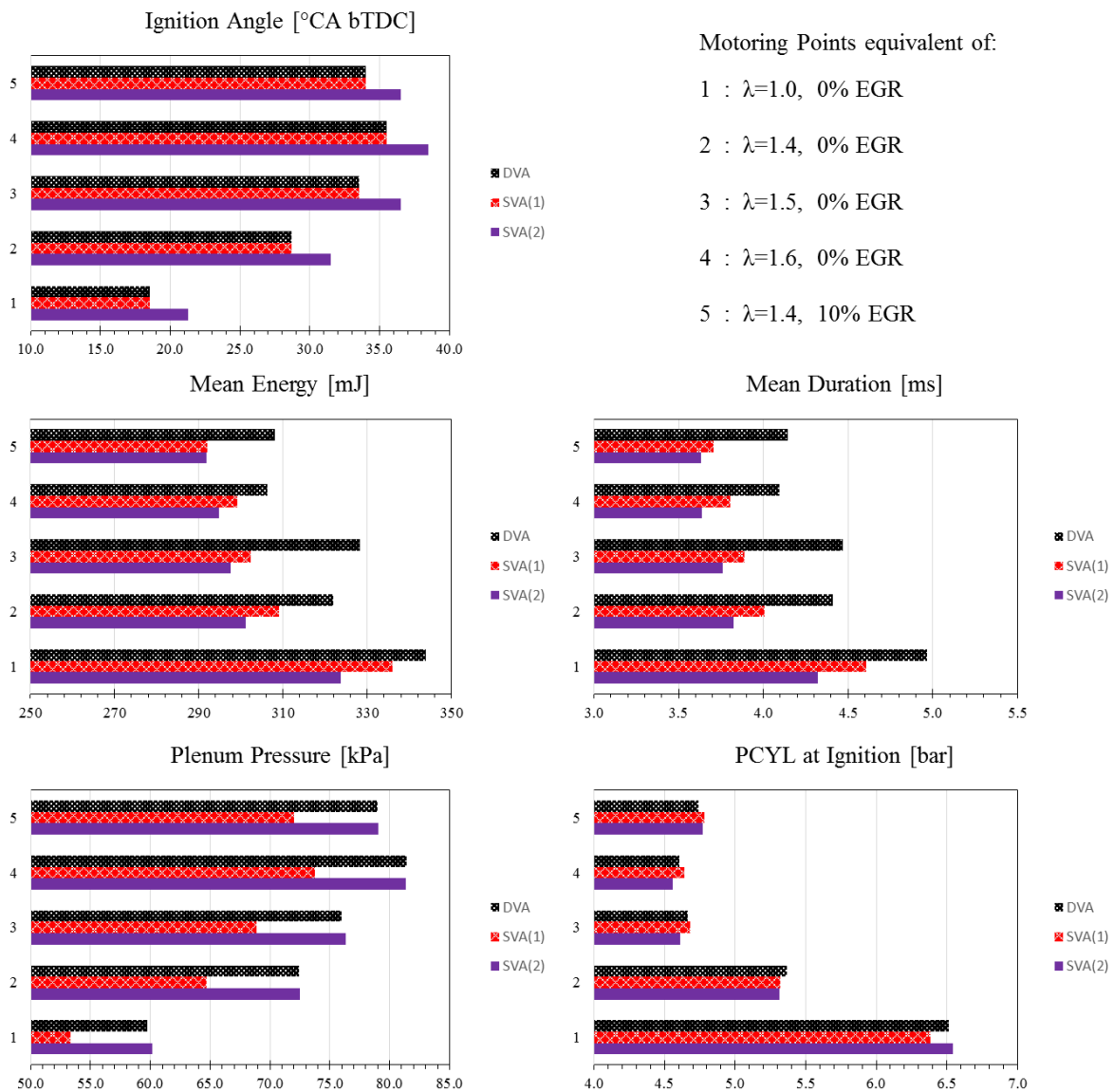


Figure 5.45 - Analysis of 5 ms BW DCI system DVA vs SVA spark measurements for motored conditions at 1500 RPM

Multistrike

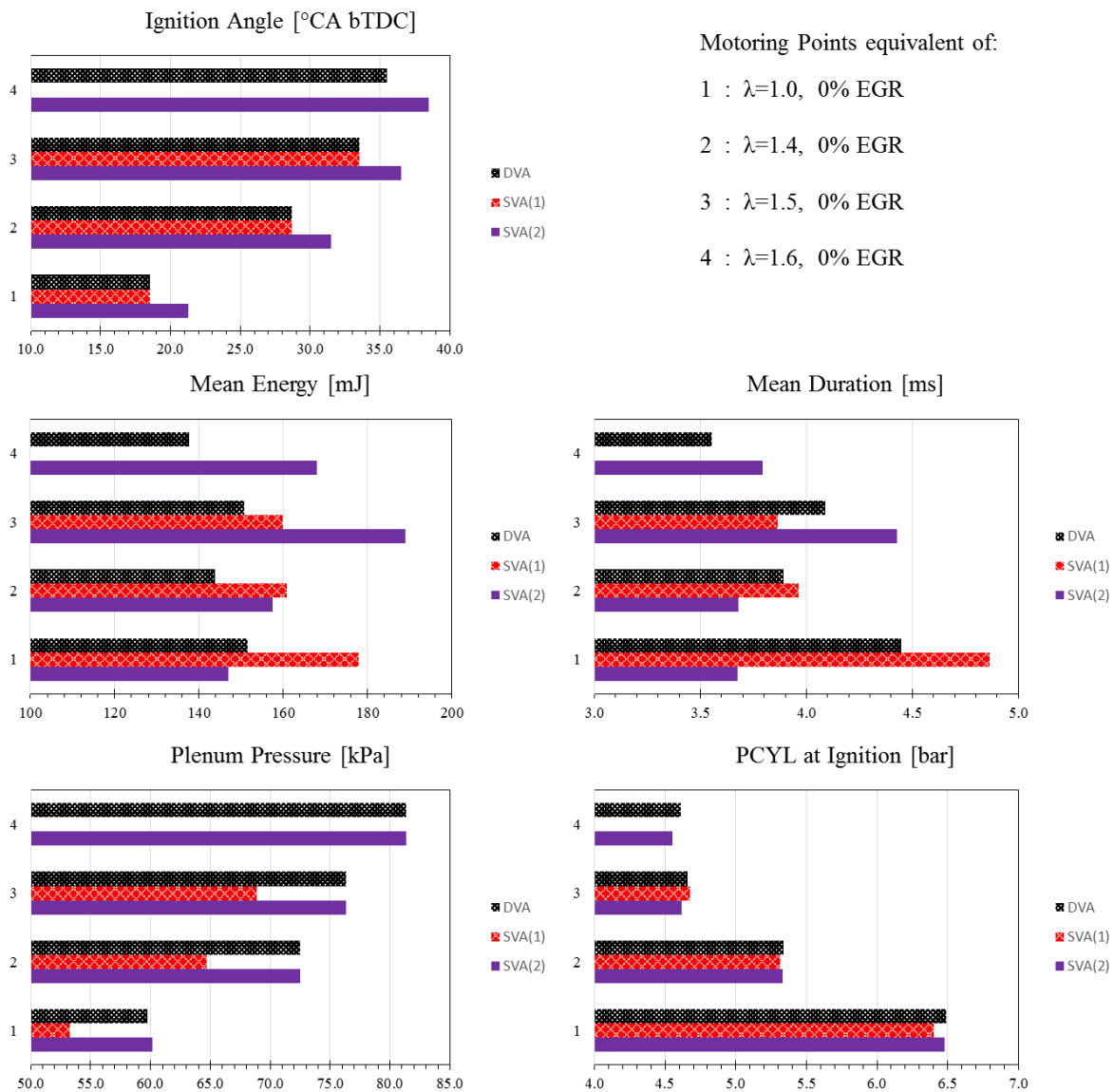


Figure 5.46 - Analysis of 5ms BW Multistrike system DVA vs SVA spark measurements for motored conditions at 1500 RPM

5.4. Concluding Remarks

Ignition measurements were taken using the high frequency system designed and built specifically for this study. It enabled acquiring data at 10 MHz and performing subsequent data post-processing which calculated spark energy and duration.

Three different systems were tested under diluted conditions at 1500 RPM, 3.6 bar GMEP in total of five various configurations. Conclusions reached after completing the analysis of both fired and motored conditions are as follows:

- All systems showed correlation of spark duration and energy. It was positive for Borg Warner continuous spark mode indicating that longer spark durations were characterised by higher energy delivered to the spark plug. The complete opposite was observed for all other systems, indicating that high energy cycles were more likely to shorter than others.
- SVA running condition was found to be significantly more sensitive to improvements in combustion stability when aided by higher energy ignition than DVA.
- The highest improvements in combustion stability and extension of dilution limit were achieved for Borg Warner continuous spark mode. Regardless of the fact that it delivered the highest total energy, it is believed that the benefit should be attributed to mode of delivery (quality), rather than absolute energy values.
- Borg Warner system operated in 5ms DCI mode proved to provide the fastest heat release among all systems. Difference became the most pronounced under diluted conditions ($\lambda \geq 1.4$), where it also translated to combustion stability benefits.
- Under stable operating conditions with little dilution ($\lambda \leq 1.2$), the entire range of ignition systems exhibited very similar heat release profiles, regardless of the differences in energy being dissipated at the spark plug. These were also characterised by almost equally stable combustion (CoV of NMEP below 3%).

Results presented in this chapter were not yet conclusive and further examination using optical engine is needed.

Chapter 6

Conclusions and Future Work

6.1. Summary and Conclusions

The automotive industry is under pressure from regulating bodies imposing strict emission rules and customers demanding fuel-efficient cars. It has to innovate and bring to market vehicles powered by highly efficient powertrains. Charge dilution and homogenous lean engine operation were identified to be attractive strategies of high potential in increasing gasoline engine efficiency. The work presented in this thesis was focused on addressing how charge dilution methods can be utilised to meet such requirements with use of advanced valvetrain and high energy ignition systems. Effects of using these technologies as tools designed to increase charge dilution were presented and discussed in detail.

A single cylinder research engine test facility was designed and commissioned at Imperial College in collaboration with the project sponsoring company – Jaguar Land Rover. It was built as a highly customisable system for proof of concept engine testing. Work was divided into three main parts, and main conclusions for each are summarised below:

Valve Deactivation

- Experimental data proved that SVA strategy was an effective mean for increasing dilution tolerance (both running lean and feeding external EGR) irrespective of engine speed and load. It was found that deactivation of one valve led to increased TKE at ignition which was the governing parameter for speed of flame propagation and combustion duration. As a result, significantly shorter combustion was observed under SVA operation across all running conditions. Majority of the difference between SVA and DVA was manifested in the initial stage of combustion (0-10% MFB), whilst later stages of the combustion process were less sensitive (10-90% MFB).
- SVA valve action trapped more residuals within cylinder and was characterised by higher in-cylinder temperatures.
- Heat transfer effects were found to be offsetting faster burn for SVA. Increased heat transfer resulted in lack of fuel consumption benefit under stoichiometric conditions and marginal GSFC improvements for equivalent dilution levels under most conditions.
- NVH derived 3% CoV of NMEP combustion stability target was reached with lower dilution levels for DVA compared to SVA. Once the engine was operated targeting the limit, fuel consumption benefit (GSFC) of SVA was realised and it was quantified as 3.7% at 1500 RPM, 3.6 bar GMEP, 3.1% at 1500 RPM, 7.9 bar GMEP and only 0.2% (negligible) at 2000 RPM, 9.8 bar GMEP.
- Heat transfer analysis identified two main drivers affecting the heat transfer differences between DVA and SVA – averaged in-cylinder temperature delta between the two and flow field effects (higher gas velocity close to the engine walls). Temperature difference was found to be dominant effect on total heat transfer delta between DVA and SVA for low load, whilst flow field effects become prevailing mechanism for higher loads. Unfavourable heat transfer effects were experimentally captured through compression polytropic coefficients calculations that were lower for SVA at any given dilution.
- Heat transfer delta between DVA and SVA was increasing both in absolute and relative terms with load increase. Under stoichiometric conditions it was found to be 500 W (23.6% increase), 1500 W (50% increase) and 3000 W (56% increase) for 1500 RPM, 3.6 bar GMEP; 1500 RPM, 7.9 bar GMEP and 2000 RPM, 9.8 bar GMEP respectively.

- SVA was found to have more homogenous mixture than DVA and it lowered THC emissions. Similarly, THC emissions were also affected by higher fuel impingement on the piston during the intake stroke for DVA.
- Introducing SVA operation was found to have undesirable effect on NO_x emissions.
- Zeldovich mechanism of NO_x production in post-flame gases was studied and it was shown that temperature plays a critical role due to its exponential term. It was found that at 1500 RPM, 7.9 bar GMEP increasing lambda by 0.1 (vs stoichiometry) resulted in the in-cylinder peak temperature being lowered by about 100K (~4%) and approximately 80% reduction in the temperature component within the NO_x formation equation. Temperature contribution was also found to be very similar for both DVA and SVA.

Staggered Lift

- LIVO (regardless of the maximum lift) resembled SVA running condition studied through valve deactivation package. It was caused by the similarities in flow forming during the induction stroke. Second valve always remained shut in the beginning, and was only opened halfway through induction stroke at the earliest for 5 mm scenario.
- All LIVO lifts on the second valve were characterised by significantly increased TKE during ignition, leading to faster burn and improved combustion stability for diluted mixtures when compared against DVA.
- EIVC strategy was more closely matched to DVA for maximum second valve lifts of 4 mm and 5 mm. It was caused by the initial part of the induction stroke being subjected to both valves opening, hence creating in-cylinder flow field matched to that of DVA. TKE at ignition was actually lower than that of DVA which resulted in slower combustion.
- Lower TKE at ignition for higher lifts (4 mm, 5 mm) was attributed to reduced swirl motion. Second valve opened late during induction stroke when jet velocities already decreased and strong swirl motion was unable to be generated. EIVC was subjected to lifts of 3 mm or lower, second valve closed early enough to generate swirl motion that increased TKE levels above that of DVA at ignition and improved burn durations.
- Heat transfer penalties (compared to DVA) were significantly different when employing LIVO or EIVC. For lifts of 3 mm and higher relative heat transfer penalty was more than twice as high for LIVO compared to EIVC.

- Maximum heat transfer reduction of approximately 55% (compared to SVA) could be achieved without negatively affecting combustion metrics and making them inferior to standard DVA condition.
- Overall indicated engine efficiency was significantly increased compared to baseline, stoichiometric DVA operation as well as optimised DVA and SVA diluted conditions. It was found that 3 mm maximum lift EIVC strategy was optimum for both load cases (3.6 and 7.9 bar GMEP) at 1500 RPM. It achieved gain of 7.1% and 12.7% for these two loads.
- Best overall efficiency reached for EIVC vs LIVO (in contrary to GSFC benefit) indicated that EIVC strategy had less pumping losses, however the difference was marginal ($>0.5\%$).
- Staggered lift cases optimised for fuel consumption were located at much higher lambda values for a given EGR rate (vs DVA), and exhibited overall dilution levels similar to that of SVA.
- Emissions formation was found to be more sensitive to EIVC strategy rather than LIVO, especially at increased loads. Significantly altered charge motion within the cylinder resulted in higher piston wetting and THC formation compared to LIVO. Fuel impingement on the piston was also found to be increasing as second valve lift increased for EIVC with lifts of 2 mm and higher.
- Introducing staggered lift resulted in NO_x production rates lower than SVA, but higher than DVA. Generally, as the maximum lifts were increased, EIVC started approaching lower limit set by than DVA faster than LIVO of equivalent lift.

Spark measurements

- All systems showed correlation of spark duration and energy. It was positive for Borg Warner continuous spark mode indicating that longer spark durations were characterised by higher energy delivered to the spark plug. The complete opposite was observed for all other systems, indicating that high energy cycles were more likely to shorter than others.
- SVA running condition was found to be significantly more sensitive to improvements in combustion stability when aided by higher energy ignition than DVA.
- The highest improvements in combustion stability and extension of dilution limit were achieved for Borg Warner continuous spark mode. Regardless of the fact that it

delivered the highest total energy, it is believed that the benefit should be attributed to mode of delivery (quality), rather than absolute energy values.

- Borg Warner system operated in continuous spark mode was also found to provide the fastest heat release across number of tested conditions, whilst the same system in Multispark mode was the slowest. Differences became more pronounced as the dilution increased beyond $\lambda=1.4$.

6.2. Recommendations for Future Work

Results presented and discussed in this work met objectives set for this thesis, however they also helped to identify where further work would be highly desirable. Key areas include:

- Split injection studies where fuel would be delivered at multiple points during the engine cycle. It is believed that introducing such operation could be quite effective in lowering emissions. Ideally further work on engine fuelling should be accompanied with introducing particle counter (PM is also needed for engine certification).
- Increasing geometrical compression ratio of the engine. Knowing that compression ratio is key to cycle efficiency, it is certainly worth investigating. Having capability of CVVL system, effective compression ratio could be lowered whilst retaining expansion ratio.
- Introducing EGR stratification. Numerical simulations were published, however there has been little research addressing the issue using experimental studies. Having good understanding of mixture preparation, it is believed that concept is worth investigating. It is envisaged to increase overall dilution and cycle efficiency with limited penalty associated with slow burn. Installing EGR feed to only one valve, and aim to control its distribution within the cylinder by valvetrain action.
- Spark studies on optical engine. Lack of corresponding optical work has already been identified as main drawback to spark measurements. Optical imaging would also be hugely beneficial in qualitative assessment of CFD models.

- Spark plug gap and orientation studies. It is known that spark plug orientation can affect flame development. Flow within the cylinder changed with valve deactivation, hence examining other spark plug positions could lead to further optimisation.
- Extension of staggered lift study to run throttleless. Load control could be achieved through setting EIVC on both valves, thus reducing the pumping losses. Valves should be staggered to extend the benefit of swirl motion and increased TKE discussed in detail earlier in this thesis.

References

- Aleiferis, P.G. and Van Romunde, Z.R., 2013. An analysis of spray development with iso-octane, n-pentane, gasoline, ethanol and n-butanol from a multi-hole injector under hot fuel conditions. *Fuel*, 105, pp.143-168
- Alger, T., Chauvet, T., & Dimitrova, Z., 2008. Synergies between High EGR Operation and GDI Systems. *SAE International Journal of Engines*, 1(1), 101–114.
- Alger, T., & Mangold, B., 2013. Dedicated EGR : A New Concept in High Efficiency Engines. *SAE*, 2(1), 620–631.
- Alger, T., Gingrich, J., Mangold, B., & Roberts, C., 2011. A Continuous Discharge Ignition System for EGR Limit Extension in SI Engines. *SAE International Journal of Engines*, 4(1), 677–692.
- Alger, T., Gingrich, J., Roberts, C., Mangold, B., & Sellnau, M., 2013. A High-Energy Continuous Discharge Ignition System for Dilute Engine Applications. *SAE Technical Paper* 2013-01-16, (2013-01-16).
- Alger, T., & Wooldridge, S., 2011. Measurement and Analysis of the Residual Gas Fraction in an SI Engine with Variable Cam Timing. *Library*, (724).
- Angelberger, C., Poinot, T. and Delhay, B., 1997. Improving near-wall combustion and wall heat transfer modeling in SI engine computations (No. 972881). *SAE Technical Paper*.
- AVL List GmbH, 2010, INDICATING SYSTEMS AVL OT-SENSOR 428, Product Description Brochure.
- Ayala, F. a, Gerty, M. D., & Heywood, J. B., 2006. Effects of Combustion Phasing, Relative Air-fuel Ratio, Compression Ratio, and Load on SI Engine Efficiency. *SAE Technical Paper* 2006-01-0229, (724).

Bernard, L., Ferrari, A., Micelli, D., Perotto, A., Rinolfi, R., & Vattaneo, F., 2009. Electro-hydraulic Valve Control with MultiAir Technology. *ATZautotechnology*, 9(6), 32–37.

Briggs, T., Alger, T., & Mangold, B., 2014. Advanced Ignition Systems Evaluations for High-Dilution SI Engines. *SAE International Journal of Engines*, 7(4), 1802–1807.

Bücker, I., Karhoff, D.-C., Klaas, M., & Schröder, W., 2013. Engine In-Cylinder Flow Control via Variable Intake Valve Timing. *SAE International*, (Ccv), 11.

Cairns, A., Blaxill, H., & Irlam, G., 2006. Exhaust Gas Recirculation for Improved Part and Full Load Fuel Economy in a Turbocharged Gasoline Engine. *SAE Technical Paper*, (724).

Caton, J.A., 2013. A Comparison of Lean Operation and Exhaust Gas Recirculation: Thermodynamic Reasons for the Increases of Efficiency (No. 2013-01-0266). *SAE Technical Paper*.

Cleary, D. and Silvas, G., 2007. Unthrottled engine operation with Variable Intake Valve Lift, Duration, and Timing. *SAE paper number 2007-01-1282*.

Copeland, C. D., Gao, X., Freeland, P. a., Neumeister, J., & Mitcalf, J., 2013. Simulation of Exhaust Gas Residuals in a Turbocharged, Spark Ignition Engine, 2–11.

D. Gruden (Porsche AG), R. H. (n.d.). Performance, Exhaust Emissions and Fuel Consumption of an IC Engine Operating with Lean Mixtures

Duchaussoy, Y., Lefebvre, A., Bonetto, R., & Renault, S. a., 2013. Dilution Interest on Turbocharged SI Engine Combustion Reprinted From : SI Combustion. *Sae*, (724).

European Commission., 2015. Reducing CO₂ emissions from passenger cars.

Fontana, G. and Galloni, E., 2009. Variable valve timing for fuel economy improvement in a small spark-ignition engine. *Applied Energy*, 86(1), pp.96-105.

Frierl, R. and Kluting, M., 2000. The Third Generation of Valvetrains - New Fully Variable Valvetrains for Throttle-Free Load Control. *SAE paper number 2000-01-1227*.

Grandin, B., Angstroem, H.-E., Stalhammar, P., & Olofsson, E., 1998. Knock suppression in a turbocharged SI engine by using cooled EGR. SAE Technical Papers, (724).

Grandin, B., & Angstroem, H.-E., 1999. Replacing Fuel Enrichment in a Turbo Charged SI Engine: Lean Burn or Cooled EGR. SAE Technical Papers, (724).

Hacohen, J., Ashcroft, S. J., & Belmont, M. R., 2015. SAE TECHNICAL PAPER SERIES Lean Burn Versus EGR SI Engine, (412).

Hardalupas, Y., Taylor, A.M.K.P., Whitelaw, J.H., Ishii, K., Miyano, H. and Urata, U., 1995. Influence of Injection Timing on In-Cylinder Fuel Distribution in a Honda Vtec-E Engine. SAE paper number 950507.

Gülder, Ö.L., 1991, January. Turbulent premixed flame propagation models for different combustion regimes. In Symposium (International) on Combustion (Vol. 23, No. 1, pp. 743-750). Elsevier.

Heywood, J.B., 1988. Internal combustion engine fundamentals (Vol. 930). New York: McGraw-hill.

Horie, K., Nishizawa, K., Ogawa, T., Akazaki, S. and Miura, K., 1992. The development of a high fuel economy and high performance four-valve lean burn engine (No. 920455). SAE Technical Paper.

Ivanic, Z., Ayala, F., Goldwitz, J., & Heywood, J. B., 2005. Effects of Hydrogen Enhancement on Efficiency and NO_x Emissions of Lean and EGR-Diluted Mixtures in a SI Engine Reprinted From : SI Combustion and Direct Injection SI Engine Technology. SAE International, (724).

Jovanovic, Z.S., Basara, B.S., Tomić, M.V. and Petrović, V.S., 2011. Some subtleties concerning fluid flow and turbulence modeling in 4.-valve engines. Thermal science, 15(4), pp.1065-1079.

Kim, J.N., Kim, H.Y., Yoon, S.S. and Sa, S.D., 2008. Effect of intake valve swirl on fuel-gas mixing and subsequent combustion in a CAI engine. International journal of automotive technology, 9(6), pp.649-657.

Koga N., 2001. An Experimental Study on Fuel Behaviour during the Cold Start Period of a Direct Injection Spark Injection Engine. SAE paper number 2001-01-0969.

Lee, J.H., Goto, S., Tsurushima, T., Miyamoto, T. and Wakisaka, T., 2000. Effects of injection conditions on mixture formation process in a premixed compression ignition engine (No. 2000-01-1831). SAE Technical Paper.

Li, T., Gao, Y., Wang, J., & Chen, Z., 2014. The Miller cycle effects on improvement of fuel economy in a highly boosted, high compression ratio, direct-injection gasoline engine: EIVC vs. LIVC. *Energy Conversion and Management*, 79, 59–65.

Lumley, J.L., 1999. *Engines: an introduction*. Cambridge University Press.

MAHLE Powertrain Ltd., 2019, MAHLE Jet Ignition® (MJI) Product Information.

Mahrous, A.F.M., Wyszynski, M.L., Xu, H., Tsolakis, A. and Qiao, J., 2007. A CFD Investigation Into the Effects of Intake Valves Events on Airflow Characteristics in a Motored 4-Valve Engine Cylinder With Negative Valve Overlapping (No. 2007-24-0032). SAE Technical Paper.

Mendl, G., Heiduk, T., Knirsch, S., & Ag, A., 2015. The new Audi 2.0l Engine with innovative Rightsizing - a further Milestone in the TFSI Technology, *Kurzfassung Einführung*, 60–77.

Moore, W., Foster, M., Lai, M.C., Xie, X.B., Zheng, Y. and Matsumoto, A., 2011. Charge motion benefits of valve deactivation to reduce fuel consumption and emissions in a gdi, vva engine (No. 2011-01-1221). SAE Technical Paper.

Murase, E, Ono, S., 1989. Plasma Jet Ignition in Turbulent lean Mixtures. SAE Technical Paper.

Nakata, K., Nogawa, S., Takahashi, D., Yoshihara, Y., Kumagai, A. and Suzuki, T., 2016. Engine technologies for achieving 45% thermal efficiency of SI engine. *SAE International Journal of Engines*, 9(1), pp.179-192.

Nakayama, Y., Maruya, T., Oikawa, T., Fujiwara, M. and Kawamata, M., 1994. Reduction of Emission from VTEC Engine During Cold-Start Condition (No. 940481). SAE Technical Paper.

Okahara, A., 2017. DENSO TEN Technical Review Vol.1. AE Engineering Group Advanced System Development 2017

Ozdor, N., Dulger, M., & Sher, E., 1994. Cyclic Variability in Spark Ignition Engines A Literature Survey. SAE Technical Paper, (940987).

Pan, M., Shu, G., Wei, H., Zhu, T., Liang, Y., & Liu, C., 2014. Effects of EGR, compression ratio and boost pressure on cyclic variation of PFI gasoline engine at WOT operation. Applied Thermal Engineering, 64(1-2), 491–498.

Patel, R., Ladommatos N., Stansfield P., Wigley G., Garner P., Pitcher G., Turner J., 2008. Comparison between Unthrottled, Single and Two-valve Induction Strategies Utilising Direct Gasoline Injection: Emissions, Heat-release and Fuel Consumption Analysis. SAE paper number 2008-01-1626.

Pischinger, S., Geiger, J., Neff, W., Böwing, R., Thiemann, J., 2003. Einfluss von Zündung und Zylinderinnenströmung auf die ottomotorische Verbrennung bei hoher Ladungsverdünnung. MTZ Motortechnische Zeitschrift 63(5), 388–399

Pischinger, M., Salber, W., van der Staay, F., Baumgarten, H., Kemper, H., 2000. Benefits of the Electromechanical Valve Train in Vehicle Operation. SAE paper number 2000-01-1223.

Salber, W., Wolters, P., Esch, T., Geiger, J. and Dilthey, J., 2002. Synergies of Variable Valve Actuation and Direct Injection. SAE paper number 2002-01-07006.

Scarcelli, R., Matthias, N., & Wallner, T., 2013. Numerical Investigation of Combustion in a Lean Burn Gasoline Engine. SAE International, (X), 18.

Schneider, A., Hettinger, A., Schünemann, E., Rottengruber, H., 2016. Optische Untersuchungen zur Flammenkernbildung im Ottomotor bei Ladungsverdünnung. 12 Internationales Symposium für Verbrennungs-diagnostik, Baden Baden

Schwarz, F., & Spicher, U., 2003. Determination of Residual Gas Fraction in IC Engines, (724).

Shayler, P. J., Winborn, L. D., & Hill, M. J., 2000. SAE TECHNICAL The Influence of Gas / Fuel Ratio on Combustion Stability and Misfire Limits of Spark Ignition Engines, (724).

Smith, P., & Cheng, W. K., 2013. Assessing the Loss Mechanisms Associated with Engine Downsizing, Boosting and Compression Ratio Change. Sae.

Su, J., Xu, M., Li, T., Gao, Y., & Wang, J., 2014. Combined effects of cooled EGR and a higher geometric compression ratio on thermal efficiency improvement of a downsized boosted spark-ignition direct-injection engine. *Energy Conversion and Management*, 78, 65–73.

Subramaniam, M. N., Kleeberg, H., Bhattacharyya, A., Chomic, N., & Tomazic, D., 2010. A Multi-Cylinder Airflow & Residual Gas Estimation Tool Applied to a Vehicle Demonstrator. Sae, 1–21.

Stokes, J., Lake, T. H., & Osborne, R. J., 2000. A Gasoline Engine Concept for Improved Fuel Economy - The Lean Boost System. SAE International, (724).

Takahashi, D., Nakata, K., Yoshihara, Y., & Omura, T., 2016. Combustion Development to Realize High Thermal Efficiency Engines. *SAE International Journal of Engines*, 9(3), 2016–01–0693.

Tang, Q., Liu, J., Zhan, Z., & Hu, T., 2013. Influences on Combustion Characteristics and Performances of EGR vs. Lean Burn in a Gasoline Engine. SAE International.

Toulson, E., Watson, H. C., & Attard, W. P., 2007. Sae technical paper series 2007-24- 0120.

Toulson, E., Watson, H. C., & Attard, W. P., 2007. The Effects of Hot and Cool EGR with Hydrogen Assisted Jet Ignition. Sae, (724).

Trask, N.R., Hammoud, M., Haghoorie, M., Megli, T.W. and Dai, W., 2003. Optimization Techniques and Results for the Operating Modes of a Camless Engine. SAE paper number 2003-01-0033.

Wang, Y., Lin, L., Zeng, S., Huang, J., Roskilly, A. P., He, Y., Li, S., 2008. Application of the Miller cycle to reduce NO_x emissions from petrol engines. *Applied Energy*, 85(6), 463–474.

Wei, H., Zhu, T., Shu, G., Tan, L., & Wang, Y., 2012. Gasoline engine exhaust gas recirculation - A review. *Applied Energy*, 99(X), 534–544.

Xu, M., Chen, G., Daniels, C. and Dong, M., 2000. Numerical study on swirl-type high-dilution stratified EGR combustion system (No. 2000-01-1949). SAE Technical Paper.

Yamakawa, M., Youso, T., Fujikawa, T., Nishimoto, T., Wada, Y., Sato, K. and Yokohata, H. 2012. Combustion technology development for a high compression ratio SI engine. *SAE International Journal of Fuels and Lubricants*, 5(1), pp.98-105.

Zeldovich, Y.A., Frank-Kamenetskii, D. and Sadovnikov, P., 1947. Oxidation of nitrogen in combustion. Publishing House of the Acad of Sciences of USSR.

Appendices

Chapter 2

Fuel Certificate of Analysis:



Haltermann Carless UK LTD. Cedar Court Guildford Rd, Leatherhead, Surrey,
KT22 9RX United Kingdom

Mechanical Engineering Stores
Room 124, City & Guilds Building
Imperial College London
London
SW7 2BX

Certificate 100000100187

Date: 20/02/2017
Customer PO: ME/3622783
Delivery Note: 80082258 000010
Order No.: 66083571 000010
Customer No.: 1764609

GMID: 2010872
Material: Carcal EURO 6 GASOLINE E10
146 kg Steel Drum
Cust. Mat.:
Batch: 0000012082
Dlvy. Qty: 146.0 KG

Revision No.: 2

Container ID: COATES
Ship from: Harwich Pack

Harwich, ES, United Kingdom

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Feature	Units	Results	Limits		Method
			Minimum	Maximum	
Density 15° C	kg/m3	749.6	743.0	756.0	EN ISO 12185
Clear & Bright at -7° C	-	Pass	-	-	ASTM D4176-1
I.B.Pt.	° C	39.1			EN ISO 3405
% Evaporated at 70° C, E70	%(V)	40.4	34.0	46.0	EN ISO 3405
% Evaporated at 100° C, E100	%(V)	56.8	54.0	62.0	EN ISO 3405
% Evaporated at 150° C, E150	%(V)	87.1	86.0	94.0	EN ISO 3405
F.B.Pt.	° C	178.6	170.0	195.0	EN ISO 3405
Residue	%(V)	1.1	-	2.0	EN ISO 3405
R.O.N (corrected)		96.5	95.0	98.0	EN ISO 5164
M.O.N. (corrected)		86.5	85.0	89.0	EN ISO 5163
Olefin content	%(V)	7.6	6.0	13.0	EN ISO 22854
Aromatic content	%(V)	30.1	25.0	32.0	EN ISO 22854
Benzene content	%(V)	0.2	-	1.0	EN 238
Saturate content	%(V)	52.9			EN ISO 22854
Vapour pressure (DVPE) 37.8° C	kPa	59.1	56.0	60.0	EN 13016-1
Water Content	%(V)	0.036	-	0.050	EN ISO 12937
Oxidation Stability (Induction Period)	min.	> 480	480	-	EN ISO 7536
Oxygen Content	%(m)	3.46	3.30	3.70	EN ISO 22854
Gum - washed	mg	2.0	-	4.0	EN ISO 6246
		per 100 cm3			
Sulphur content	mg/kg	< 3.0	-	10.0	EN ISO 20846
Copper Corrosion, 3hrs at 50° C	-	1A	-	-	EN ISO 2160
Lead content	mg/l	< 2.5	-	5.0	EN 237
Phosphorous content	mg/l	< 0.2	-	1.3	ASTM D3231
Ethanol	%(V)	9.4	9.0	10.0	EN ISO 22854
Carbon Content	%(m)	83.35			ASTM D5291
Hydrogen Content	%(m)	13.19			ASTM D5291
C/H Mass Ratio		6.32			CALCULATION
C/O Mass Ratio		24.09			CALCULATION
Atomic H/C Ratio		1.8860			CALCULATION

Mechanical Engineering Stores
Imperial College London
London SW7 2BX

Delivery item/date
80082258 000010 / 21/02/2017

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Feature	Units	Results	Limits		Method
			Minimum	Maximum	
Gross Heat of Combustion	MJ/kg	44.34			IP 12
Net Heat of Combustion	MJ/kg	41.54			IP 12
Density at 20° C	kg/m3	745.5			ASTM D4052

COA Additional Information

The certificate is electronically generated and valid without signature.

Haltermann Carless UK Ltd.

For inquiries please contact Customer Service or local Sales

Additional Info:

Atomic H/O ratio: **60.51**
Stoichiometric AFR: **13.97**

Sample Error Analysis for GSFC

Measurements presented in this thesis are subject to cumulative errors and their respective propagation through calculation chains. This section provides an example as to how uncertainty is being developed and dealt with to minimise its impact on the overall results and drive conclusions while chasing relatively small enhancements in engine operating efficiency.

1500 RPM; 3.6 bar GMEP test point which had GSFC of 241.7 g/kWhr is analysed as a reference.

Given that 0.1°CA error in TDC determination results in approximately 1% error in GMEP, by using capacitive probe for TDC determination, it is assumed that 0.5% error (corresponding to 0.5°CA inaccuracy in TDC setting) is present, equal to 0.018 bar. Maximum error of 9.5 mbar caused by the in-cylinder piezoelectric pressure sensor, taking into account resolution of 14 Bit system is estimated at 0.010 bar. Total error coming from both pressure sensor inaccuracy and volume determination is estimated to be 0.028 bar.

$$GSFC = \frac{m_{fuel}[\frac{kg}{h}]}{Gross\ Indicated\ Power\ [kW]} \quad (A.1)$$

$$Gross\ Indicated\ Power\ [kW] = GMEP * 1000 * \frac{Displacement[cc]}{1000} * \frac{Speed\ [RPM]}{60*2} \quad (A.2)$$

$$Uncertainty\ \delta R = |R| \sqrt{\left(\frac{\delta X}{X}\right)^2 + \left(\frac{\delta Y}{Y}\right)^2 + \left(\frac{\delta Z}{Z}\right)^2}, \text{ where} \quad (A.3)$$

X, Y, Z nominal recorded values

$\delta X, \delta Y, \delta Z$ respective tolerance in measurement

As stated previously, accuracy of Coriolis measurement of fuel flow was given as 0.1% of full scale 25 kg/h (0.025 kg/h), whilst speed measurement error is assumed to be 0.1%.

$$\delta R = 241.73\ g/Kwhr \sqrt{\left(\frac{0.025\ kg/hr}{0.545\ kg/hr}\right)^2 + \left(\frac{0.028\ bar}{3.614}\right)^2 + \left(\frac{1.5\ RPM}{1500\ RPM}\right)^2}$$

Therefore, the accuracy of GSFC for this specific test condition can be expressed as 241.7 g/kWhr $\pm$ 5.6 g/kWhr, which is approximately 2% error.

It does appear to be quite significant given that small enhancements are being chased, however to further minimise impact of inaccuracies and ensure repeatability of measurements, all recorded test points were sampled for either 300 cycles (indicating equipment) or 40 seconds (standard test bed channels), ensuring that much of the uncertainty and cycle-to-cycle variations are being removed. It is being illustrated by cumulative average GMEP and CoV of GMEP in Figures A.1 and A.2.

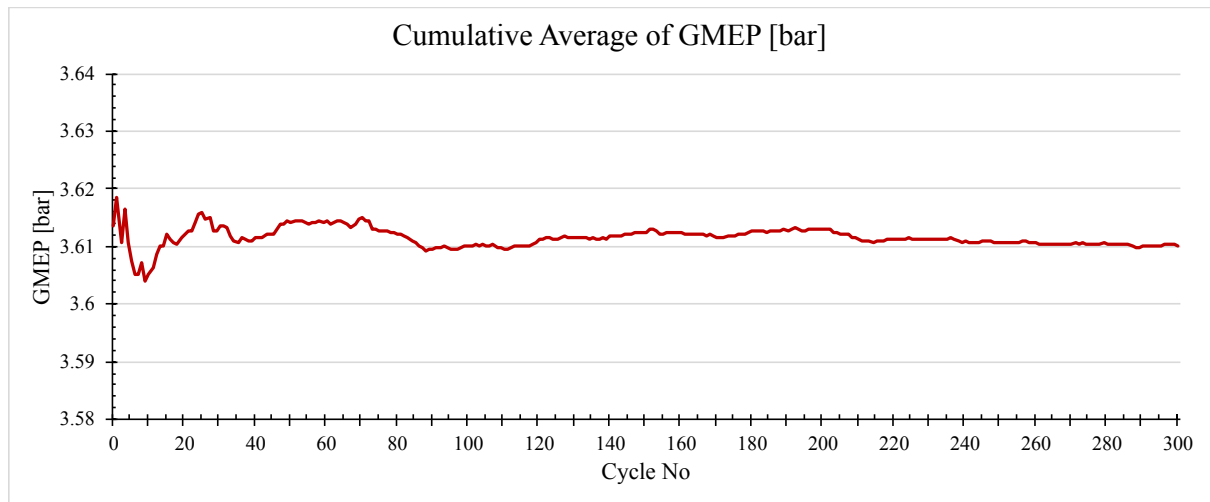


Figure A.1 – Cumulative average GMEP for 300 cycles; illustrated for 1500 RPM, 3.6 bar GMEP test condition

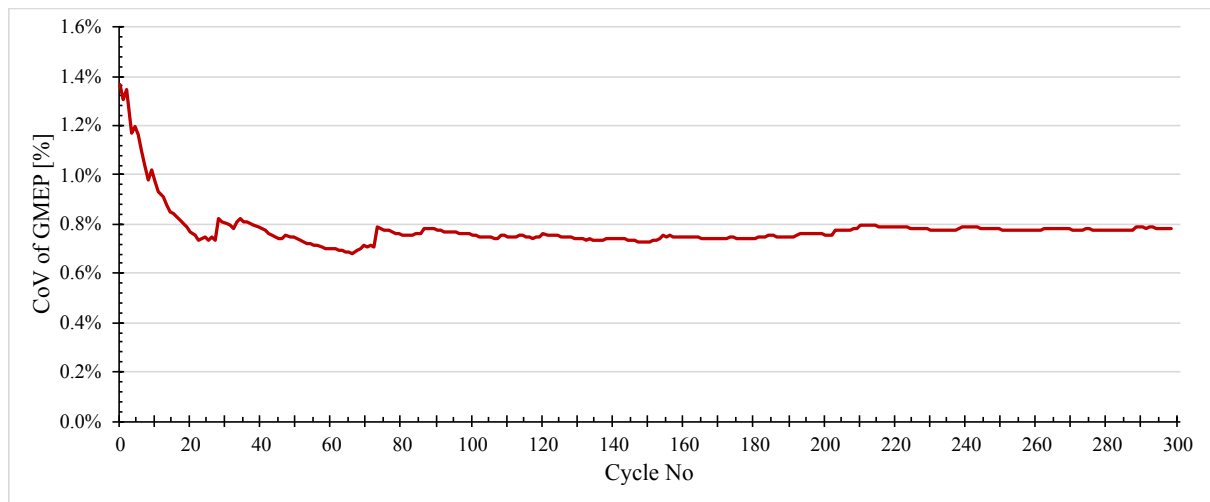


Figure A.2 – CoV of GMEP progression during accumulation of 300 consecutive cycles at 1500 RPM; 3.6 bar GMEP test condition

Matching to Multi-Cylinder Engine:

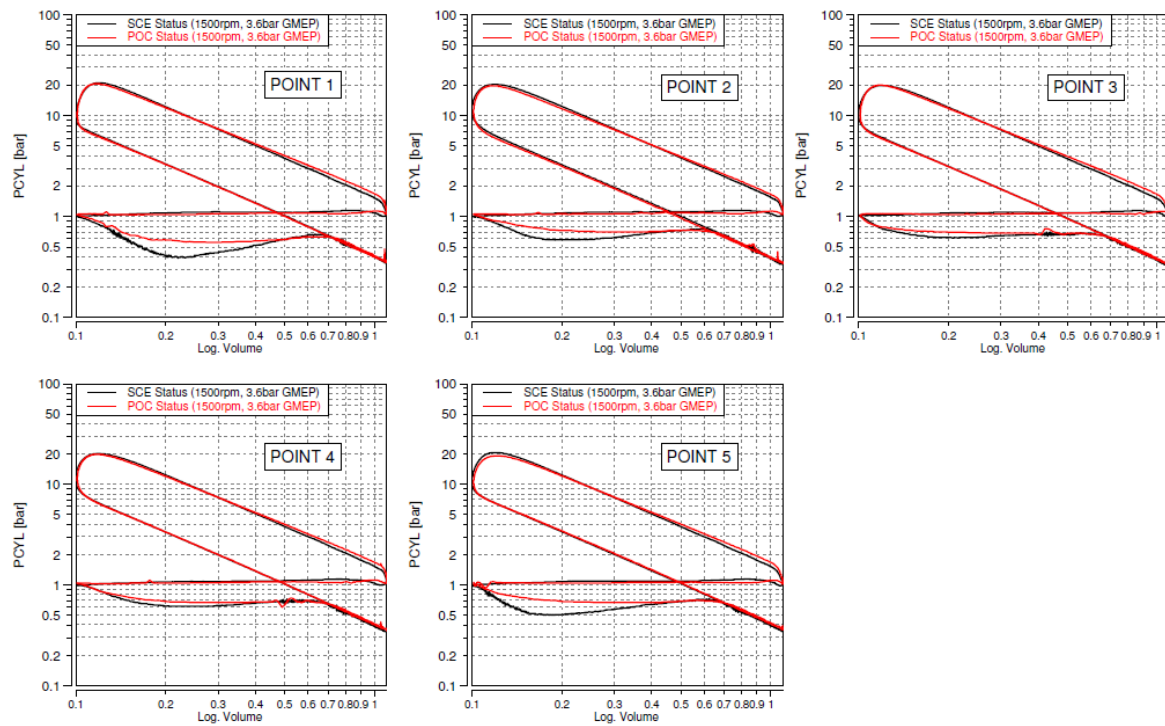


Figure A.3 – LogP logV traces for various test points used to match SCRE with multi-cylinder JLR engine

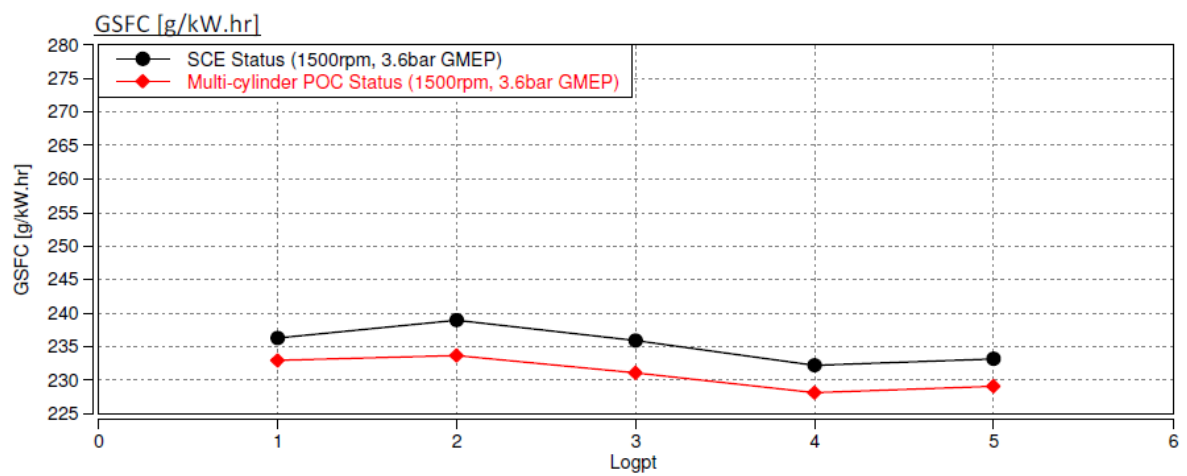
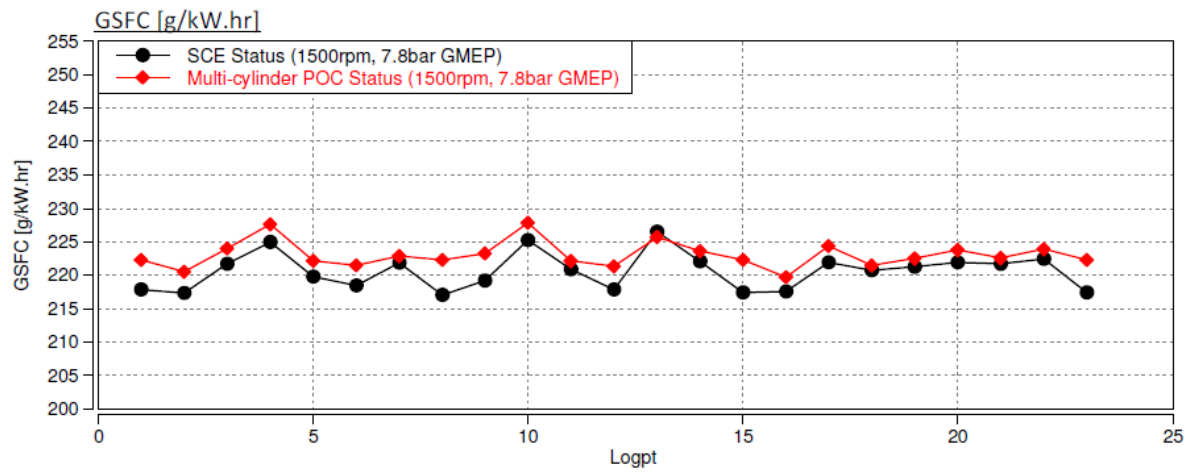


Figure A.4 – Validation exercise for chosen matching methods

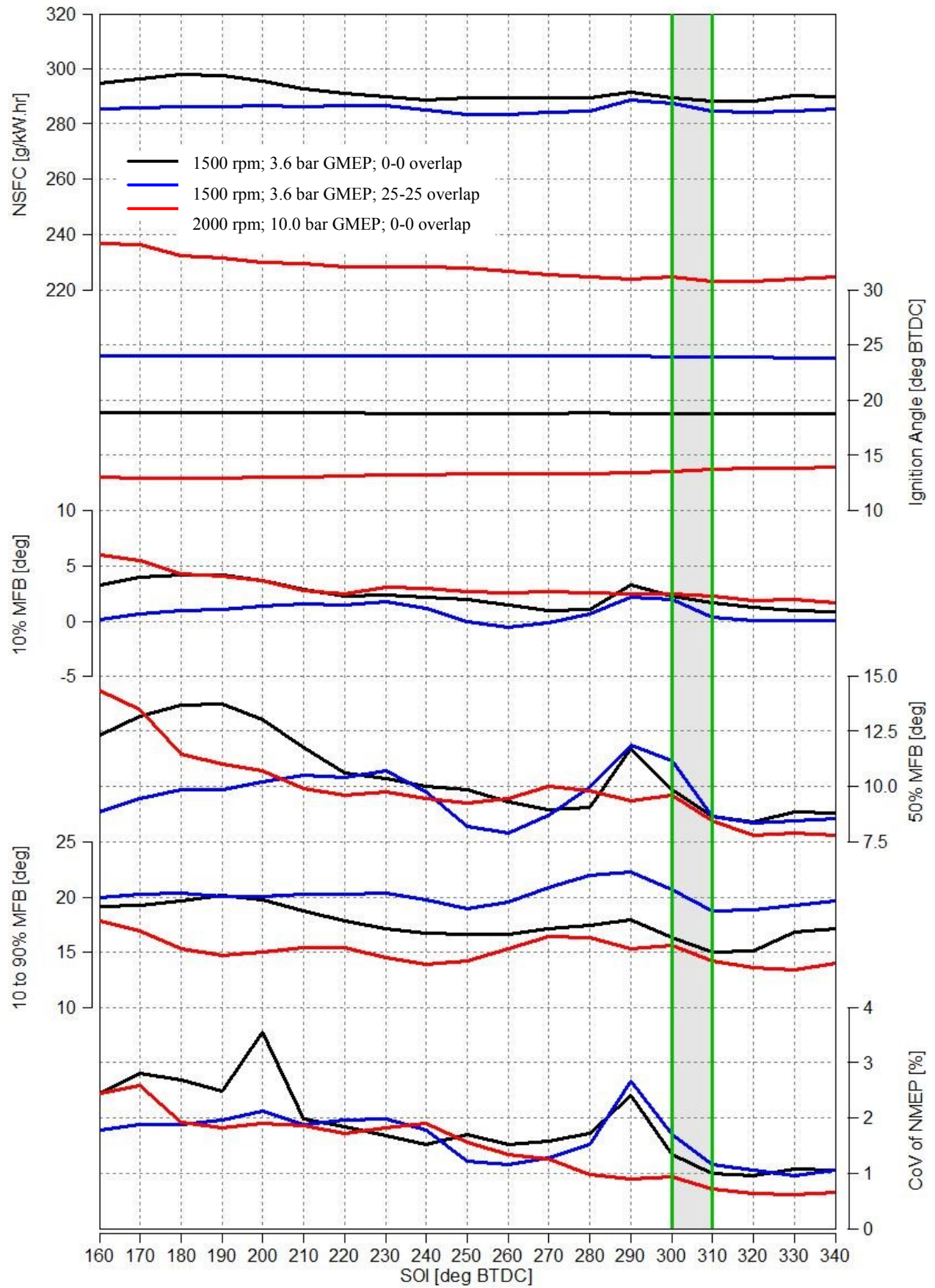


Figure A.5 – Combustion related metrics measured during SOI sweeps

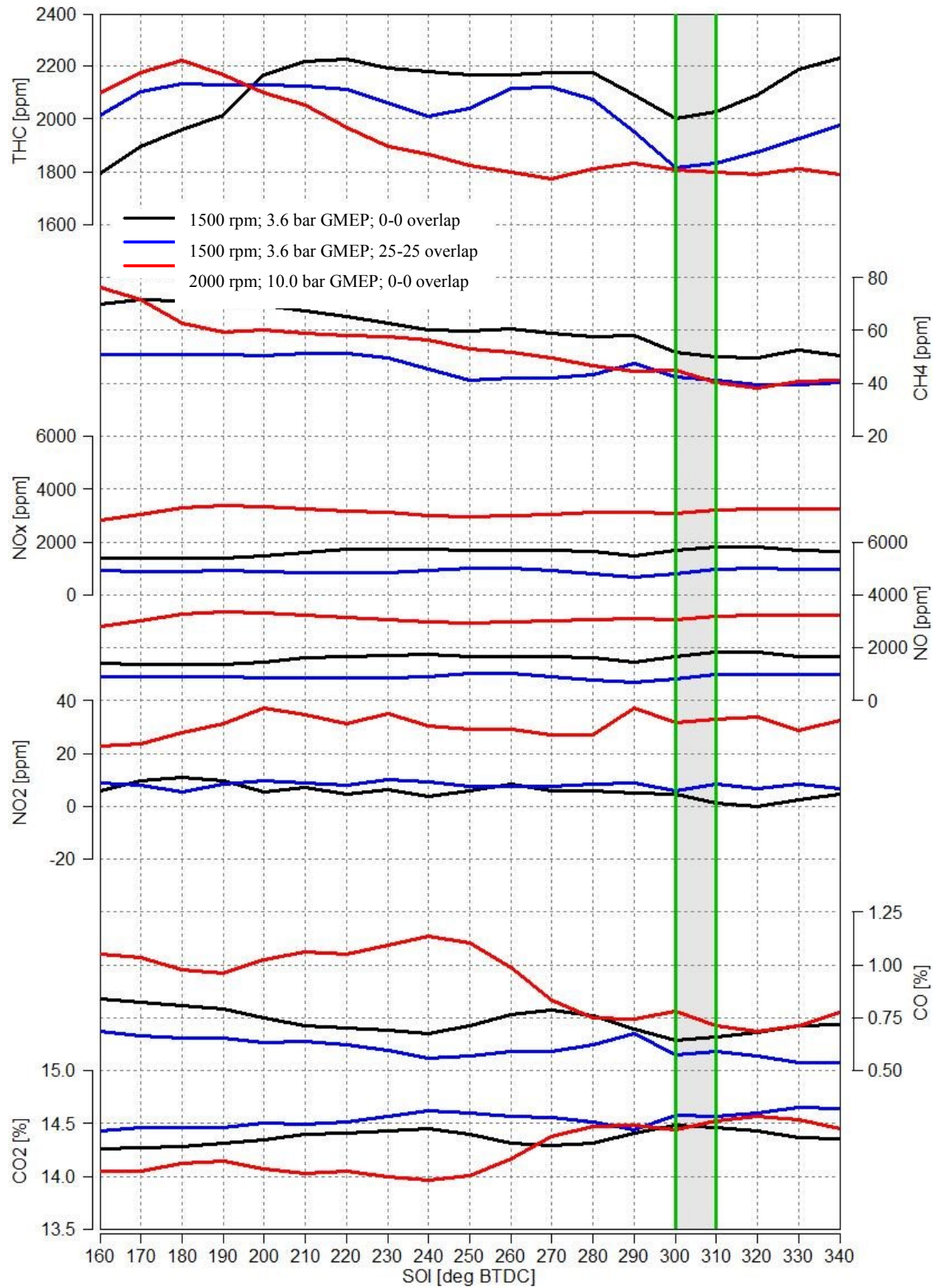


Figure A.6 - Emissions metrics measured during SOI sweeps

Daily Check – Work Instruction Sheet (WIS)

IMPERIAL COLLEGE LONDON ENGINE TEST FACILITY - WORK INSTRUCTION SHEET AND TEST REQUIREMENT KEY FACTS									
SWIFT JC:	N/A	ENGINE NUMBER:	SCE AJ21-PX P6 Gen 2 CVVL	TEST CELL:	1				
PROJECT:	SCE AJ21-PX P6 Gen 2 CVVL	ENGINEER:		ENGINEER CONTACT:					
TEST TITLE:	Engine Daily Checks			WIS NUMBER:	00xx				
TEST OBJECTIVE:	Daily check of engine performance			TESTER(S):	TBC				
EXPECTED START DATE:	xx/xx/xx	EXPECTED END DATE:	xx/xx/xx	NUMBER OF TEST POINTS:	2				
MEASURING EQUIPMENT				OTHER RELEVANT MEASUREMENT INSTRUCTIONS:					
TEST BED LOG ☒ INDICOM ☐ ASAP ☒ MEXA ONE ☐ FTIR ☐ DMS500 ☐ INCA LOG ☐ CAMEO ☐ SMOKE METER ☐				Fluid temperatures must be within +/-1°C before measurement is taken. Naturally aspirated Stability limit: NMEP < 3.3bar, NMEPTs < 0.1bar Stability limit: NMEP > 3.3bar, NMEPTv < 3%					
FUEL:	95RON ES	FUEL PROPERTIES:	HCR/OCR	See test log	ratio	ENSURE Dataq Pro is loaded with correct fuel properties			
			LHV:	See test log	MI/kg				
ENGINE SPECIFIC SETUP		EMS SPECIFIC SETUP			DYN0 CONTROL				
ENGINE SPEED [rev/min]	See table	IGNITION: ["ATDC"/"AIS0]	See table		MODE:	Speed/Load			
ENGINE LOAD [NMEP/bar]	See table	KNOCK CONTROL:	Tester		Speed/direct	See test detail			
PLENUM AIR TEMP [°C]	25	FUELLING: [MODE/STATE]	Lambda 1		EXHAUST BACKPRESSURE SETUP				
FUEL TEMP [°C]	30	SOI1: ["BTDCfiring]	See table		Open				
COOLANT TEMP [°C]	90	EOI2: ["BTDC]	-		NOTES				
OIL TEMP [°C]	90	SPLIT RATE [-]	-		Build level: P6 Gen 2 Valvetrain: CVVL Cylinder head: Gen 0 (Phase 0) Piston: Gen 0 (Phase 0) Spark plug: NGK HR#8 (T20336K-A) Fuel injector: EA1				
PCJ ON/OFF [-]	OFF	FUEL PRESSURE: [Bar]	See table						
OIL LEVEL:	Max	IVO: ["CA]	See table						
OIL GRADE:	0W20	EVC: ["CA]	See table						
EXHAUST FANS:	On								
CALIBRATION SOFTWARE LEVEL:									
CALIBRATION NAME:									
CHECKLIST									
ITEM	DETAIL			COMPLETE	TESTER INITIALS				
1	Ensure oil and coolant are approximately 30°C before starting engine.								
2	Make sure correct fuel properties are entered into Dataq Pro.								
3	Check Indiset, ASAP and Mexa One are all working correctly.								
4	When starting engine ensure dyno is in absorb and motor.								
5	Run engine to 1000rev/min and turn on fuel and ignition.								
6	File name: DATEBACKWARDS_SCEGEN2CVVL_DAILYCHECK_0XXX								

IMPERIAL COLLEGE LONDON ENGINE TEST FACILITY - WORK INSTRUCTION SHEET AND TEST REQUIREMENT KEY FACTS									
SWIFT JC:	N/A	ENGINE NUMBER:	SCE AJ21-PX P6 Gen 2 CVVL	TEST CELL:	1				
PROJECT:	SCE AJ21-PX P6 Gen 2 CVVL	ENGINEER:		ENGINEER CONTACT:					
TEST TITLE:	Engine Daily Checks			WIS NUMBER:	00xx				
TEST OBJECTIVE:	Daily check of engine performance			TESTER(S):	TBC				
TEST PROCEDURE AND INSTRUCTIONS									
For this test please run the engine in EIVC mode. To achieve this, you will need to set the INA lift mode to 2. Reduce the value of the closing angle, watching the valve lift profile in Indicom, to bring the engine load down. At the same time, open the throttle to 100%.									
Stage Number [#]	Engine Speed [rev/min]	Target Load NMEP [bar]	Int Cam Advance [°CA]	Exh Cam Retard [°CA]	mb50 [°CA aTDC]	Fuel Pressure [bar]	SOI1 [°BTDC]	Stage Time [mins]	Log Number
1	2050	2.6	0	25	8	110	330	5	
2	2500	10.0	10	30	8	150	300	5	
NOTES									
* 60 second log to be taken at the end of each stage * Ensure measured Indicom data has been passed over to Dataq Pro before log is taken at each stage * Ensure ifile has been generated before moving onto next stage									
TEST FILE ID:				END DATE:					

Chapter 3

Emissions measurements

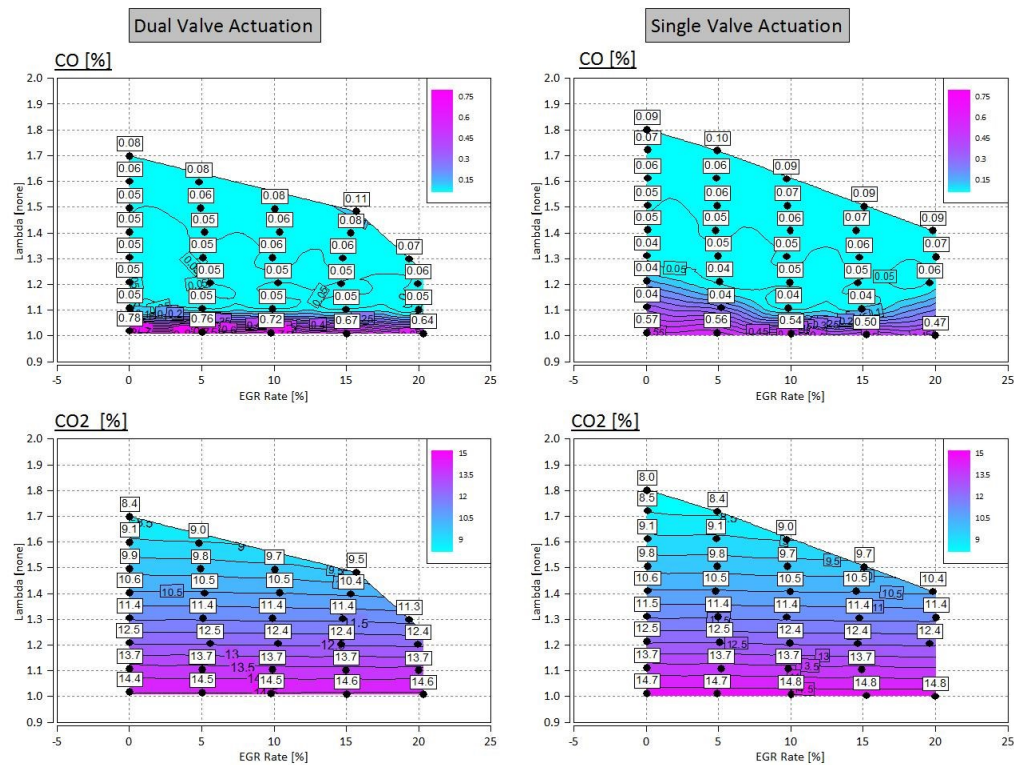


Figure A.7 - CO and CO₂ emissions comparison for DVA and SVA at 1500 RPM, 7.9 bar GMEP

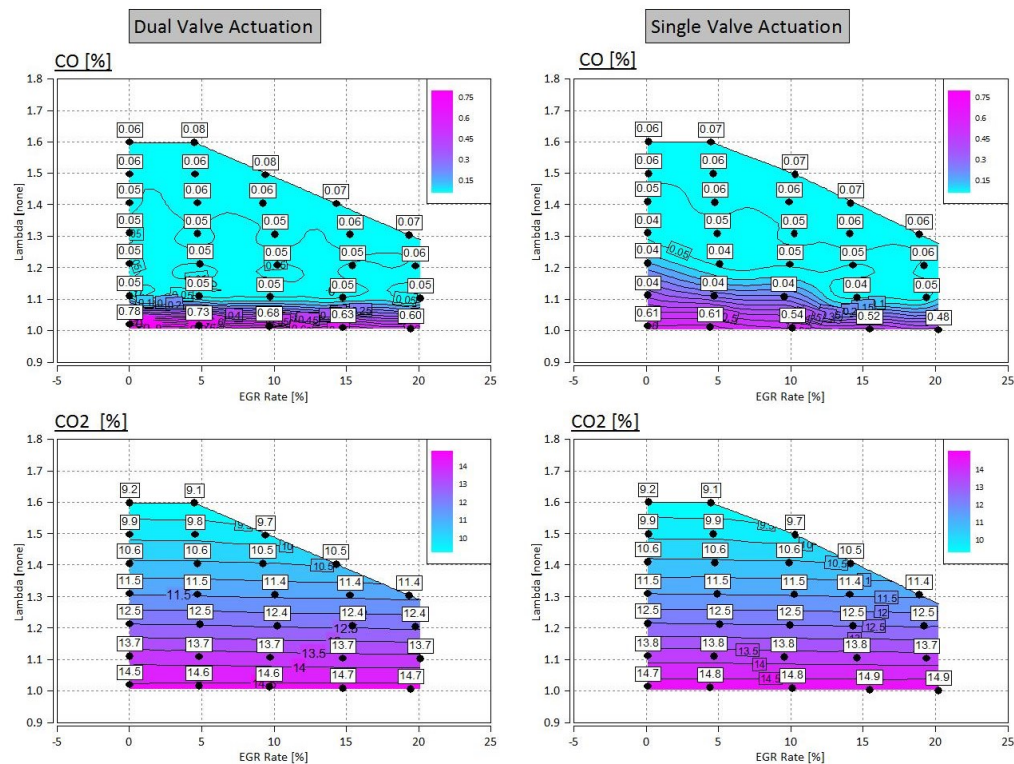


Figure A.8 - CO and CO₂ emissions comparison for DVA and SVA at 2000 RPM, 9.8 bar GMEP