Effect of Different Fluids on Injection Strategies to Suppress Pre-Ignition

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Abstract:

Pre-ignition is an abnormal engine combustion phenomenon where the inducted fuel-air charge ignites before the spark ignition. This premature combustion phenomenon often leads to heavy knocking events. The mixture preparation plays a critical role in pre-ignition tendency for a given load. Literature shows efforts made towards improving pre-ignition-limited-IMEP by splitting the injection pulse into multiple pulses. In this study, two direct injectors are used in a single cylinder research engine. A centrally mounted direct injector was used to inject Coryton Gasoline (RON 95) fuel early in the intake stroke. A second fluid was injected late in the compression stroke to suppress pre-ignition. The fluids used in the second direct injector was varied to see the effects of the molecule and its physical and chemical property on pre-ignition suppression tendency. Methanol, ethanol, water, and gasoline were tested as second fluid. Engine tests were conducted at 2000 rpm and at an intake pressure of 2.1 bar (abs). Although alcohols show high pre-ignition tendency as fuels, they were most effective at pre-ignition suppression when injected later in the compression stroke. The pre-ignition suppression led to a decrease in IMEP and an increase in cycle-to-cycle variation. Water injection was highly effective at maintaining peak IMEP values. Water injection was further explored for pre-ignition suppression. The water injection helped reduce pre-ignition count when injected at two different injection times each in intake, compression and late exhaust stroke.

Introduction:

Pre-ignition is a bottleneck to further downsizing, and downspeeding of modern spark ignited engines [1-3]. Downsizing is a widely accepted strategy to reduce the overall carbon footprint of vehicles and provide improved engine efficiency [4-8]. Turbocharging is employed in downsized engines to provide power rivalling larger displacement naturally aspirated engines, while reducing throttling losses at part-load operation. Researchers report upto 25% efficiency improvements from downsized and turbocharged engines in NEDC (New European Driving Cycle) test cycle, where a major fraction of the engine is run at throttled conditions [8, 9]. With the recent introduction of the real-world driving cycle (RDE cycles), the engine is operated at higher load points more often [10]. The operation in the RDE cycle would benefit as much from a reduction in enrichment requirement at high loads, as from parasitic losses. Engines are more knock limited at boosted intake pressures, which necessitates a lower compression ratio design for downsized engines. These factors often lead to lower efficiencies at peak load, compared to a naturally aspirated engine at same load. Not opting for extreme-downsizing is often referred to as ‘right’ sizing. ‘Right’ sizing leads to lower peak load, avoiding knock and hence, fuel enrichment. These trends may point towards a pushback from extreme downsizing towards ‘right’ sizing of engines.

Pre-ignition is not a recent problem that has ushered with downsizing alone and its causatives remain elusive. As early as 1900s, pre-ignition had its source in hot engine parts such as exhaust valves and spark plugs [11, 12]. Spark plug made of porcelain insulators were deemed inadequate at cooling, as the electrode tip was found to be the hotspot for pre-ignition [13]. Mica replaced porcelain as a substitute. The specific power of engines increased by 25% (right around world war II), making mica inadequate at cooling. Moreover, mica was scarce for mass producing spark plug in those demanding times. Ceramic replaced mica and has been in use since. Liquid Sodium cooling was also introduced for exhaust valve cooling [14, 15]. Later, deposit induced pre-ignition became an issue. Lubricants with calcium additives increased deposit formation and were prohibited in high-power density applications (like aircraft engines) [16, 17]. These efforts combined with better engine design led to pre-ignition being a non-issue in the later half of the 20th century [18].

Now in the 21st century, there is a push towards more efficient engines, stemming from increasingly stringent legislations to reduce global CO_2 emissions. A few trends are prominent: (1) more than 50% vehicles in the U.S. have turbocharged engines (2) Vehicles with 6 or more gears (including CVTs) account for around 50% of the total vehicles. (3) Gasoline Direct Injection vehicles make up 60% of the total vehicles [19, 20]. Together, (1) and (2) adoption trends have led to a larger fraction of engine operating at low speed and high load area. Adoption trends (1) and (3) have steered to a large fuel fraction impinging the cylinder liner, while still in liquid phase. Pre-ignition is usually (but not always) observed at low speed, high load condition in turbocharged engines [21].

Researchers attribute the modern form of pre-ignition to liquid fuel diluting the oil film on the liner. The combustion of the mixture droplets triggers the ignition of the bulk charge. These pre-ignition events are highly stochastic and phenomenologically different from hot-spot induced pre-ignition observed at high engine speeds or in aircraft engines in yesteryears [22]. The current work focuses on stochastic pre-ignition phenomenon and strategies to suppress stochastic pre-ignition.

Due to the stochastic nature of pre-ignition, it has been difficult to pinpoint one source of pre-ignition that explains all the experimental observations in literature. For example, spray-wall interaction is hypothesized to generate precursors for pre-ignition. However, similar observations were also made in gas engines having no liquid spray impinging the liner [23]. Fuel injection strategies remain critical to pre-ignition tendency of an engine [24-29]. Current literature attributes the effectiveness of fuel injection strategies to lower liquid fuel impingement [30, 31]. Split injection often leads to a lower in-cylinder temperature near the top dead center (TDC) when pre-ignition events are extremely probable. We reported a good correlation between the charge cooling tendency of the injection strategy with its effectiveness at pre-ignition suppression [30].

Although benefits of using split injection is emphasized, there is no consensus on why the strategy is effective [32]. Common explanations include the charge cooling effect and fuel-in-air heterogeneity, and increased turbulence near spark plug [33]. In this regard, injecting different fluids during late injection could help identify properties that are critical to pre-ignition suppression. Present study used four different fluids, gasoline, ethanol, methanol and water in the late split injection pulse. These fluids vary in stoichiometric air requirements, enthalpy of vaporization, density and enthalpy of combustion. Thus, trends observed could be related to their respective physical and/or chemical properties.

Addition of fluids separately has been experimented before. 'Octane-on-Demand' concept uses methanol or ethanol addition to low-octane gasoline. By operating the engine on low-octane gasoline at low loads and adding methanol/ethanol when the engine is knock limited, the concept achieves lower well-to-wheel greenhouse emissions [34-37]. Water injection has recently gained much interest [38-44]. Water+Methanol mixture are being employed commercially in cars to extend the knock limit of spark-ignited engines [45]. There is extensive literature on the effect of water addition on knock suppression, but little to no research is available on the effect of water addition on pre-ignition.

The current work had two broad aims. First, to study the late split injection strategy with different fluids to gauge the pre-ignition suppression tendency and its associated compromises. Gasoline, alcohols, and water were used. Water usage showed substantial benefits. Hence, in the second aim water injection was explored further for pre-ignition suppression. Water was injected at two different injection timings each, in the intake, compression, and late exhaust stroke. Previous studies that explored methanol, ethanol or water addition were limited to knock suppression. Therefore, the current study is focused on the effects of these additives on pre-ignition suppression.

Methodology:

There are two parts to the current study. In the first part, different fluids were varied in the late split injection strategy and the effect on (a) pre-ignition count, and (b) compromises associated with reducing pre-ignition (like reduced IMEP and CoV) were observed. In the second part, water injection was explored for testing its potential to suppress pre-ignition.

The AVL single-cylinder research engine was used. This engine is equipped with two direct injectors, placed centrally and laterally on a pent-roof cylinder head, connected to two different fluid lines; thus allowing different fluids to be supplied simultaneously in the cycle. The details of the engine are given in Table 1 and the operating conditions are deleniated in Table 2. The same methodology has been continued from previous works in [30, 46, 47]. The intake pressure is kept constant at 2.1 bar (abs), a reasonably high value that increases pre-ignition frequency and provides a repeatable data. Initial experiments were done to showcase the benefits of using split injection strategy over a single pulsed injection strategy. Around 10% (by mass) fuel was injected late in the compression stroke, while the rest was injected early in intake stroke.

After that, a second direct injector was used to supply fuel in the late injection pulse. The use of a different injector allowed varying the fluids in the second pulse of a split injection strategy.

Table 1. Single cylinder engine specifications.

Displaced volume	454 cm^3
Stroke	86 mm
Bore	82 mm
Compression ratio	9.7
Valvetrain	4 valve DOHC
Cylinder head	Pent-roof chamber

Table 2. Test conditions used in this study.

Engine speed (rpm)	2,000
Fuel Temperature (ºC)	20
Coolant temperature (ºC)	80
Oil temperature (ºC)	80
Relative air-fuel ratio (λ)	1.00
Spark timing (CAD aTDC)	0 (methodology 1) variable (methodology 2)
Intake pressure (abs)	2.1 bar

Gasoline used in the current study is Coryton Euro V certified. The details of the gasoline used are provided in Table 3 and Fig. 1 [48]. An SAE 5W30 compliant lubricant was used. The lubricant was kept the same for the tests.

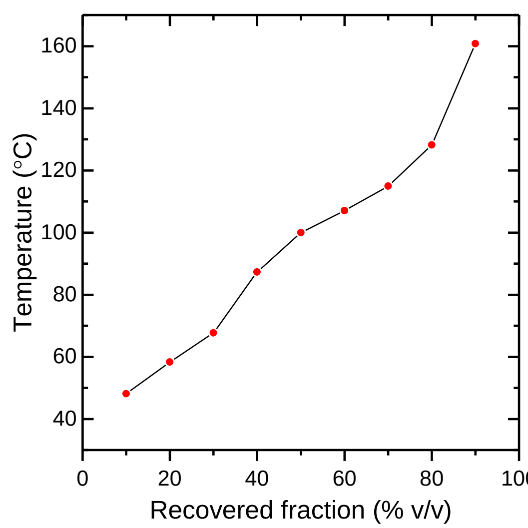

Figure 1. Measured distillation properties for Coryton Gasoline.

Table 3. Properties of the Coryton Gasoline used in this study.

Research octane number (RON)	97.5
Motor octane number (MON)	86.6
Specific Gravity (SG)	0.7485
Lower heating value (MJ/kg)	42.4

Energy density (MJ/L)	31.7
Aromatics (% v/v)	30.5
Olefins (% v/v)	8.2
Ethanol (% v/v)	5.0
H/C ratio	1.776
O/C ratio	0.015

The fuel was supplied to a 6-hole nozzle (diameter 0.18 mm each) via a fuel flow meter (based on Coriolis principle) and a fuel conditioning unit that maintains the temperature of the fuel at 20 °C. The second fluid was supplied to a side injection having a 7-hole nozzle (diameter 0.17 mm each) via a fuel flow meter (based on gravimetric principle) and a fuel conditioning unit that maintains the temperature of the fuel at 20 °C. The injection pressure for both the fuel lines was maintained at 130 bar via a solenoid control. Air mass flow was controlled using a mass flow controller to fix the intake pressure at 2.1 bar (abs). The fuel flow rate was controlled to maintain stoichiometric operation, as measured by a wideband UEGO lambda sensor, mounted 10 cm downstream of the exhaust valves. No back pressure was applied in the current tests. Turbocharged engines have a high exhaust back pressure usually. Increasing exhaust pressure increases the residual mass, which may carry pre-ignition precursors. Hence, increasing exhaust back pressure (more representative of a real engine) could yield more pre-ignition events [49]. However, when comparing different strategies, a constant backpressure needs to be applied and fixed for all cases (zero in the current study). Doing so provides a good one-to-one comparison of different strategies [50]. A steady state was maintained in the experiments by controlling oil and coolant temperature. A warm-up of the engine and steady state was ensured before recording the data. Figure 2 shows the relative location of sensors, spark plug, and injectors inside the cylinder head.

An in-cylinder K-type thermocouple was used to measure the average temperature of an operating point. The thermocouple is embedded in a glowplug, supplied from Borg Warner. The details of the thermocouple are added as an appendix to the main text. The choice of measuring this parameter is driven by previous works by Downs *et al* [17, 51], wherein pre-ignition rating of fuels and operating conditions were inversely proportional to such temperature measurements. However, there have been no efforts to relate pre-ignition tendency to the thermocouple reading lately.

The fuel injection was split into two pulses. Main injection start of injection (SoI) was fixed at -300 CAD aTDC, while the SoI of the second injection was parametrically varied from -45 CAD aTDC to -15 CAD aTDC. At each SoI point, the duration of injection (DoI) of the fluid in the second injection was varied from 0.3 ms to 1.1 ms. For a fixed DoI, the mass of fluid injected depends on the density of the fluid. Therefore, values were converted into mass of fluid injected in each cycle for each fluid and reported. For each SoI-DoI combination, three sets of 15,000 cycles were recorded, and the average data of the 45,000 cycles is presented in the results and discussion section. Spark timing was fixed at 0 CAD aTDC for the first set of experiments where fluids in the late injection are varied.

For the next set of experiments, the effect of water injection was studied in intake, compression and exhaust stroke. Spark timing was varied for being knock limited (KI ~ 0.5 bar). More details on the operating conditions for this case are provided later.

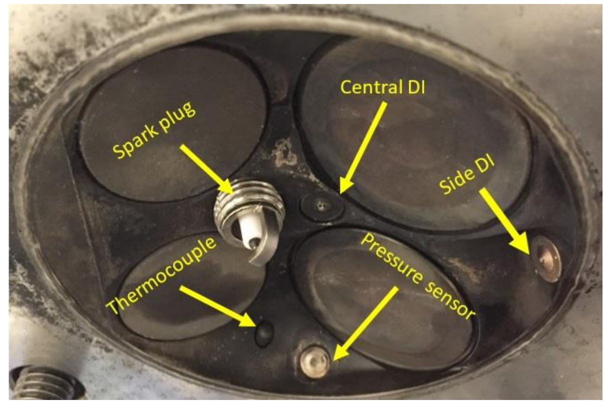

Figure 2. The cylinder head is showing the relative location of the spark plug, the pressure sensor, direct injectors, and thermocouple.

Pre-ignition metrics used:

Pre-ignition is a stochastic phenomenon and hence recording a few cycles can lead to misleading results. The current study was taken with 3 sets of 15,000 cycles (total of 45,000 cycles recorded) at each operating condition. The reported data is an average of the three sets. CA05 (start of combustion) was used as a marker for pre-ignition determination. A robust statistical method proposed by Zaccardi *et al.* [52] for use in pre-ignition studies, was applied to quantify pre-ignition counts. Equation (1) shows the criterion used:

$$PI\ count = \#(R\theta(CA05) - 4.7R\sigma(CA05)) \ldots \ldots \ldots \ldots \ldots \ldots (1)$$

Where $R\theta$ refers to robust mean and $R\sigma$ refers to robust standard deviation of the dataset. Pre-ignition often occurs as alternating sequence, wherein a pre-ignition event is followed by a seemingly-normal cycle, followed by another pre-ignition event. In such cases, only one pre-ignition event was counted in our study. The pre-ignition events were counted as one, if the next pre-ignition event occurs within 3 cycles.

Fluids used:

As previously mentioned, the fluid injected in the second pulse of the split injection was varied. Four different fluids: gasoline, ethanol, methanol, and water were used for the tests. The properties of the fluids are given in Table 4.

Table 4. Properties of the fluids used in late injection pulse.

Property	Gasoline	Ethanol	Methanol	Water
Molecular formula	$(CH_{1.9}O_{0.02})_x$	C_2H_5OH	CH_3OH	H_2O
Molecular weight	111.52	46.07	32.04	18.02
Density (kg/m^3)	747	789	792	1000
RON/MON	96/85	109/90	109/89	-
Latent heat of evaporation (J/g)	422	841	1104	2257
Calorific Value (MJ/kg)	41.12	26.7	19.9	-
Stoichiometric air-fuel ratio	14.7	9	6.47	-

Boiling Point (ºC)	Figure 1	78.37	64.7	100

Results and Discussions:

A late split injection strategy was suggested by [32]. The authors reported pre-ignition free operation at an IMEP of 22.7 bar while using ethanol as a fuel. We carried out a similar study using gasoline as fuel. The injection pulses were split into two and three, with the aim of suppressing pre-ignition occurrence while not reducing IMEP or increasing CoV greatly [47]. Results from one of the optimized injection strategies are shown in figure 3, which shows the knock intensity (KI) as a function of CA05 for the recorded cycles. The injection pulse was split into two pulses; the second pulse (~10% of total fuel) was injected at SoI of -30 CAD aTDC. Stoichiometric operation was maintained. The pre-ignition count and knock intensity greatly reduced when using a late injection strategy. There were two pre-ignition events in this case, but none of them led to super-knock.
Our previous work [30] demonstrated that split injection (with equally split pulses) led to a lower in-cylinder temperature near the top dead center. Pre-ignition usually starts in a turbocharged engine around the top dead center, as observed in figure 3. Two factors, the mass of liquid fuel hitting the liner and the temperature drop (relative to a motoring case) were seen to affect pre-ignition [30]. However, as the fraction of fuel in the split pulse is small in the suggested strategy (~10%), the late split injection strategy may not heavily reduce the mass of fuel hitting the liner. The disproportionate reduction in pre-ignition indicated that charge cooling could be the dominant effect.

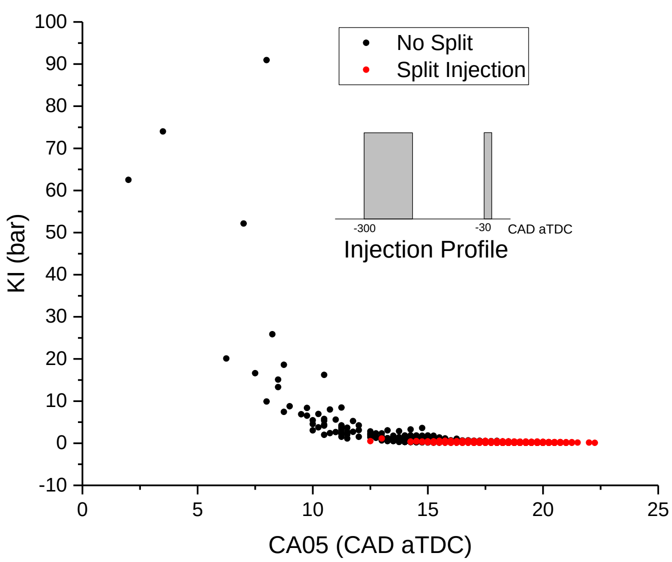

Figure 3: Knock Intensity plotted versus the start of combustion (CA05). In black is a baseline case, with single pulse injection at -300 CAD aTDC. Red points are for a split injection strategy with 10% fuel injected at -30 CAD aTDC. Spark timing was fixed at 0 CAD aTDC.

Effect of varying fluids in late split injection pulse

A second injector located laterally, was used to supply gasoline in the late injection strategy. Using the second injector allowed injecting different fluids in the late injection pulse. This injector is different from the centrally mounted injector used to supply majority of the fuel (details provided in methodology section). Previous studies have shown that injection via side-mounted injector led to lower liquid spray impingement on the liner and higher temperature reduction from fuel evaporation. Consequently, injecting complete fuel via side mounted injector led to lower pre-ignition frequency [30]. In contrast, we supplied most of the fuel via centrally mounted injector, except, during the late injection, where the fuel was supplied via side-mounted injector.

The SoI of the late injection fuel and the DoI of the fuel was varied parametrically. Figure 4 shows the effect of varying mass of fuel injected for an SoI of -15 CAD aTDC. The figure shows IMEP (black), average thermocouple temperature (green), pre-ignition count (red) and CoV of IMEP (blue). It was observed that the IMEP decreases and CoV increases with increase in fuel fraction injected in late injection strategy. The late split injection was highly effective at suppressing pre-ignition. The reduction in IMEP was due to smaller pre-mixed mixture available at spark timing, which was coincidental with the reduction of average in-cylinder temperature.

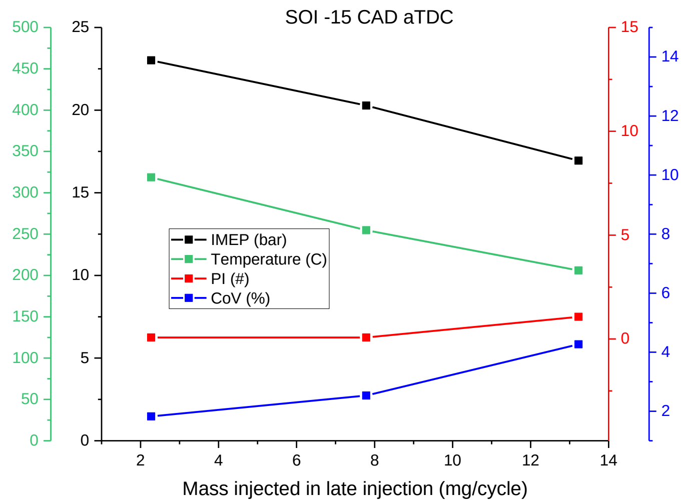

Figure 4: IMEP (black), Average in-cylinder temperature (green), number of pre-ignition events per 15,000 cycles (red) and coefficient of variation (CoV) of IMEP (blue) shown for varying fraction of gasoline injected in late injection strategy. Late injection SoI is -30 CAD aTDC.

Moreover, CoV increased as DoI increased. CoV is often correlated to heterogeneity around the spark plug, at the time of spark ignition [53]. With fuel injection closer to TDC, the heterogeneity of fuel-in-air was expected to increase with an increase in DoI, as the time for fuel to evaporate and completely mix is limited.

The SoI of the late injection strategy was varied parametrically to -30 and -45 CAD aTDC. The results are illustrated in figure 5. As the SoI was advanced from -15 to -45 CAD aTDC, the average IMEP increased. This observation corroborates with the fact that there is more time for the late injected fuel to mix and take part in combustion, providing higher combustion efficiency. Total hydrocarbon emissions increased with late injection strategy, and furthermore with the increasing mass of fuel in the late injection strategy. The emissions analysis is not in the scope of current work and will be part of future work. Pre-ignition suppression was highly effective at all the SoIs considered. The addition of fuel close to spark timing also inevitably increased heterogeneity close to the spark plug, which increases the cycle-to-cycle variation, thus increasing CoV as the DoI increases. With a more advanced timing, there was more time for fuel to mix before the spark plug ignition (as the spark timing was kept fixed at TDC), which resulted in lower CoV.

Next, ethanol was used for the second pulse of the split injection. Main fuel was still Coryton Gasoline. Ethanol has a relatively higher latent heat of vaporization that should result in a greater pre-ignition suppression tendency, if the cooling effect is responsible for the previous observations. Parametric sweeps of SoI and DoI are shown in figure 6. In terms of IMEP and CoV, using ethanol in late injection followed a similar trend as using gasoline. Notably, ethanol is reported to have a high pre-ignition tendency as a fuel [17, 51, 54, 55]. However, when using ethanol in late injection, it was found to be highly efficient. There was no increase in pre-ignition count upon raising the ethanol content.

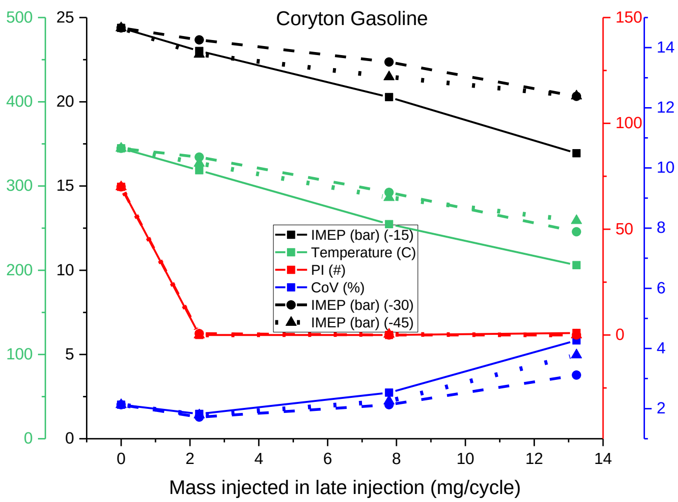

Figure 5: IMEP (black), Average in-cylinder temperature (green), number of pre-ignition events per 15,000 cycles (red) and coefficient of variation (CoV) of IMEP (blue) shown for varying fraction of gasoline injected in late injection strategy. Late injection SoI is varied from -15 CAD aTDC (solid line), -30 CAD aTDC (dashed line) and -45 CAD aTDC (dotted line).

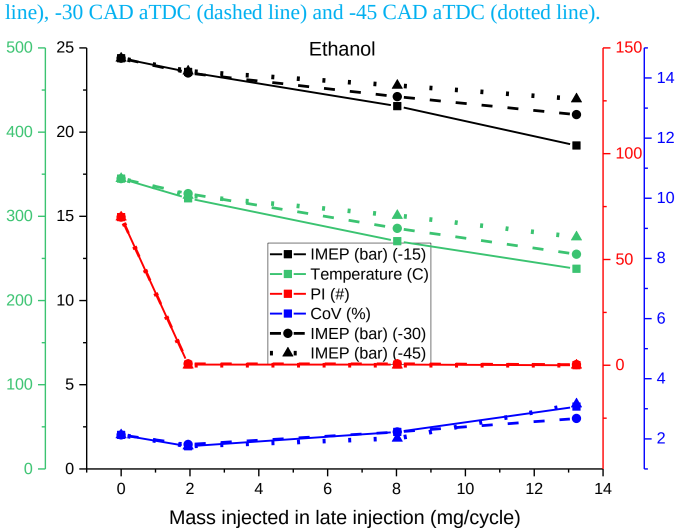

Figure 6: IMEP (black), Average in-cylinder temperature (green), number of pre-ignition events per 15,000 cycles (red) and coefficient of variation (CoV) of IMEP (blue) shown for varying fraction of ethanol injected in late injection strategy. Late injection SoI is varied from -15 CAD aTDC (solid line), -30 CAD aTDC (dashed line) and -45 CAD aTDC (dotted line)

The same tests were performed using methanol in the late injection. The results are depicted in figure 7. Methanol has a greater pre-ignition propensity in comparison to ethanol as a fuel [17, 51]. Nonetheless, using methanol for the late split pulse only, proved extremely effective at suppressing pre-ignition. There was a trend of increasing pre-ignition with increasing methanol injection in late injection, which was contrary to other fuels tested till now. IMEP and CoV followed a similar trend as seen in previous cases.

Lastly, water was injected in late injection pulse. The results are shown in figure 8. As water has zero calorific value, stoichiometric fuel quantity is injected at -300 CAD aTDC. All the injected fuel mass got relatively larger time to mix before the spark timing, which led to a relatively higher IMEP. Apart from being effective at pre-ignition suppression, there were no compromises of increased CoV and decreased IMEP, as observed in previous cases. There was enough time for fuel-air mixture formation before water was injected, which means that water injection did not interfere with the mixture formation process. This allowed a negligible change in CoV compared to the baseline case. On the contrary, CoV slightly decreased in some cases. It must be noted that injecting stoichiometric fuel mass in a single pulse led to slightly more fuel-oil interaction in this case. Moreover, the higher in-cylinder temperature is also correlated to higher pre-ignition count [17, 47, 51]. Hence water injection was slightly less effective at pre-ignition suppression, as will be seen later.

The four fluids tested in the current work were compared at SoI of -15 CAD aTDC in figure 9. It was observed that IMEP was lowest upon injecting gasoline in late injection pulse, followed by ethanol, methanol, and water respectively. Following the stoichiometric air-fuel ratio in Table 3, for the same DoI of gasoline, the smallest fraction was injected at -300 CAD aTDC (available for pre-mixing), hence leading to larger IMEP drop. Ethanol had the second highest air-fuel ratio, followed by methanol and then water. Conclusively, fraction of the fuel injected in the first pulse to maintain the stoichiometry is gasoline<ethanol<methanol<water. This was also the order of IMEP as seen in figure 9.

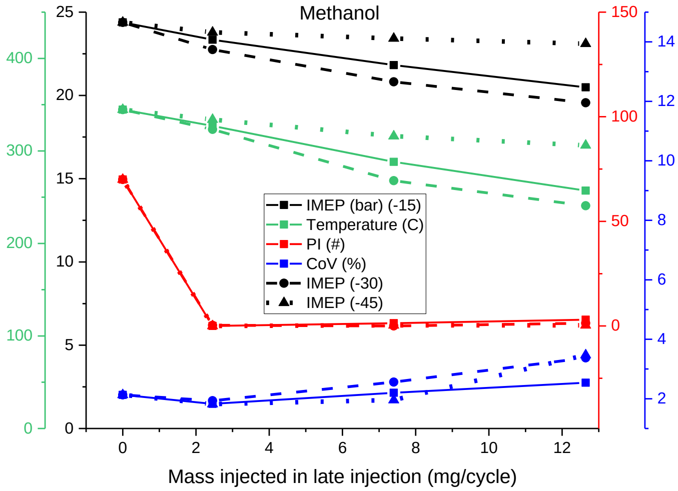

Figure 7: IMEP (black), Average in-cylinder temperature (green), number of pre-ignition events per 15,000 cycles (red) and coefficient of variation (CoV) of IMEP (blue) shown for varying fraction of methanol injected in late injection strategy. Late injection SoI is varied from -15 CAD aTDC (solid line), -30 CAD aTDC (dashed line) and -45 CAD aTDC (dotted line).

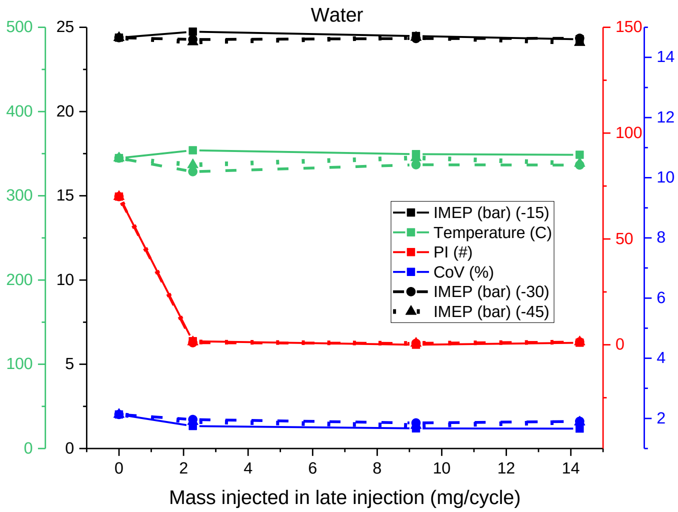

Figure 8: IMEP (black), Average in-cylinder temperature (green), number of pre-ignition events per 15,000 cycles (red) and coefficient of variation (CoV) of IMEP (blue) shown for varying fraction of water injected in late injection strategy. Late injection SoI is varied from -15 CAD aTDC (solid line), -30 CAD aTDC (dashed line) and -45 CAD aTDC (dotted line).

Overall, splitting the injection with SoI of second pulse close to TDC is a very effective strategy to suppress pre-ignition. All the fluids tested were successful at suppressing pre-ignition, with varying degree of effectiveness. The strategy, however, leads to compromises on IMEP and CoV, when using flammable fluids, gasoline, ethanol, and methanol. Higher the mass of fuel injected in the first pulse, greater is the IMEP. For a fixed DoI, the contribution to stoichiometric operation from the fluids follow the trend: water<methanol<ethanol<gasoline. Hence, the mass of gasoline in the first pulse follows the reverse trend (water>methanol>ethanol>gasoline), which is also the order of IMEP observed in figure 9. Latent heat increased in the order: water>methanol>ethanol>gasoline. Conversely, pre-ignition suppression did not follow this trend. The late injection timing may not allow enough time for full vaporization by the time of ignition. Consequently latent heat of vaporization could only be partially utilized to cool the chamber. Considering the pre-ignition tendency of the fluid by themselves, researchers depict the following trend: methanol>ethanol>gasoline [17, 51]. This effect was not observed in our experiments, which is a testament to the dominance of physical properties in the effectiveness of the late split injection strategy. The effect of increasing pre-ignition count with increasing DoI was only observed in the case of methanol.

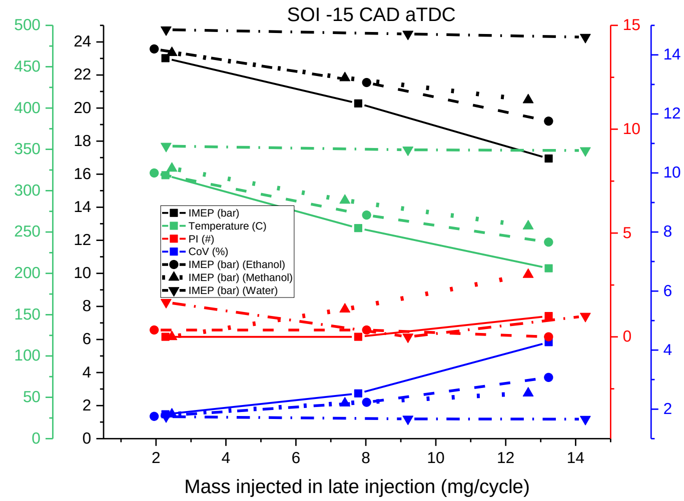

Figure 9: IMEP (black), Average in-cylinder temperature (green), number of pre-ignition events per 15,000 cycles (red) and coefficient of variation (CoV) of IMEP (blue) shown for varying fraction of water injected in late injection strategy. Late injection SoI is -15 CAD aTDC. Gasoline (solid line), Ethanol (dashed line), Methanol (dotted line) and Water (dash-dot).

Effect of water injection on pre-ignition suppression

Water injection proved to be best at reducing compromises associated with late injection strategy. Due to the standard methodology, previous tests were performed with a fixed spark timing. However, in the next section, the effect of water injection in the second pulse was tested for SoI in intake, compression, and exhaust stroke, while varying the spark timing for knock limited condition. Initially, water was injected in the intake stroke. Two different SoI were chosen: -330 CAD aTDC and -200 CAD aTDC, while the intake valve was open. Two different SoI were chosen in compression stroke: -90 CAD aTDC and -30 CAD aTDC. Two SoIs were also chosen in late exhaust stroke: 330 CAD aTDC and 350 CAD aTDC. The SoIs in exhaust stroke were used to selectively cool and quench the residuals that may trigger pre-ignition in the next cycle, without interfering with the next cycle in any other way. Therefore, the injection was made slightly before the exhaust valves closed, and the intake valves opened.

The three cases considered in our study refer to the differing mechanisms by which pre-ignition suppression was targeted. Injecting in the intake stroke is akin to introducing cooled EGR, which is a known strategy to suppress reactivity of the charge. The charge cooling effect is present, but not dominant. Injecting in the compression stroke targets charge cooling more than chemical effects. Injecting a fuel closer to TDC in the compression stroke utilizes the enthalpy of vaporization better, provided all of the fuel is evaporated [30].

Injecting in the exhaust stroke targets reducing the potential of the residuals in triggering pre-ignition in the next cycle. Particles, that can trigger pre-ignition in the next cycle, can be completely quenched by water. Alternatively, the average temperature of the residuals can be reduced to the point that it cannot trigger pre-ignition. The various combinations employed in our study are shown in figure 10.

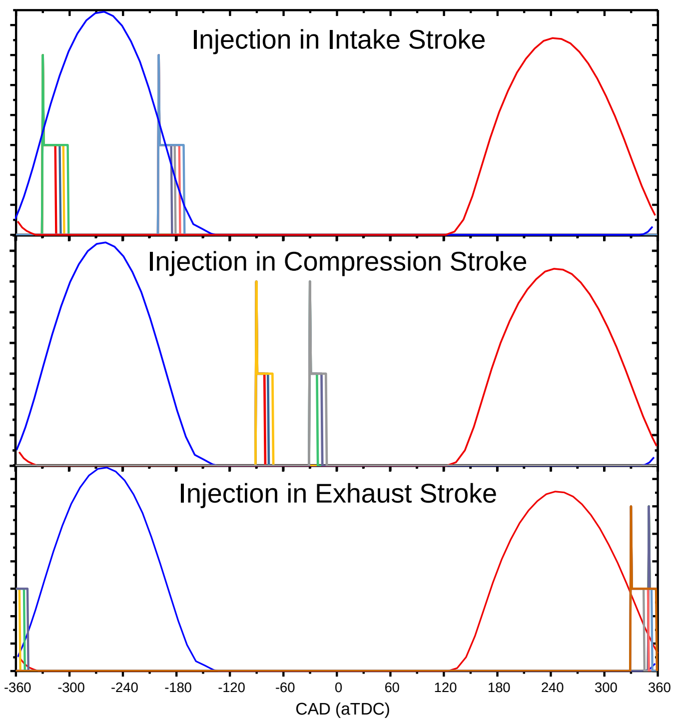

Figure 10: Injection strategies employed for water injection. Six different SoI in intake, compression and exhaust stroke are employed with 3-4 DoI at each DoI. 45,000 cycles are recorded at each point, and average data is reported. Intake (blue) and exhaust valve (red) lift is shown for reference. The spike is to show the SoI, while the width corresponds with the DoI of each case.

In first case, water was injected into the intake stroke. The results are shown in figure 11. Injecting water in the intake stroke reduced the specific heat ratio of the charge, leading to lower in-cylinder temperature. Reduction in overall in-cylinder temperature led to a reduction in knock tendency. This allowed spark timing to be advance further, improving the net IMEP. Bulk mixture temperature near TDC has frequently been correlated to pre-ignition tendency of an operating condition. However, the effect is not drastic compared to the charge cooling effect, which is more prominent when injecting late in the compression stroke. The later the injection is done, charge cooling is more dominant and effective [30]. SoI of -200 showed higher IMEP in general.

For the second case, water was injected in the compression stroke. The results are shown in figure 12. Injecting in the compression stroke did not affect the overall specific heat ratio, as much as it reduced bulk gas temperature via charge cooling. Two different SoI were used for this reported case: SoI of -90 CAD aTDC and -30 CAD aTDC. As evident, SoI of -90 CAD aTDC performed exceptionally well for IMEP growth and CoV reduction. SoI of -30 CAD aTDC did not show pre-ignition-free operation, even at a higher injection duration. This can be attributed to a smaller fraction of injected water being vaporized as the SoI is closer to spark timing. The later injection timing did not allow the full potential of charge cooling to be utilized before the spark ignites the mixture.

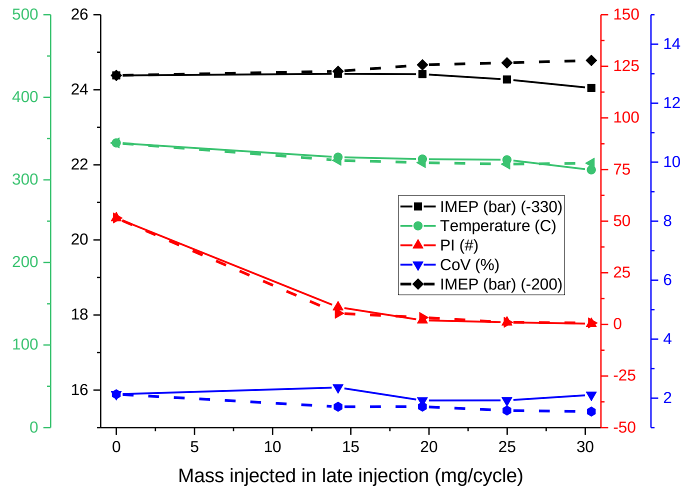

Figure 11: IMEP (black), Average in-cylinder temperature (green), number of pre-ignition events per 15,000 cycles (red) and coefficient of variation (CoV) of IMEP (blue) shown for varying fraction of water injected in late injection strategy. Water injection SoI is -330 CAD aTDC (solid line) and -200 CAD aTDC (dashed line).

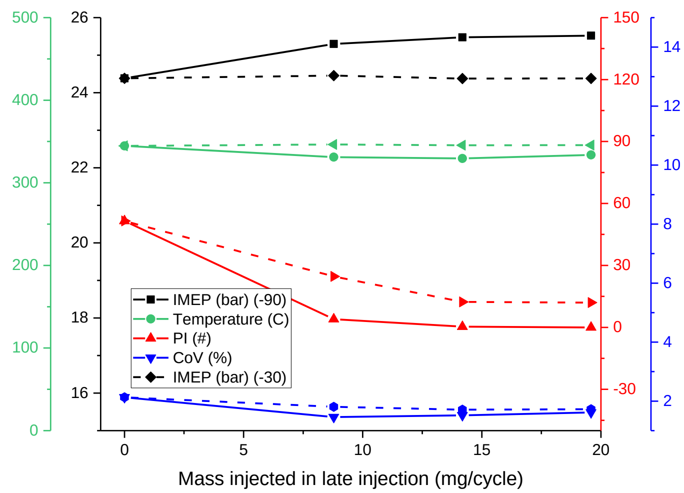

Figure 12: IMEP (black), Average in-cylinder temperature (green), number of pre-ignition events per 15,000 cycles (red) and coefficient of variation (CoV) of IMEP (blue) shown for varying fraction of water injected in late injection strategy. Water injection SoI is -90 CAD aTDC (solid line) and -30 CAD aTDC (dashed line).

For the third case, water was injected into the exhaust stroke. Recent research elucidates that pre-ignition precursors may miss the scavenging process and be trapped in the cylinder [3, 54, 56, 57]. These particles can trigger pre-ignition in the next cycle. Therefore, the injection was made before the exhaust valves close and the intake valves open. This should allow quenching of the reactive particles in the residuals. Figure 13 shows the effect of injecting water late in the exhaust stroke. The reduction in pre-ignition was comparable to previous cases, although the fresh charge was not cooled in this case. The IMEP was also comparable to the case without water injection, with the added advantage of no pre-ignition. SoI of 330 was more effective at suppressing pre-ignition than SoI of 350 CAD aTDC. Taking a closer look at figure 10, it is observed that SoI of 330 CAD aTDC had the end of injection before the intake valve opened, at all the considered DoIs. It is 'selectively' cooling the residuals more, while SoI of 350 CAD aTDC overshoots slightly in the intake valve open region.

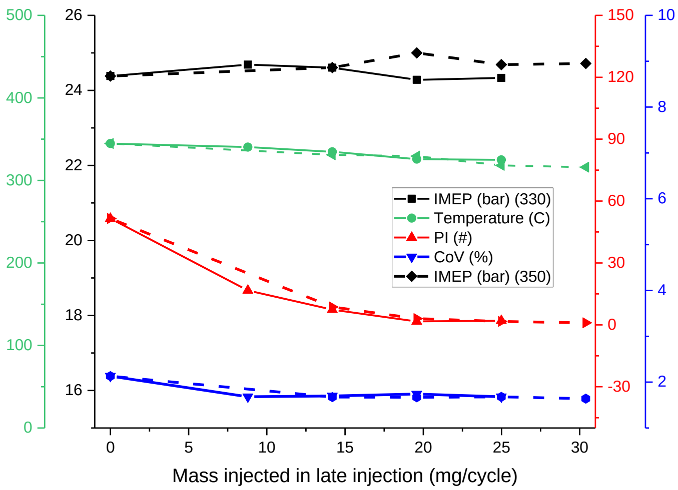

Figure 13: IMEP (black), Average in-cylinder temperature (green), number of pre-ignition events per 15,000 cycles (red) and coefficient of variation (CoV) of IMEP (blue) shown for varying fraction of water injected in late injection strategy. Water injection SoI is 350 CAD aTDC (solid line) and 330 CAD aTDC (dashed line).

Injecting water in the intake and compression stroke gives ample time for water to evaporate. The effectiveness of water injection in the exhaust stroke to suppress pre-ignition was surprising and implicates the role of residuals in triggering pre-ignition event. Figure 14 shows a comparison between the pre-ignition count for different water injection SoIs. SoI of -90 CAD aTDC was the best performing strategy. Although, both. -90 and -30 CAD aTDC SoIs were in the compression stroke, SoI of -90 CAD aTDC ensured that all the water was evaporated up to TDC. Temperature reduction was relatively lower (and later) with SoI -30 CAD aTDC. The other SoIs fell in between these two SoIs.

As previously mentioned, the spark timing was varied for these cases. The varying spark timing reflects the bulk mixture reactivity. Figure 15 shows the spark timing for the varying SoIs mentioned before. This figure highlights that spark timing (and hence, the bulk mixture reactivity) for the water injection in intake and compression stroke correlated well with the pre-ignition count (figure 14). The same cannot be said for water injection in the late exhaust stroke. SoIs in intake and compression stroke directly impacted the temperature near TDC (from charge cooling). In contrast, injecting in the exhaust stroke should cool the residuals (and quench the pre-ignition pre-cursors), instead, the bulk mixture temperature was reduced only slightly. This explains the fact that although pre-ignition suppression is good for late exhaust stroke injection timing (due to quenching of pre-ignition precursors), it doesn't affect the bulk mixture temperature as much.

Pre-ignition occurrence depends on the probability of activated precursors surviving the exhaust scavenging process (residuals for next cycle) and their ignition in the presence of an ignitable atmosphere [49]. Injecting water in the late exhaust stroke should reduce the probability of the former from occurring (should quench the precursors) while injecting water in the intake and compression strokes would ensure that the latter is reduced (charge cooling effect).

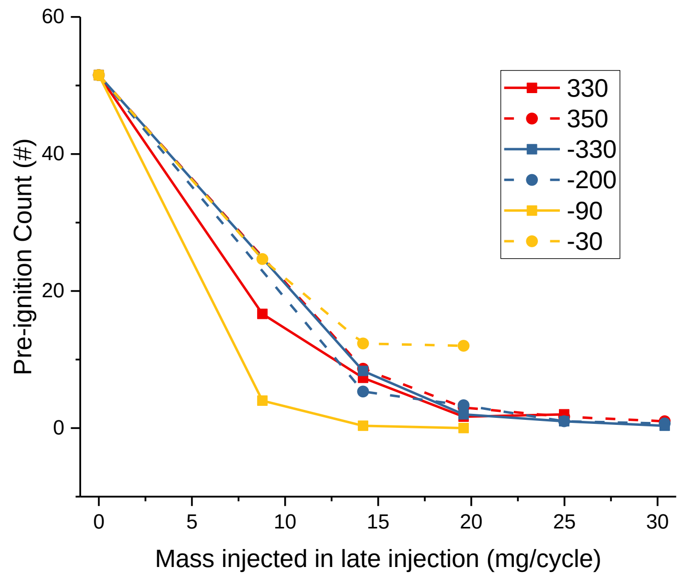

Figure 14: Pre-ignition count for water injection at different SoIs. Red points refer to injection in the late exhaust stroke, blue points refer to injection in the intake stroke and yellow points refer to injection in the compression stroke.

Overall, these observations fall in line with the previous observations by other authors and the authors of this work [30, 58]. Quenching pre-ignition precursor reduces the probability of potential precursors in the next cycle. Additionally, ensuring a colder bulk mixture will reduce the probability of those precursors to form a successful flame front, leading to pre-ignition suppression.

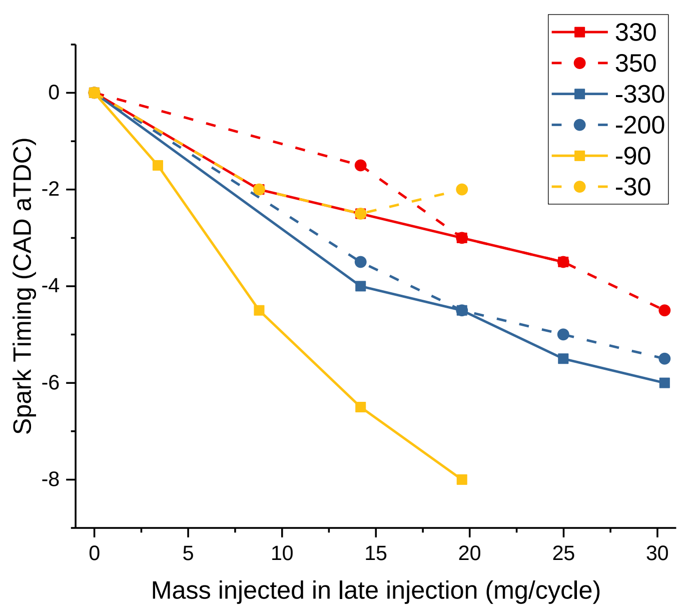

Figure 15: Spark timing for water injection at different SoIs. Red points refer

to injection in the late exhaust stroke, blue points refer to injection in intake stroke and yellow points refer to injection in compression stroke.

Conclusions:

This study explored the effectiveness of different fluids in suppressing pre-ignition. A split injection strategy is suggested for pre-ignition suppression. Gasoline was injected in the first (main) pulse, while gasoline, ethanol, methanol and water were injected in second pulse. The key findings of the study are as follows:

- A split injection strategy was proposed to suppress pre-ignition in a turbocharged engine, where a small fraction of the total fuel is injected late in compression stroke.
- Retarding the injection timing of late injection and increasing the injected mass in the late injection reduced the IMEP and increased the CoV. This observation was common to gasoline, ethanol or methanol, whereas injecting water had no major effect on IMEP and CoV. The fraction of fuel injected in the main pulse was critical to the magnitude of these compromises.

Water injection showed considerable benefits in terms of the compromises associated with late injection strategy, like IMEP reduction and CoV increase. Water injection was tested in intake, compression and late exhaust stroke.

- IMEP values greater than baseline cases (with an added advantage of negligible pre-ignition) were achieved.
- Most effective utilization of charge cooling effect was realized by injecting water late in compression stroke. However, injecting very close to top dead center may not ensure complete vaporization before the ignition.
- Water was injected in the late exhaust stroke to selectively cool the residuals. The effectiveness of water injection in late exhaust stroke hints at the role of residuals on next cycle pre-ignition.

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Nomenclature

CAD	Crank Angle Degree	CA10	Crank angle degree for 10% mass fraction burned
aTDC	After Top Dead Center	MPRR	Maximum Pressure Rise Rate
bTDC	Before Top Dead Center	DI	Direct Injection
KI	Knock intensity	Pmax	Maximum Pressure
IMEP	Indicated Mean Effective Pressure	CO2	Carbon Dioxide
CoV	Coefficient of variation	θ	Average value
ST	Spark Timing	σ	Standard Deviation
Pin	Intake Pressure	R	Robust algorithm
SOI	Start of Injection	DOI	Duration of Injection
CA02	Crank angle degree for 2% mass fraction burned	NEDC	New European Driving Cycle
CA05	Crank angle degree for 5% mass fraction burned	PI	Pre-ignition
		DOE	Design of Experiment

Appendix

In-cylinder thermocouple measurement

The thermocouple used in the current study was manufactured by Borg Warner, and was embedded inside a glow plug that was mounted inside the combustion chamber (figure A1). A K-type thermocouple was used as feedback for closed-loop control of glow plug temperature. A cross-sectional view of the glow plug is presented in figure A2.

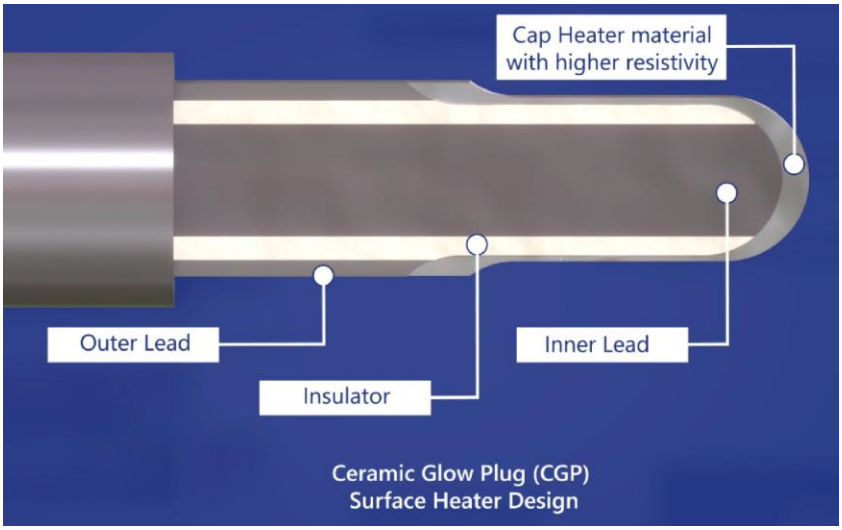

Figure A1. Glowplug used in current study as a thermocouple (courtesy: www.borgwarner.com)

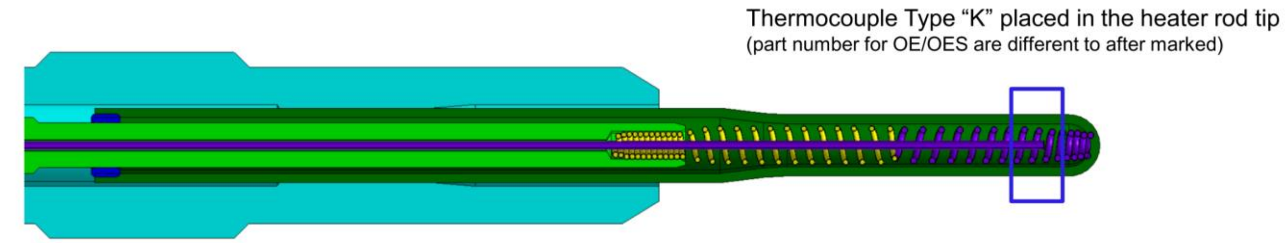

Figure A2. Cross-section of the glowplug showing K-type thermocouple used in current study (courtesy: www.borgwarner.com)