

Fundamental Approach to Investigate Pre-Ignition in Boosted SI Engines

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ABSTRACT

A comprehensive experimental and theoretical approach was undertaken to understand the phenomenon of pre-ignition and to assess parameters to improve or even eliminate it completely. Oil mixing with fuel was identified as the leading theory of self ignition of the fuel. End of compression temperature has to meet a minimum level for pre-ignition to take place. In this work a comprehensive list of parameters were identified that have a direct and crucial role in the onset of pre-ignition including liner wetting, injection targeting, stratification, mixture motion and oil formulation. Many secondary effects were identified including ring dynamics, ring tension, spark plug electrode temperature and coolant temperature. CFD has been extensively used to understand test results including wall film, A/F ratio distribution and temperature at the end of compression when looked at in the context of fuel evaporation and mixing. Consecutive high speed imaging and other optical measurements were implemented to capture and understand pre-ignition. Quantitative post-processing techniques have been used to convert video to quantifiable parameters. A 3-D comprehensive model of ring dynamics was created to

understand the stochastic nature of oil intrusion into the combustion chamber.

INTRODUCTION

Pre-ignition is premature combustion that takes place before the spark in a highly boosted engine causing a loud objectionable noise and a cylinder pressure up to 350 bar. It is extremely important to understand and control pre-ignition since it is the building block for the downsizing and boosting strategy to improve fuel economy and is extremely damaging to engine structure.

This combustion phenomena might not be new since previous papers from the 1950ies [10, 11] did report pre-ignitions. Those observations might be linked to the subject of this paper although the charge densities, compression ratios and especially the level of deposits in the engines are different. The approach which has been taken in this work is to attempt to fundamentally understand why fuel that has very high auto ignition temperature ($\sim 1050\text{K}$) self ignites before spark at temperatures ranging from (650-700K). One could assume that this is the result of surface temperature due to hot spots in the combustion chamber from either hot components

(spark plug or valve) or even soot accumulation. In the testing which was conducted for this study both of these two most obvious causes were ruled out as will be shown in the experimental section. Furthermore the stochastic pre-ignition phenomena even has a self curing character, since it often shows intermittent events, where a regular combustion follows after a pre-ignition before the next pre-ignitions starts again. Only extremely rarely two pre-ignitions in a row can be observed. Similar to other authors [4] we also have not found any evidence in the combustion data before the pre-ignition, which would indicate the following pre-ignition.

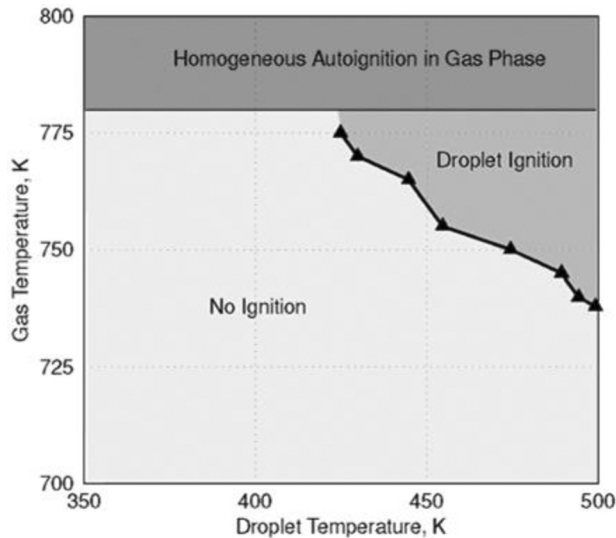


Figure 1. Ignition limits assuming an ignition delay time of 5 ms and n-heptane droplets [13]

Another theory is the lowering of auto ignition temperature of fuel using an agent such as engine oil. This hypothesis has been supported qualitatively by reaction kinetic results in [13] although the assumed delay time is not realistic for the experiments Figure 1. This work will establish oil mixed with fuel as the leading theory of lowering self ignition of fuel and hence causing premature combustion before spark. All the parameters affecting pre-ignition are in line with this theory as will be shown using testing, High Speed Imaging (HSI), analysis using Computational Fluid Dynamics (CFD) and dynamic simulation of piston rings. It will be shown that injection targeting is crucial in both direct and port injection engine, hence mixture preparation and stratification. Lowering coolant temperature affects fuel evaporation and negatively impacts pre-ignition. Start of injection is a crucial parameter affecting pre-ignition since this factor affects mixture preparation and interaction with combustion chamber wall (fuel penetration). Increasing inlet air temperature will reduce pre-ignition due to improved fuel evaporation and mixing, yet too high of an inlet air temperature starts affecting the end of compression temperature which in turn increases the propensity to pre-ignition. Another very important factor for Pre-ignition is spark plug indexing. This

work shows that there is a definite direction of the spark plug electrode that causes an increase in pre-ignition. In this work we investigated the effect of the crevice volume, compression ratio, and most importantly injection targeting in a direct injection engine.

In pursuit of the theory of oil mixing with the fuel, we created a three dimensional model of piston ring dynamics. This model has all the contacts between rings and cylinder liner and all gas exchanges happening between the cylinder and the crankcase. The simulation will show that the acceleration of the top ring could reach up to 2000 G. The positive flow of gases from crankcase to the combustion chamber gives the opportunity and the correct timing for particles of oil to be introduced into the combustion chamber and mixed with the fuel stochastically to cause pre-ignition.

Again, to pursue an understanding of some of the results we obtained in testing, a CFD model has been constructed to understand the effect of end of compression temperature, stratification in chamber, fuel preparation and fuel to wall interaction on pre-ignition.

This paper investigates all factors affecting pre-ignition utilizing testing and analysis/simulation. A theoretical foundation of understanding and masking the symptoms of pre-ignition is presented.

APPROACH

The approach taken in this study was to list all factors that could have any effect on pre-ignition including all the theories and design specific testing to either understand the affect of that factor, or in the case of theories to either confirm or disapprove them.

The theories we looked at included; one, hot spot where a potential of surface ignition could take place when the fuel contacts it. Thermal imaging was used in the combustion chamber to prove that pre-ignition is not due to a hot spot, as in the engine we tested it was not a runaway combustion. Pre-ignition occurred only once every 15000 cycles on average. Another theory involved fuel reformulation and radicals. This theory was never proven or disproven in this work and further studies are needed to specifically investigate that aspect of pre-ignition. The third theory was that soot particles accumulating in the chamber will be soaked with fuel and oil and hot enough to start premature combustion. This theory was disproven when the combustion chamber was thoroughly cleaned and as soon as the engine restarted pre-ignition reoccurred while the chamber was still soot free. The other theory which was pursued in this work was that for the Pre-ignition to take place in the engine, one has to lower the self ignition temperature of the fuel by about 500° K, or add a readily available agent like engine oil. The theory is that fuel would mix with oil in a certain area of the combustion

chamber in order to have enough combustible mixture to initiate Pre-ignition before the spark. As will be shown in this paper, much of the testing supported this theory and this remains the prevailing theory that could explain how pre-ignition takes place. All of the testing associated with masking the symptoms of pre-ignition are in support of this theory.

A comprehensive test matrix was put together to understand the effect of all the variables affecting pre-ignition. To ensure that all the testing was done under similar conditions for Pre-ignition, one single load and RPM Point of the engine, 2000 RPM, 22.1 bar bmep was selected. A standard procedure was used for preparing the engine and for the Pre-ignition test where we brought the engine gradually to the testing point.

Three things were pursued in the area of fuel preparation: rail injection pressure, injection targeting, and injection atomization characteristics. All of the three factors associated with fuel preparation have a great influence on the frequency of Pre-ignition. The test included-, injectors with injection targeting the liner walls, injection targeting the piston crown, narrow and wide injection umbrella angle, multiple hole injection vs. swirl injectors. Pistons were tested with compression ratios ranging from 8.4:1 to 10.3:1. Different crevice volume sizes were investigated to observe their influence on fuel and oil accumulation. Multiple tests were conducted by fixing the gap position of the rings and clocking it every 90° to investigate the influence of the ring gap on Pre-ignition. Spark plug electrode protrusion depth was investigated along with spark plug electrode clocking effects on Pre-ignition. Also the effects of coolant temperature, inlet air temperature were tested. From the above one can see that the list prepared for testing is very long and exhausting because when we started this experiment we had a very little knowledge about which factors influence the onset of pre-ignition. The prevailing theory of that time was that soot particles survived the intake stroke and injection of fuel to cause the premature combustion. Testing proved that soot has very little influence on pre-ignition. One could operate the engine with very high soot levels yet very low pre-ignition and vice versa. What was achieved in this work is identification of the knobs which have influence on pre-ignition to allow calibrators to calibrate out of pre-ignition zones and avoid those regions where pre-ignition could occur.

ENGINE HARDWARE

In this study two different types of 4-cylinder engines were used. The first one was a 2.0L spark-ignition direct injection (SIDI) engine with dual continuously variable cam phases (D-CVCP) and twin scroll turbo charger. This production engine is the same as the one used in the initial pre-ignition study published in [2]. Further details of the engine have been published in [1]. Some additional parameter studies and

especially the high speed imaging were conducted with a 1.6L prototype engine with central direct injection configuration. The specification of both engines is shown in [Table 1](#). The testing was conducted on engines with modifications to the cylinder head to accommodate four cylinder pressure transducers. The transducers were used for detecting pre-ignition during all the testing. Other modifications had to be made to the engine head to enable an optical access to the combustion chamber. Details will be discussed in the high speed imaging section.

Table 1. Engine Specifications.

Engine Type	Unit	2.0 I ECOTEC SIDI Turbo	1.6 L SIDI Turbo
Development level		Production	Prototype
Displacement	cm ³	1998	1598
Fuel Injection System	-	High Pressure SIDI	High Pressure SIDI
Injector location	-	Lateral	Central
Injector Type	-	Multihole	Multihole
Bore / Stroke	mm	86/ 86	79/ 81.5
Piston Cooling	-	Cooling channel & oil squirters	oil squirters
Valvetrain Type	-	DOHC w/ RFF Hydraulic lash adjuster	DOHC w/ RFF Hydraulic lash adjuster
Valve Diameter Ex / In	mm	30.1/ 35.1	27.5/ 30.9
Valve Lift Ex / In	mm	10.3 / 10.3	8.1/ 8.1
Cam Drive (phasing authority)		D-CVCP 50° / 50°	D-CVCP 60° / 60°
Opening Duration Ex / In (@1 mm valve lift)	°CA	194/204	186/191
MOP-Angle Ex / In @ parking position	°CA after gas exchange TDC	-120 / 126	-130/ 85
Compression Ratio	-	9.2	
BMEP @ 2000 rpm	Bar	22.1	
Specific Power	kW/l	97	
Engine Controller Software	-	Bosch MED 9.6.1	Inhouse GMPT
Recommended Fuel Quality	RON	95	95

DYNAMOMETER SETUP

The engines were equipped with the standard pressure and temperature transducers in the intake and exhaust systems, coolant and the oil circuit. Further measurement systems included a fuel consumption meter, wide band O₂-sensor and smoke meter. INCA 5.4 software was used to control ECU functions and engine parameters. Temporary exhaust gas analyzer, fast FID, opacimeter, mass spectrometer and low pressure analysis in intake and exhaust ports were applied. For dedicated thermal surveys the spark plug, cylinder head and cylinder block were equipped with thermocouples. Optical studies were conducted with fiber-optical spark plugs and through boroscopes. For detecting the pre-ignition, the indicating system is the most important measuring device. Cylinder pressure transducers in all four cylinders were used to trigger the recording of the pre-ignition event, whereupon 25 cycles before and after the event were logged from the continuous data acquisition buffer.

TEST PROCEDURE

Standard test procedure was strictly used to enable consistent and statistically reproducible results of all variants of the tests. All measurements were conducted at 2000 rpm and 22.1 bar bmep unless specified differently. For the 2.0L lambda was controlled to $\lambda \approx 1.1$ to represent transient fueling during acceleration. Other parameters were used as follows. [Figure 2](#) standard dyno test procedure2 shows the standard dyno test procedure. Each parameter or hardware variation was measured 3 times for 15 minutes each time. Between the 3 full load phases the engine was run at part load (2000 rpm, 2 bar bmep) for about 5 min. During the testing, engine oil was not conditioned. The oil temperatures in the main gallery reached a maximum value of about 106°C for the 2.0 L and about 102 °C for the 1.6 L at the end of the 15 min phase. If the maximum cylinder pressure exceeds 105 bar during the full load phase, saving of the pre-ignition pressure curve (and the other cycles before and after) is triggered and the number of trigger “events” are counted by the indicating system automatically. The triggered pre-ignition cycle was then visually checked, if the heat release did actually start before time of ignition in order to avoid miscounting e.g. due to extreme knocking events. Also the 25 cycles following

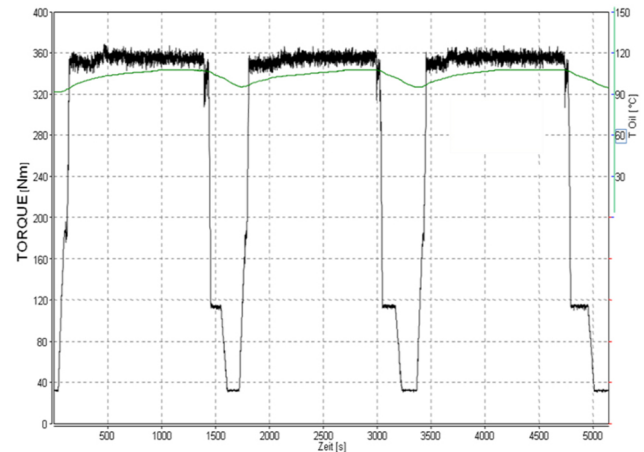


Figure 2. standard dyno test procedure

the initially triggered pre-ignition cycle was “manually” analyzed for follow up events. All those pre-ignitions were also included in the pre-ignition numbers quoted. Finally from all three full load phases the average pre-ignition frequency per 15 min is calculated. This number then represents the total number of pre-ignitions of all four cylinders. All other dyno parameters (e.g. pressures, temperatures, ECU parameters, fuel consumption, exhaust emissions and FSN) are saved as mean values during each 15 min phase.

Table 2. EMS and Dyno Settings.

EMS & dyno settings	units	2.0 L	1.6 L
		engine	engine
SOI	°CA bTDC	305	270
Fuel Rail Pressure	bar	150	180
Spark timing (KL)	°CA bTDC	~2	~ - 4
MOP ex	°CA aGexTDC	-84	-84
MOP in	°CA aGexTDC	86	90
Lambda exhaust	-	1,1	0,95
Lambda cyl (GT power)	-	0,83	0,88
Residuals (GT power)	%	4,5	1
Coolant out temperature	°C	90	95

HIGH SPEED IMAGING

In the pursuit of understanding the root of pre-ignition, a high speed camera was employed to acquire high speed imaging in the combustion chamber. The engine head was fitted with two optical access holes, one for the boroscope and the second for a light source or a second boroscope interchangeably. A Phantom 7.3 camera was used which is capable of acquiring images at a rate up to 160,000 frames/second (fps). A photo intensifier was attached between the boroscope and the camera to improve clarity and maintain a high image acquisition rate. Another advantage of the intensifier is that it allows a wider range of light wavelengths to be captured, especially when laminar flames are present. See [Figure 3](#) for the experimental setup

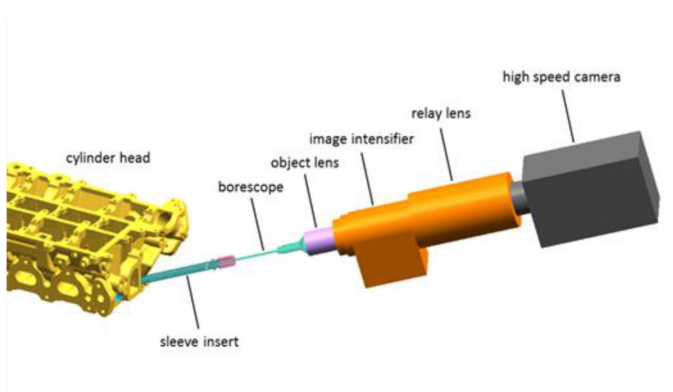


Figure 3. High Speed Imaging Instrumentation

The bracket of the camera has to be specially designed to have a certain stiffness to prevent excessive vibration induced by engine firing or valve train. [Figure 4](#) shows camera bracket modal analysis and modal shapes.

Mode 1: 60.3 Hz

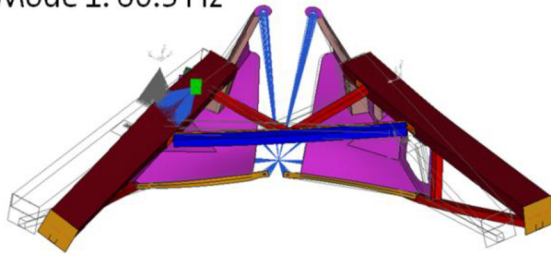


Figure 4. High Speed Camera Bracket

The bracket was tuned to have a minimum natural frequency of 60 Hz of the mode which allowed the survival of optics and camera. [Table 3](#) shows detailed specifications of the optics used in this testing

Table 3. Details of the Optical Instrumentation.

Combustion chamber window	Material	Quartz glass
	Form	hemispherical
Boroscope	Outer diameter	7 mm
	Observation cone	67°
	Viewing direction	30°
	Cooling	air or water cooled
Lighting	Outer diameter of light dispersing stick	7 mm
	Light field cone	60°
	Lighting direction	straight
	Cold light source	150 W
Image Intensifier	Spectral response	200 - 650 nm
	Output diameter	25 mm
	Resolution on output (with 1:1 relay lens)	20 lp/mm
High Speed Camera	Pixel resolution	512 x 512
	Color	no
	Acquisition rate	11500 fps
	Crank angle resolution	1.04 °CA @ 2000 rpm

The data was acquired in the camera buffers and synchronized with engine crank shaft position using a shaft encoder. Pressure data was recorded in the AVL indicom system. The pressure data and the video was synced using special scripts in MATLAB since the pressure data has to be interpreted at the same location where images were acquired. The video and cylinder pressure were collected based on a cylinder pressure pre-trigger criteria. The Post Processing for high speed data acquisition and the synchronization of the cylinder pressure data has been performed using special scripts in MATLAB.

[Figure 5](#) shows the cylinder heads and the two holes for optical access. In the upper left corner of the image is a bar graph of the heat release associated with that crank angle, and in the upper right corner, cylinder pressure data which has been interpreted to match the exact location of the crank shaft position at which the video data was acquired. The image on the left lower corner has been extracted from the high speed video where consecutive images were acquired during the Pre-ignition event. Old Pre-ignition Data in this paper will be shown for two consecutive cycles, cycle 25 and cycle 26. Cycle 25 is a normal combustion event while cycle 26 is a pre-ignition event with cylinder pressure reaching up to 300 bar. [Figure 5](#) shows 16 consecutive images acquired from cycles 25 and 26. To set the stage for the analysis, we will use 0° crank shaft position from top dead center firing. The spark in this engine is at 5.5° after top dead center (ATDC).

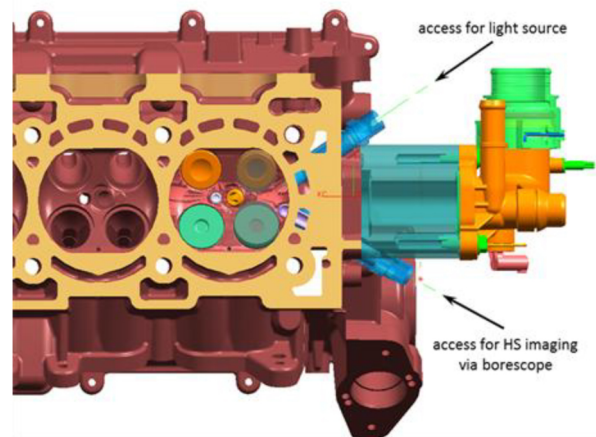


Figure 5. Optical Access Holes

[Figure 6](#) shows a total of 16 consecutive images that has been processed and assembled from single frames as shown in [Figure 7 - Pre-Ignition Event \(single Frame\)](#) starting at -2.8° BTDC for cycle 26 and cycle 25. At -2.8° BTDC, one can see that the heat release in both cycles is very similar. Each consecutive cycle after that point shows that the Pre-ignition cycle starts having more positive heat release in the combustion chambers compared to the normal cycle. At close to top dead center, one can see that in cycle 26 the heat

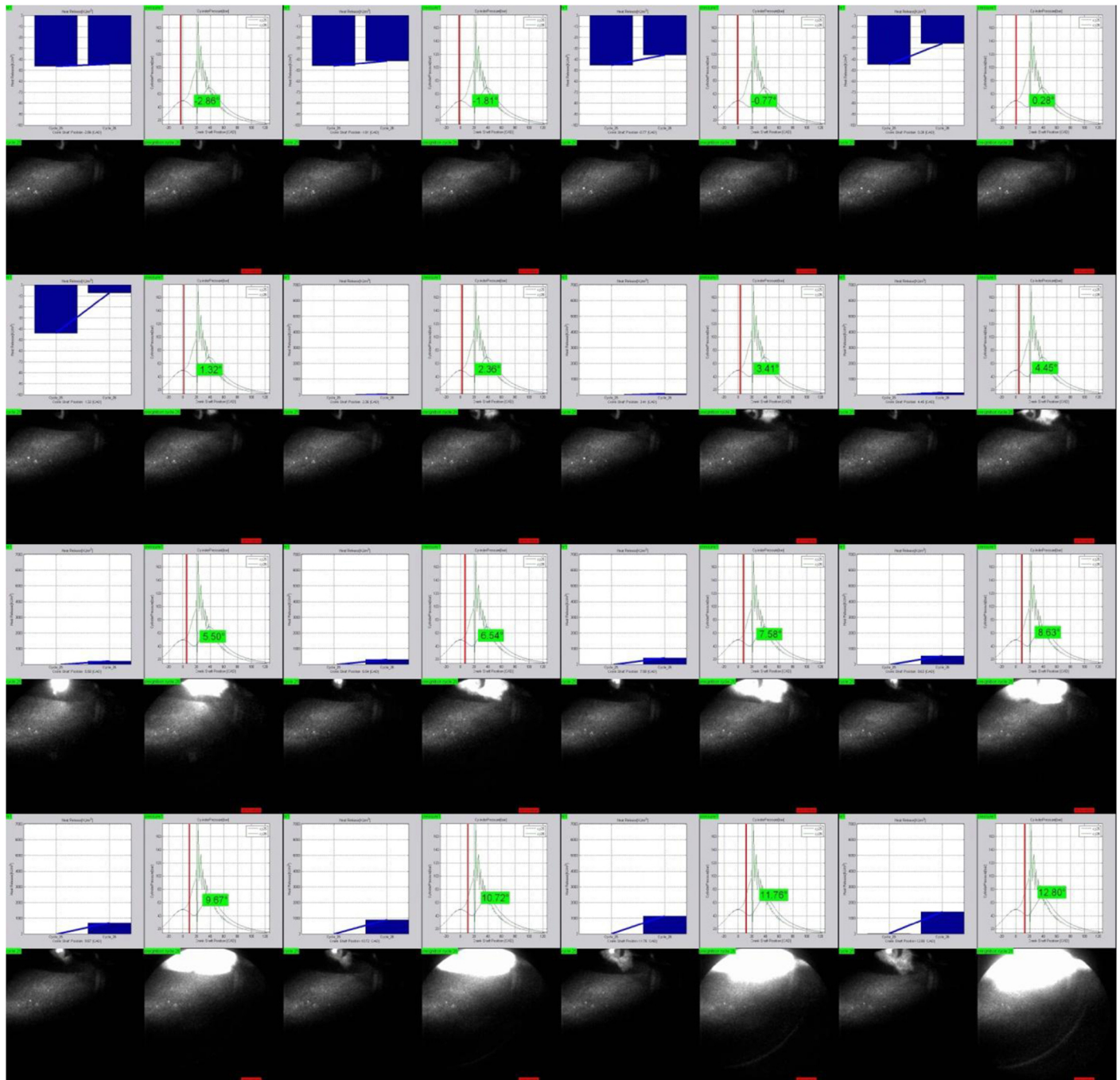


Figure 6. Consecutive High Speed Images of Pre-ignition

release is becoming positive which is evidenced by the pressure traces. In addition, a flash of light to the right of the spark plug coming in from the top of the combustion chamber starts becoming visible. One has to observe at this point that the spark still has 5.5° to take place, so the ignition in the combustion chamber is purely a result of self ignition (auto ignition). At this point it is very obvious from the pressure trace and the heat release that ignition already started and the flame propagation is underway in the pre-ignition cycle. At 5.5 degrees one can see the flash of the spark plug but at this

point this has very little effect on the pressure rise of the pre-ignition cycle since the fast rate of heat release is overpowering the relatively small amount of energy coming from the spark. The flame front in a Pre-ignition cycle will engulf the combustion chamber completely in about 22° CA while it takes twice the amount of time in the normal cycle. This is further evidence of the nature of fast combustion during the Pre-ignition cycle. In the pursuit of creating more quantitative data on the high speed video, MATLAB was used to create a colors contour plot as shown in Figure 8. The

x axis of this plot represents the crank angle at which the data was acquired and the Y axis shows the spatial location of the combustion chamber (pixel location). The data was processed by averaging the pixels in one direction to have one image (frame) collapsed to one column of pixels to represent that crank angle, adding all the crank angles (columns of pixels) would result in a color contour as shown in [Figure 7](#).

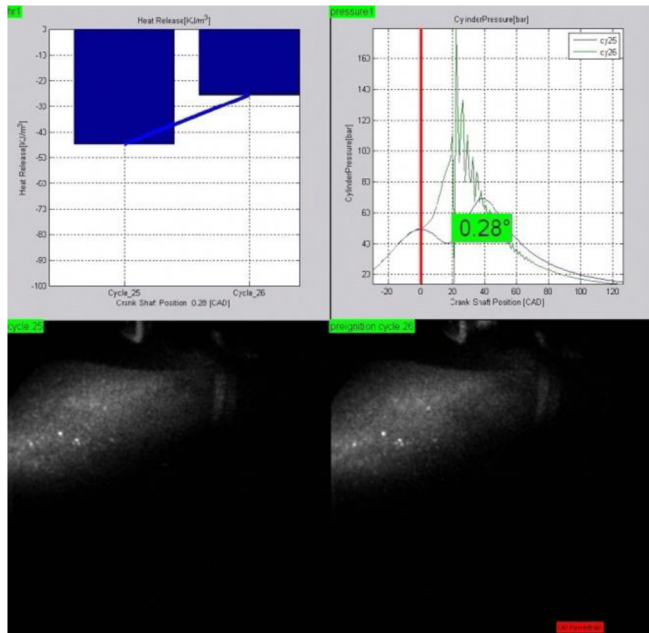


Figure 7. Pre-Ignition Event (single Frame)

Another method of assessing the existence of pre-ignition was found by plotting the average light intensity of each frame of the video as a function of crank angle seen in [Figure 9](#). This video will show the clear pinpointing of the discharging of the spark denoted as a spike in the light intensity. The video shows that the light intensity signal very closely correlates with the pressure data and hence with the rate of heat release. This technique was shown to be useful for measuring in-cylinder soot formation since the diffusion flame correlates very well with this soot formation. Another advantage of this technique is the ability to scan a full length video (40,000 to 80,000 frames) to pinpoint anomalies and areas of interest to simplify the grueling task of post-processing videos. Further analysis of heat release and pressure data in conjunction with the images brought us to the conclusion that this is the cycle that's prone to pre-ignition. The flame propagates very fast (almost twice as fast) and extremely fast combustion is present. Yet from heat release analysis one could see that there is something taking place well before the actual auto-ignition which the instruments were not able to capture. Another point is that the top of the combustion chamber was masked in the video to enable us to establish if the ignition was initiated on the surface or in the volume of the combustion chamber.

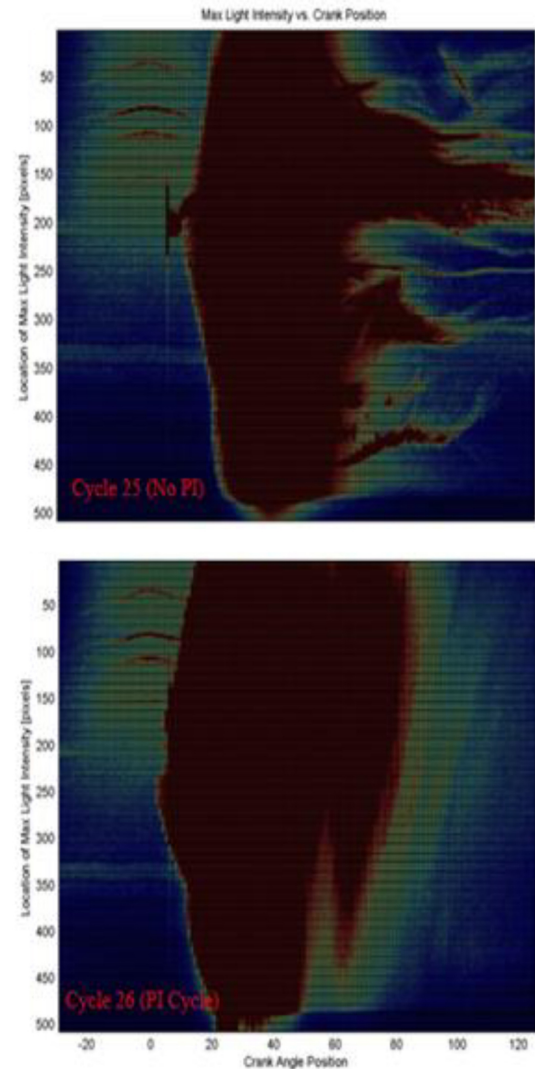


Figure 8. Color Coutuure Plot of Pre-ignition and Normal Cycles

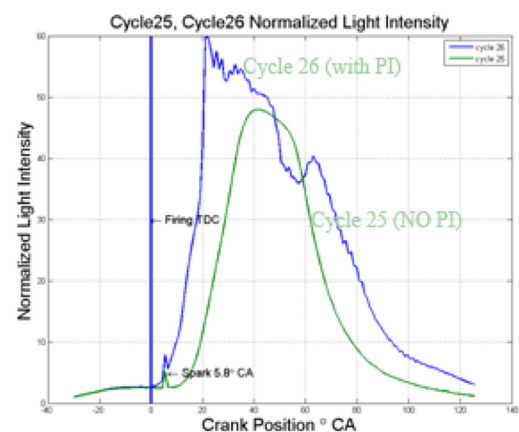


Figure 9. Light Intensity Plot of Pre-ignition and normal cycle

SPATIAL DISTRIBUTION OF PRE-IGNITIONS

The baseline spark plug (NGK HR 6) in cylinder 1 was replaced by a fiber optical spark plug with colder heat range (Bosch HR 5) see [Figure 10](#) - *Optical Spark Plug sensor Positioning*. Further details concerning the test method can be reviewed in [8]. Testing showed pre-ignition frequency was not changed significantly after the installation of the optical spark plug.

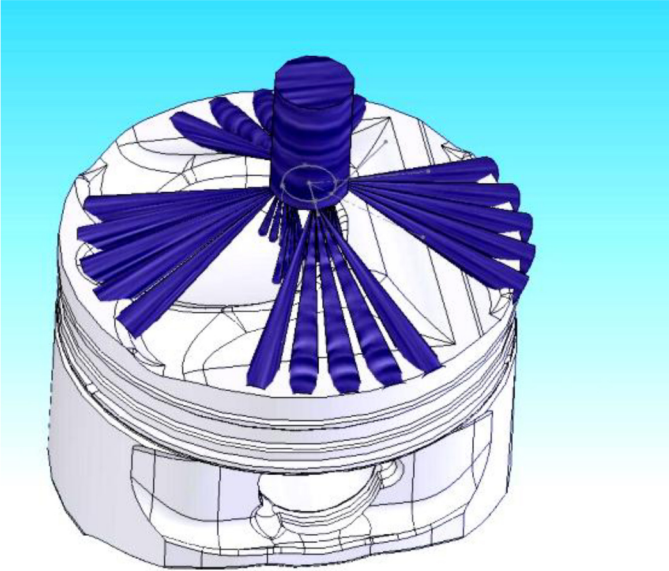


Figure 10. Optical Spark Plug sensor Positioning

In the baseline configuration as shown in [Figure 11](#) *Pre-ignition Distribution using Spark Plug Sensor*, the pre-ignition distribution is fairly random with a fairly high portion of 60% around the injector location. This distribution excludes any consistent hardware hot spot as the root cause for pre-ignitions. When reducing the fuel rail pressure to 90 bar, hence changing the mixture distribution and wall wetting, the focus of the pre-ignition moves from intake to exhaust side (50%). Similarly reducing the spark advance by 4.5 °CA and increasing the boost pressure moves the zone of most pre-ignitions away from the injector to the exhaust side (40 %). Here the hotter exhaust valve might have caused that trend, although spark timing usually appears as less relevant compared to the charge density. Low coolant temperature increases the pre-ignition frequency even further and concentrates the locations of the start of pre-ignitions along the axis of the injector. The colder cylinder liner promotes more fuel wetting and more interaction of oil and fuel

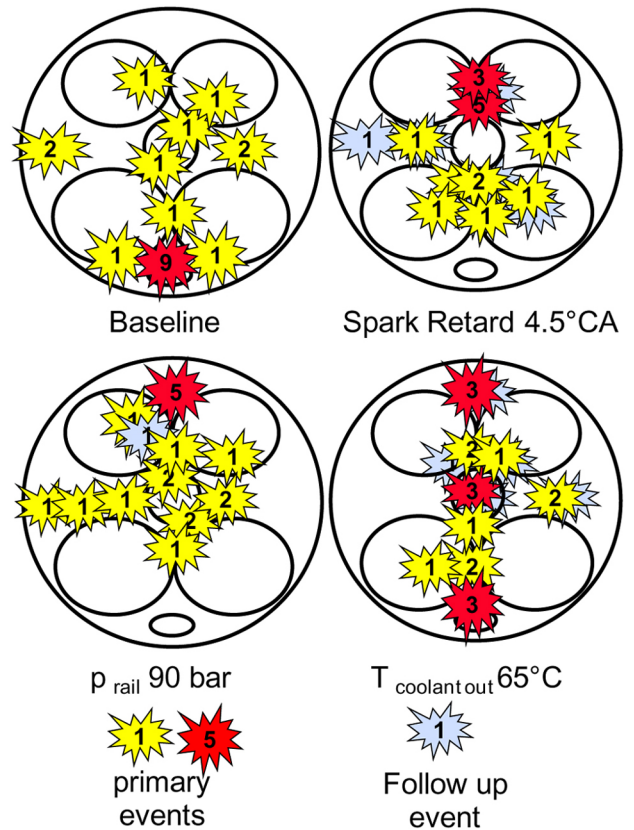


Figure 11. Pre-Ignition Distribution using Spark Plug Sensor

EXPERIMENTAL RESULTS AND TESTING

EFFECT OF BOOST PRESSURE

Increasing the boost pressure by 20kPa (~ 10%) resulted in doubling the frequency of pre-ignition if it is compared to baseline engine with standard calibration and loading conditions as shown in [Figure 12](#).

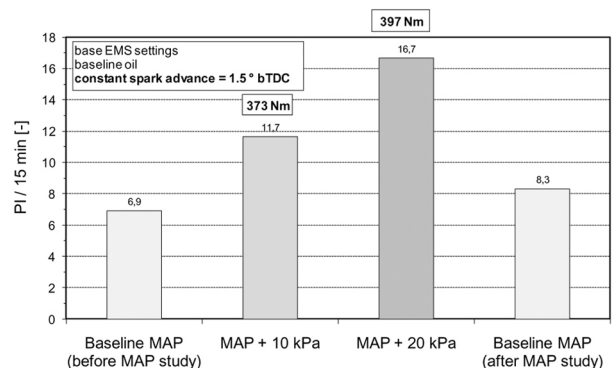


Figure 12. Influence of Manifold Pressure on Pre-ignition

FUEL ENRICHMENT

Enriching the exhaust lambda from a 1.1 to 0.9 for the prototype engine and to 0.8 for the production engine basically eliminated the pre-ignitions as shown in Figure 13. Based on GT power engine simulation this equates to in cylinder lambdas of 0.83 for the 1.6 L and 0.7 for the 2.0 L engine. Both enrichment levels are not desirable at all for any competitive application from fuel consumption point of view. The effect of enrichment is illustrated in Figure 14 and Figure 15. The x-axis shows Lambda, y-axis the normalized value of latent heat of evaporation divided by LHV of the fuel. The color counter represents the bulk temperature of cylinder gas at two points during the compression stroke; one at -10° BTDC and the second at TDC firing. 10° BTDC represents the location that pre-ignition takes place and 0° BTDC represents the spark firing. Figure 14 shows a cooling of $\sim 15^\circ\text{K}$ when lambda is reduced from 1.05 to .85. The effect at TDC is only 10°K for the same change of lambda. One could conclude that enrichment helps in eliminating pre-ignition by reducing the end of compression temperature.

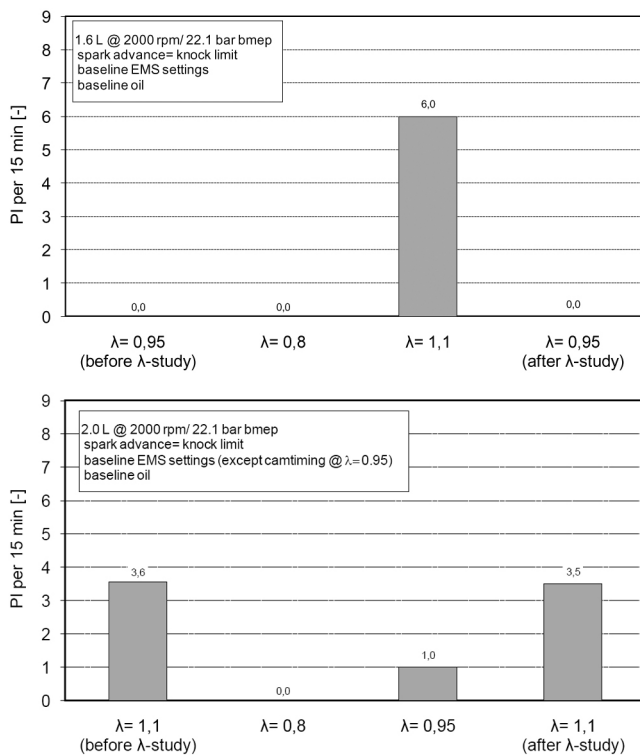


Figure 13. Effect of Fuel Enrichment on Pre-ignition

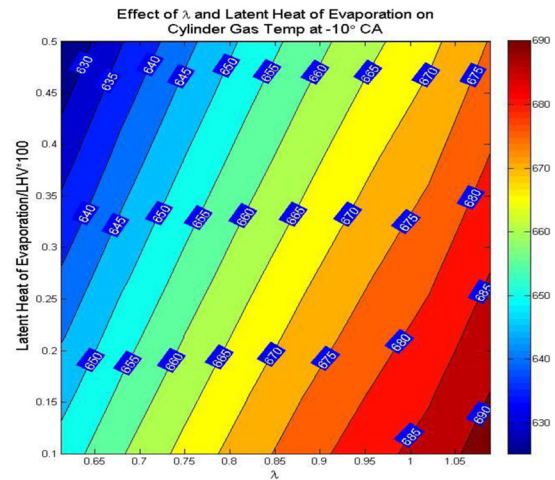


Figure 14. Bulk Gas Temperature -10° BTDC

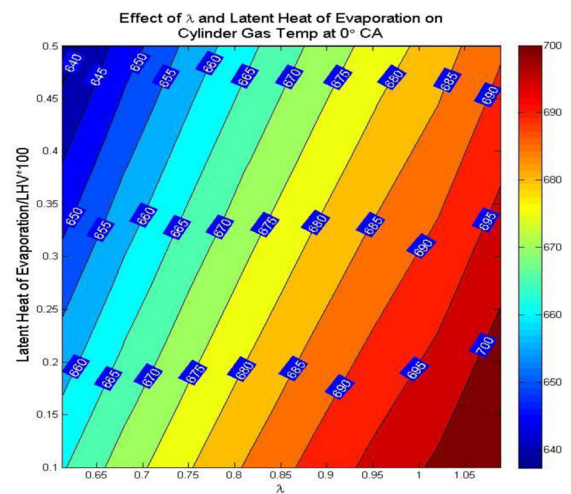


Figure 15. Bulk Gas Temperature 0° BTDC

EFFECT OF FUEL VOLATILITY AND OCTANE NUMBER

Most of the tests published in this paper were conducted with standard RON 95 pump fuel, which was only controlled in regards to the Research Octane Rating (RON 95-96) and the European Fuel Specification EN 228. Fuel delivery was monitored in terms of key characteristic data. A dedicated fuel survey was conducted in order to investigate the main fuel parameters relevant to the pre-ignition phenomena. Target parameters of interest were RON, dry vapour pressure equivalent (DVPE), Ethanol content and volatility, see Table 4 and Figure 16 - Boiling Curve of Fuels. All tests were conducted at their individual knock limits. One could observe that a high research octane rating is beneficial for reducing pre-ignition, see Figure 16. This couldn't be attributed to RON itself but is due to the reduced boost pressure (reduced charge density) necessary for achieving the target torque. 10 Octane numbers are equivalent to about 6° CA of spark