

Spark Ignition and Pre-Chamber Turbulent Jet Ignition Combustion Visualization

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ABSTRACT

Natural gas is a promising alternative fuel as it is affordable, available worldwide, has high knock resistance and low carbon content. This study focuses on the combustion visualization of spark ignition combustion in an optical single cylinder engine using natural gas at several air to fuel ratios and speed-load operating points. In addition, Turbulent Jet Ignition optical images are compared to the baseline spark ignition images at the world-wide mapping point (1500 rev/min, 3.3 bar IMEP_n) in order to provide insight into the relatively unknown phenomenon of Turbulent Jet Ignition combustion. Turbulent Jet Ignition is an advanced spark initiated pre-chamber combustion system for otherwise standard spark ignition engines found in current passenger vehicles. This next generation pre-chamber design simply replaces the spark plug in a conventional spark ignition engine. Turbulent Jet Ignition enables very fast burn rates due to the ignition system producing multiple, widely distributed ignition sites, which consume the main charge rapidly. This high energy ignition results from the partially combusted (reacting) pre-chamber products initiating combustion in the main chamber. The distributed ignition sites enable relatively small flame travel distances enabling short combustion durations and high burn rates. Multiple benefits include extending the knock limit and initiating combustion in very dilute mixtures (excess air and/or EGR), with dilution levels being comparable to other low temperature combustion technologies (HCCI), without the complex control drawbacks.

INTRODUCTION AND BACKGROUND

In recent years there has been renewed interest in lean burn technologies, primarily due to the improvement in fuel efficiency that these technologies can provide [1,2,3]. Lean burn occurs when fuel is burnt in excess air and running an engine in this manner has many advantages over conventional stoichiometric combustion. One advantage of running lean occurs as the introduction of additional air increases the specific heat ratio, which leads to an increase in thermal efficiency. Additionally, lean engine operation reduces pumping losses for a given road load by tending towards throttle-less operation, which can further improve drive-cycle fuel economy. In this case, power output can be somewhat controlled by varying the fuel supplied to the combustion chamber (as in a diesel engine) rather than by throttling, as is done in conventional stoichiometric spark ignition engines. Lean burn technology has not been extensively implemented for several reasons including compromised combustion stability and three-way catalyst incompatibility. In addition, although NO_x emissions decrease at higher λ due to lower combustion temperatures, significant improvements in NO_x emissions can only begin to be seen at the lean limit ($\lambda \sim 1.4$) of conventional spark ignition engines. Moreover, the effectiveness of the three-way catalyst, which is the primary means to control NO_x, HC and CO emissions in conventional spark ignition engines, degrades rapidly at AFR that vary from stoichiometric.

There are also several other disadvantages of lean burn technology that have prevented it from being more widely used. The narrow flammability limits of most fuels make it difficult to run lean while maintaining adequate combustion stability with low misfire rates. Poor combustion stability leads to increased HC and CO emissions due to misfire and partial burning cycles. Partial burning is primarily caused by the slower laminar flame speeds of lean mixtures that severely affect flame kernel growth and hence flame propagation. Furthermore, as the lean limit of combustion is approached, irregular ignition and HC emissions increase while power output substantially decreases. The laminar flame speed of natural gas is lower than that of other hydrocarbon fuels due to its higher activation energy and this effect is most significant under lean conditions [4]. To compensate for the lower laminar burning velocity of lean mixtures, it is necessary to increase the in-cylinder charge motion or charge density to attain a sufficiently rapid burn rate. One method that overcomes the negative effects of lean combustion is ignition enhancement using Turbulent Jet Ignition, which utilizes a pre-chamber combustion initiation system [5,6,7,8]. The Turbulent Jet Ignition concept involves the use of a chemically active, turbulent jet to initiate combustion in lean fuel mixtures. With jet ignition, the combustion of the main charge is reliable over a much broader range of air-fuel ratios since the jet acts as a distributed ignition source. The large number of distributed ignition sites ensures that the flame travel distances are relatively small enabling short combustion durations even in traditionally slow burning lean mixtures [9, 10].

OBJECTIVE

The objective of this research is to examine and analyze natural gas combustion in an optically accessible engine. The specific objectives of this paper are to:

- Examine gaseous spark ignition combustion at several air to fuel ratios and speed-load operating points
- Compare the baseline spark ignition and pre-chamber Turbulent Jet Ignition combustion systems at the world-wide mapping point of 1500 rev/min, 3.3 bar IMEPn (~2.62 bar BMEP)
- Analyze the characteristics of Turbulent Jet Ignition combustion

EXPERIMENTAL SETUP

OPTICAL ENGINE ASSEMBLY

The normally aspirated optical single cylinder research engine used in experiments was assembled collaboratively between Michigan State University (MSU) and MAHLE Powertrain. The bottom end utilized a MSU modified Hatz platform [11] with a Bowditch arrangement allowing optical combustion visualization through a quartz window (58 mm diameter) in the piston crown [12]. The 0.4 liter single

cylinder engine featured a flat top piston crown in a modern pent roof combustion system. The top end assembly consisted of a modified single cylinder version of MAHLE Powertrain's DI-3 cylinder head [13] which incorporated double overhead camshafts and four valves per cylinder. The cylinder head was originally designed to incorporate central spray guided direct injection, however the combustion system was modified to accept PFI and hence allow the pre-chamber jet igniter to be packaged centrally in the combustion system. Standard production intake and exhaust camshaft profiles were utilized. The intake and exhaust primary runner tracts were kept to standard lengths, which were both coupled to individual large plenum volumes in order to minimize the pressure fluctuations associated with the single firing event. [Table 1](#) lists the optical engine specifications while [Figure 1](#) displays a computer design image section view through the assembly. [Figures 2](#) and [3](#) display the test rig and optical view of the combustion chamber, with the 58 mm diameter quartz window in the piston crown enabling a high portion of the combustion chamber to be viewed in the 83 mm bore.

Table 1. Test engine specifications

TYPE	Normally aspirated single cylinder, 4 stroke, liquid-cooled, aluminum head/block
OPTICAL IMAGING	Bowditch piston with 58 mm diameter quartz window in piston crown
DISPLACEMENT	0.4 liter
BORE x STROKE	83 x 73.9 mm
COMPRESSION RATIO	10:1
COMBUSTION CHAMBER	Pent roof, central ignition
VALVE ACTUATION	4-valve, DOHC
INTAKE	500 mm runner length, 12 liter plenum volume
EXHAUST	300 mm runner length, 12 liter plenum volume
FUEL DELIVERY	<i>Spark Ignition:</i> PFI <i>Jet Ignition:</i> main chamber PFI pre-chamber DI
SPARK INITIATION	Inductive ignition system (35 mJ), ionization detection

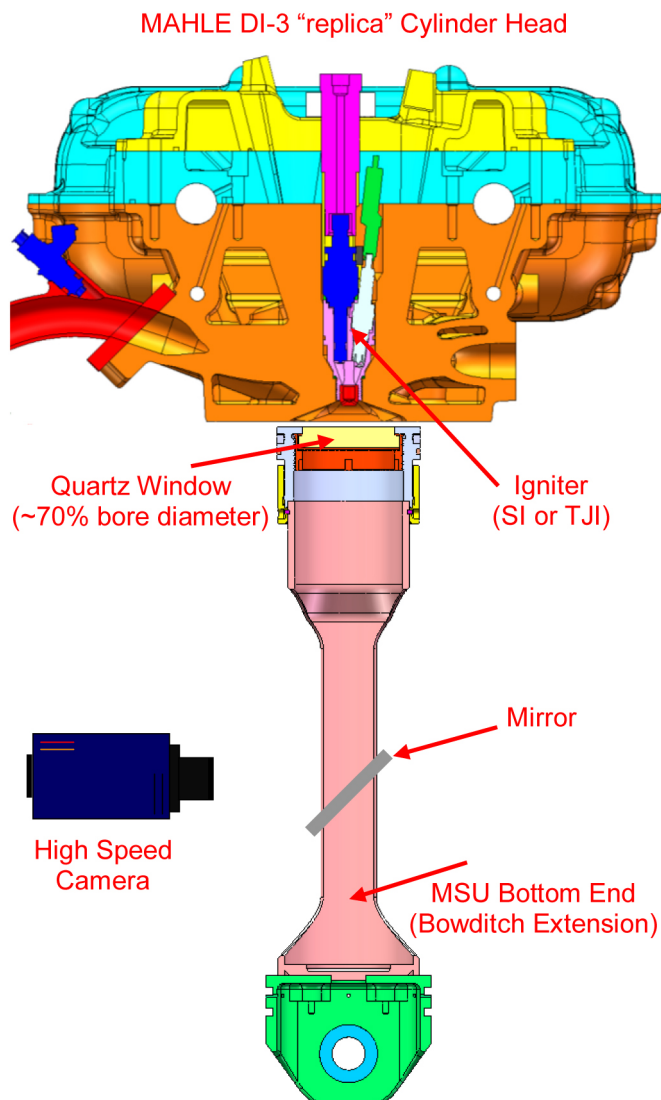


Figure 1. Computer design image section view through the assembly.

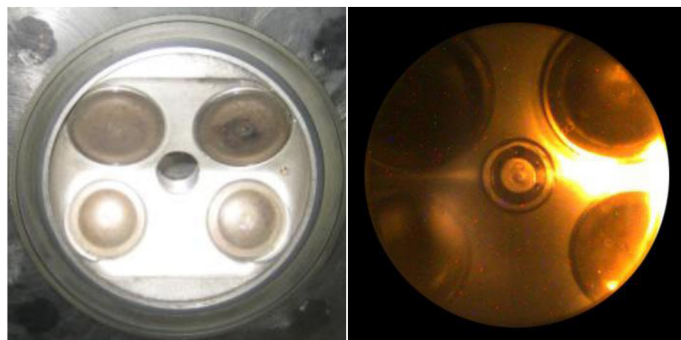
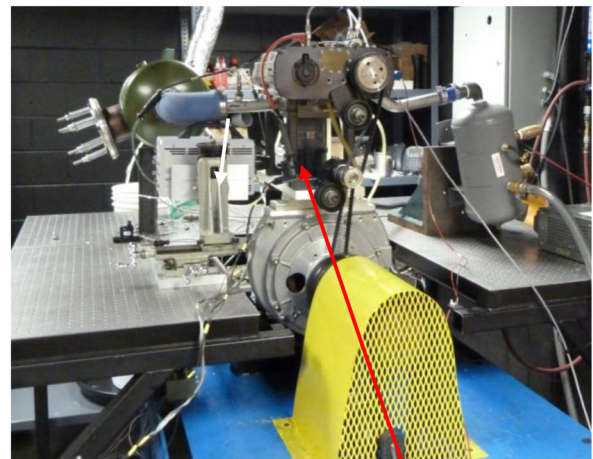


Figure 2. View of combustion chamber prior to installation (Left) and through the Bowditch extended piston with 58 mm diameter quartz insert (Right).



High Speed Camera
MSU Bottom End (Bowditch Extension)
MAHLE DI-3 "replica" Cylinder Head
Port Fuel Injector

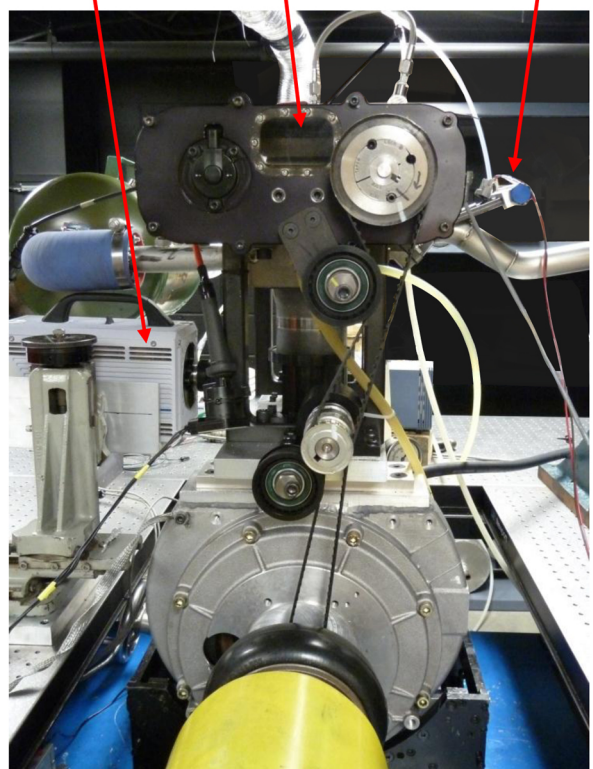


Figure 3. Experimental engine setup.

Due to the optical arrangement of the test rig, engine firing experiments were limited to 50 cycle sequences due to optical limitations and combustion image quality degradation. To ensure consistency, the cooling and lubrication (oil pressure and scavenge) systems were de-coupled from the crankshaft and externally operated. This allowed the barrel and cylinder head to be pre-heated to 90°C prior to firing experiments and

ensured lubrication integrity under the increased inertia loading from the Bowditch extended piston design.

Table 2. Gaseous fuel properties compared to gasoline
[14,15,16,17,18].

	Natural Gas (~Methane CH ₄)	Propane (C ₃ H ₈)	Gasoline
Molecular Weight	~18	44.10	~110
Energy Density (MJ/kg)	45	46.4	44.0
Energy Density (kJ/mol)	810	2002	4840
Stoichiometric AFR (by mass)	14.5	15.7	14.6
Stoichiometric AFR (by volume)	9.04	23.9	57.7
Energy in 1 mol Stoichiometric Fuel-Air Mixture	80.8	80.4	82.5
Gas Constant – R (kJ/kg K)	0.518	0.187	–
Density (kg/m ³) 0 °C and 1 atm	0.79	2.0	735
Laminar Flame Speed (m/s)	0.39	0.45	0.4
LFL (%) (by vol)	5.6	2.2	1.4
UFL (%) (by vol)	15	9.5	7.6
Lambda at LFL	1.9	1.9	1.2
Lambda at UFL	0.63	0.40	0.21
Auto Ignition Temperature (°C)	632	500	530
Min Ignition Energy (mJ)	0.33	0.30	0.8
Fuel Octane Rating (RON)	120	112	92-98
Fuel Octane Rating (MON)	120	97	80-90

The test rig measured gaseous fuel consumption on a mass basis directly using two inline thermal mass flow meters to ensure fuel consumption data integrity. Increased fuel rail volumes minimized the fuel flow pulsations experienced at the flow meters as a result of the single cylinder fuel injection configuration. All exhaust lambda (λ) values presented in this paper were obtained using a heated wideband oxygen sensor fitted in the exhaust runner. Hence for the jet ignition system, the exhaust lambda is inclusive of both the main and pre-chamber fuel. A flush mounted Kistler 6052A pressure transducer was used to measure in-cylinder pressure, with TDC alignment confirmed by interpreting motored log pressure-volume data. Additionally, in-cylinder pressure acquisition (0.5° CA sampling) and analysis was completed using A&D's CAS system, enabling combustion parameters

to be compared. The combustion imaging was accomplished with a non-intensified high speed digital video camera (Photron, Model Fastcam APX RS) which was triggered by the rising edge of the spark signal. The camera was set to record at 10,000 images per second with a resolution of 512 pixels by 512 pixels to cover a spatial imaging area of approximately 120 mm by 120 mm. Table 2 compares the gaseous fuel properties of both natural gas and propane to gasoline in order for the reader to gauge fuel differences between liquid and gaseous applications. Propane is included in the table for reference as it has previously been used successfully in the jet ignition system [5], and is a widely available gaseous fuel.

FUEL INJECTION AND IGNITION SYSTEMS

The optical test engine was designed to be equipped with either a conventional spark plug or a pre-chamber Turbulent Jet Igniter [5, 6, 19, 20] with the same cylinder head used for both systems. Figure 4 displays a computer generated image of the Jet Igniter which simply replaces the 14 mm spark plug used in the conventional spark ignition combustion system. Figure 5 displays further images of both ignition systems installed in the test engine's cylinder head. Extensive metal engine research has been completed with this specific pre-chamber Turbulent Jet Igniter, with experimental results of the system published in the literature [6, 19,20,21].

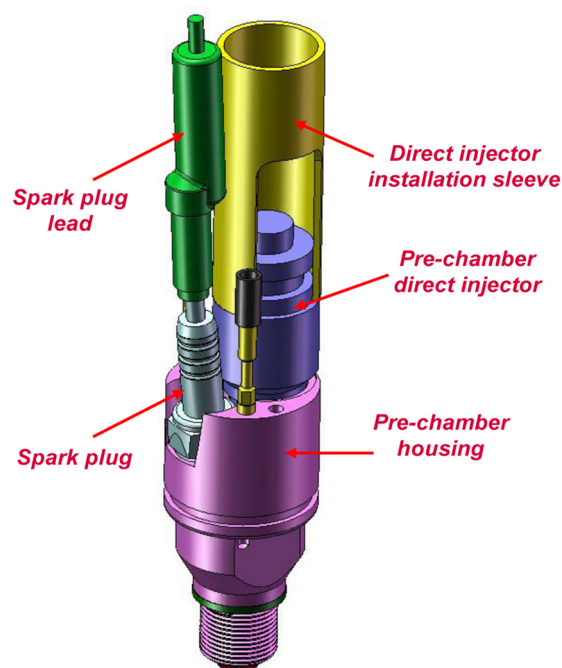


Figure 4. Design image of the Turbulent Jet Igniter, designed to replace the spark plug in a contemporary spark ignition combustion system.

Combustion initiation for both spark ignition and jet ignition combustion systems was achieved using a spark plug and an inductive ignition system (35 mJ). The inductive ignition system also incorporated a MSU developed ionization detection system for further combustion diagnostics, with the ionization detection electronics integrated into the ignition coil spark plug assembly.

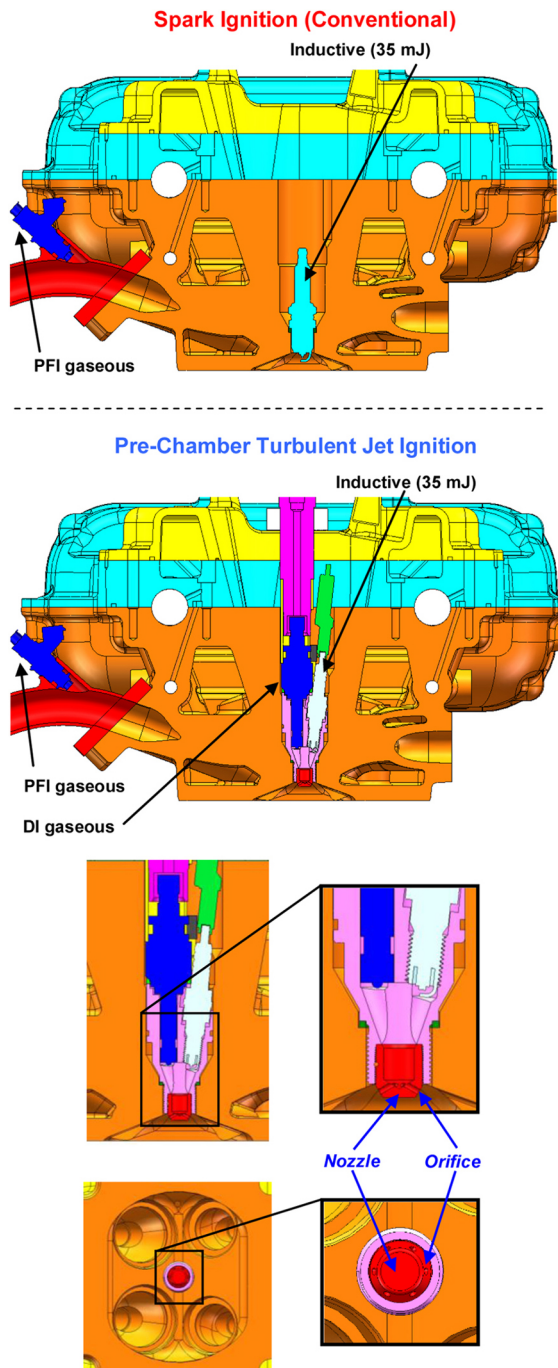


Figure 5. Spark ignition and Pre-Chamber Turbulent Jet Ignition combustion systems.

Due to the varying ignition systems and the need to supply the pre-chamber with auxiliary fueling, fuel delivery systems varied between spark ignition and jet ignition systems due to the addition of the pre-chamber fuel event. Both ignition systems utilized a Bosch gaseous NGI-2 PFI injector which port injected gaseous fuel at relatively low fuel pressure. However, jet ignition tests also utilized small amounts of auxiliary fuel when running excess air dilution which allowed adequate pre-chamber combustion. Hence for jet ignition tests, the bulk of fuel energy was port fuel injected into the main chamber while the additional auxiliary fuel was directly injected into the pre-chamber. To minimize the complexity of the fuel system, the jet ignition system PFI and DI injectors shared a common fuel rail and hence single gaseous fuel.

BASELINE RESULTS

Extensive metal engine experiments have previously been completed with the jet ignition combustion system described in this paper [5, 6, 19,20,21,22]. Experiments were completed in a similar modern 4 valve single cylinder test engine, with the addition of wide angle camshaft phasers to control both the intake and exhaust valve events. A summary of the part load fuel economy and NO_x emission benefits when comparing stoichiometric spark ignition (optimized valve events with camshaft phasers) and jet ignition combustion in the same test engine is given in Figure 6 [6]. Figure 7 shows a comparison of spark ignition and jet ignition combustion at the world wide mapping point of 1500 rev/min, 3.3 bar IMEP_n (~2.62 bar BMEP) [20]. Obvious is the significant lean limit extension with jet ignition combustion, with stable engine operation near $\lambda = 2$ compared to only $\lambda = 1.4$ for conventional spark ignition combustion. However, the rich limit at this part load operating condition is slightly reduced with jet ignition due to the increased residual hang-up in the pre-chamber cavity, which negatively affects the pre-chamber combustion. This effect is overcome at high load conditions as the residual fraction is significantly reduced.

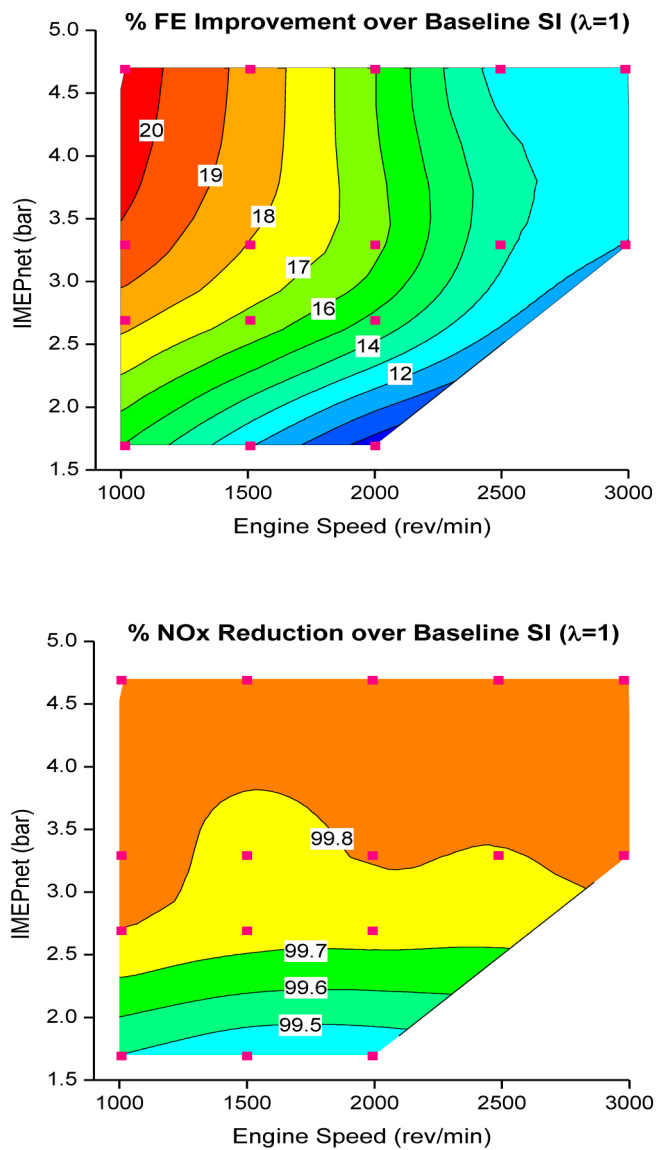


Figure 6. Fuel economy and engine out NOx emission comparisons of jet ignition and stoichiometric spark ignition combustion [6].

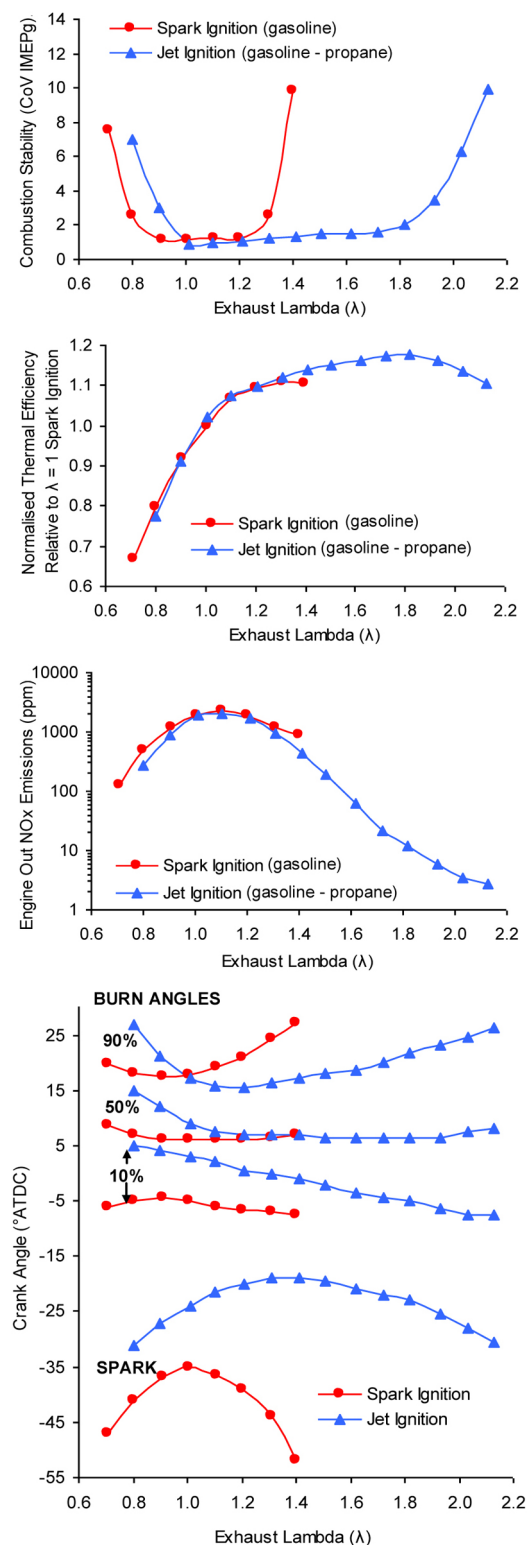


Figure 7. Spark ignition (gasoline) and Turbulent Jet Ignition (dual fuel: gasoline for main chamber, gaseous propane for pre-chamber) combustion systems with varying dilution. 1500 rev/min, 3.3 bar IMEPn (~2.62 bar BMEP) [20].

EXPERIMENTAL RESULTS

Spark ignition combustion with natural gas was evaluated over speeds of 1000 and 1500 rev/min, at loads of 1.7 and 3.3 bar IMEPn and at several air fuel ratios varying from stoichiometric to the lean limit. [Table 2](#) lists the operating sites evaluated. Turbulent Jet Ignition was also evaluated at the world wide mapping point of 1500 rev/min, 3.3 bar IMEPn at $\lambda=1.8$. The combustion visualization images presented below provide useful insight into characteristics such as flame sizes, shapes and appearances during the combustion process. However, it is noted that the images are two-dimensional representations of the three-dimensional flame development. The images shown in this work were reduced from their original size of 512×512 pixels to 325×325 pixels to improve visibility by removing the dark pixels that were outside of the cylinder. The physical size of each pixel was approximately 0.18 mm². The images are enhanced so that the early flame development is clearly visible to the reader for comparison.

Table 2. Experimental Test Conditions

Speed	1000 and 1500 rev/min
Load (IMEPnet)	1.7 and 3.3 bar
Lambda (λ)	1, 1.4, 1.5, 1.8

SPARK IGNITION NATURAL GAS COMBUSTION WITH VARYING LOAD AND SPEED

[Figure 8](#) shows the crank angle resolved images of the combustion and flame propagation for stoichiometric conditions at 1.8 and 3.3 bar IMEPn at 1000 rev/min and also at 3.3 bar IMEPn at 1500 rev/min in order to show the effects of both load and speed. At these conditions, the combustion reactions produce enough energy to excite and ionize gas molecules in the flame giving it the blue color characteristic of sootless hydrocarbon flames. The color is produced due to the chemiluminescence of radical species such as CH, CH₂O and C₂ which emit at wavelengths near the blue portion of the visual spectrum [23]. For all three cases it can be clearly seen that the flame is initiated near the spark plug region and propagates outward filling the majority of the visible area (58 mm diameter) by ~24 °CA after ignition. The differences in the flame propagation rates between the three cases are highlighted in [Figure 9](#), which shows the average flame radius versus crank angle. [Figure 9](#) indicates that the initial flame propagation for both the 3.3 bar IMEPn cases is similar but that the flame propagation for the slower speed (1000 rev/min 3.3 bar case) is more rapid after about 5 °CA after ignition. This indicates that the engine speed has an effect on the rate of flame propagation for the speeds examined here. The slower flame growth per crank angle degree is expected at 1500 rev/min relative to 1000 rev/min due to the shorter

time available per crank angle degree. The rate of flame propagation for the lower load 1000 rev/min 1.8 bar IMEPn case proceeds at a slower rate than either of the other cases, due to the decreased charge density. The crank-angle resolved images of the combustion and flame propagation together with the cylinder pressure traces for the three stoichiometric cases of 1.8 and 3.3 bar IMEPn at 1000 rev/min and 3.3 bar IMEPn at 1500 rev/min are shown in [Figures A1, A2, A3](#) in the [Appendix](#). In each figure the CA10, CA50 and CA90 images are highlighted, indicating the 10%, 50% and 90% burn locations respectively.

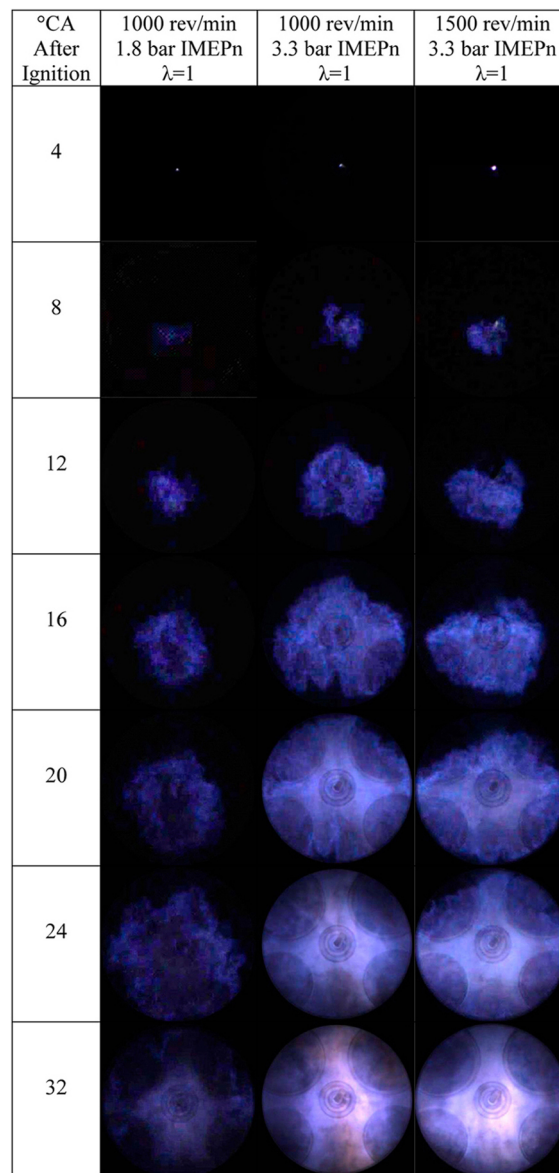


Figure 8. Varying load and speed crank angle resolved images of in-cylinder combustion through the quartz piston insert. 1.8 and 3.3 bar IMEPn at 1000 rev/min and 3.3 bar IMEPn at 1500 rev/min. Stoichiometric spark ignition combustion using natural gas fuel.

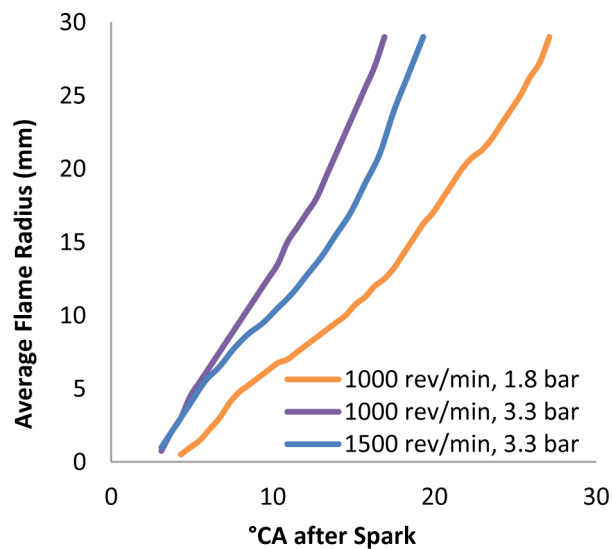


Figure 9. Varying load and speed effect on the average flame radius versus CA° after ignition. 1.8 and 3.3 bar IMEPn at 1000 rev/min and 3.3 bar IMEPn at 1500 rev/min. Stoichiometric spark ignition combustion using natural gas fuel.

EXCESS AIR DILUTION EFFECT OF SPARK IGNITION AND PRE-CHAMBER TURBULENT JET IGNITION

Figure 10 shows the crank angle resolved natural luminosity images for stoichiometric, $\lambda=1.4$ and $\lambda=1.5$ spark ignition and $\lambda=1.8$ Turbulent Jet Ignition at 1500 rev/min and 3.3 bar IMEPn. Initial flame propagation is more luminous and more rapid for the $\lambda=1$ case relative to the lean SI cases, with the luminosity greatly decreasing for the $\lambda=1.5$ case, which is expected as a strong correlation has been shown to exist between the chemiluminescence light intensity and the rate of heat release [24]. The Turbulent Jet Ignition combustion is visible at a later crank angle due to it initiating inside the pre-chamber, which is hidden from view. The jets begin to appear in the combustion chamber at $\sim 11^\circ$ CA after ignition and then rapidly consume the main charge. In addition with Turbulent Jet Ignition, although the mixture is much leaner at $\lambda=1.8$ than the two lean SI cases, the flame is much more luminous and also has a more intense blue color. It is hypothesized that this is due to the higher rate of heat release and is an indicator of the more stable combustion in this case. In the leaner spark ignition mixtures, poorer combustion is visible as indicated by the brown luminosity present in the later stages of combustion. The intensity of the blue emissions from pre-mixed hydrocarbon flames gives an indication of the radical and atom concentrations in the reaction zone [25] and therefore it can be concluded that there is a much higher radical concentration in the $\lambda=1$ and

turbulent jet ignition cases relative to the lean spark ignition cases. Furthermore, for the spark ignition cases, it can be seen that with the leaner mixtures it becomes more difficult to initiate and stabilize the flame kernel, which in turn increases the burn duration. This is further highlighted in Figure 11, which shows the observed flame radius versus crank angle for each of the three spark ignition conditions. The crank-angle resolved images of the combustion and flame propagation together with the cylinder pressure traces for the $\lambda=1$, $\lambda=1.4$, $\lambda=1.5$ spark ignition cases and the $\lambda=1.8$ Turbulent Jet Ignition cases at 3.3 bar IMEPn at 1500 rev/min are shown in Figures A3, A4, A5, A6 in the Appendix. In each figure the CA10, CA50 and CA90 images are highlighted, giving a reference for the 10%, 50% and 90% burn locations respectively. Table 3 additionally highlights the spark timing, combustion stability and pressure rise rates for both combustion systems. Evident across the short 50 cycle sequence is the improved CoV IMEPg with jet ignition combustion, even at the ultra lean condition of 1.8, which is associated with the distributed ignition sites in the main chamber. The distributed ignition sites have also resulted in significant reductions in spark timing for the jet ignition system when compared to the lean spark ignition operating conditions. It is noted that the jet ignition pressure rise rates can also be controlled to typical stoichiometric spark ignition levels using an ultra lean main chamber. However for equivalent stoichiometric conditions, jet ignition combustion typically has pressure rise rates which are approximately 50% higher when compared to spark ignition combustion.

In order to gain further understanding of the Turbulent Jet Ignition process shown in Figure 10, an analysis of the jet penetration and variability was completed and is shown graphically in Figure 12. This figure displays the jet characteristics (penetration and variability) versus crank angle for each of the jet igniter's six jets averaged over a 10 cycle sequence. It can be seen that all six jets penetrate at approximately the same rate with the jets emerging from the pre-chamber orifices at $\sim 10.8^\circ$ CA after ignition and reaching the edge of the viewable area (58 mm diameter) at approximately 14.4° CA after ignition. From this information, the average jet velocity can be calculated to be near 60 m/s, which is much faster than the spark ignition flame propagation rate even for stoichiometric mixtures.

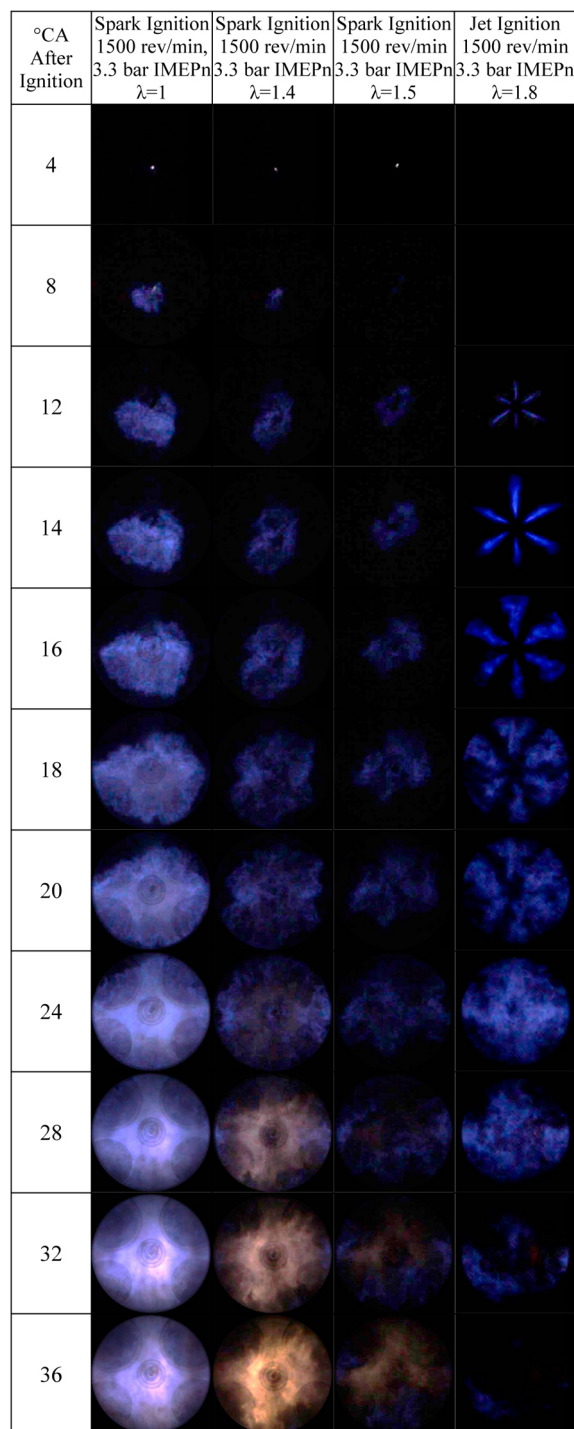


Figure 10. Varying excess air dilution crank angle resolved images of in-cylinder combustion through the quartz piston insert for both spark ignition ($\lambda=1$, $\lambda=1.4$ and $\lambda=1.5$) and jet ignition ($\lambda=1.8$) combustion. 1500 rev/min, 3.3 bar IMEPn.

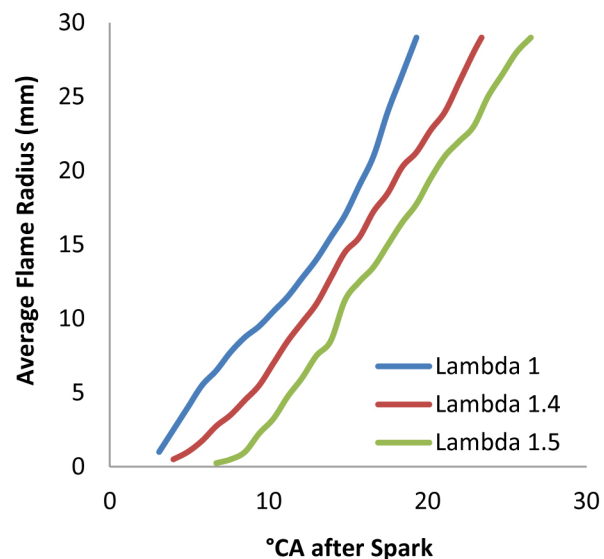


Figure 11. Varying excess air dilution effects on the average flame radius versus CA° after ignition for spark ignition combustion ($\lambda=1$, $\lambda=1.4$ and $\lambda=1.5$). 1500 rev/min, 3.3 bar IMEPn.

Table 3. Spark ignition and Turbulent Jet Ignition comparison.

Spark Ignition (Baseline)					Jet Ignition
Engine Speed (rev/min)	1500				1500
Load (IMEPn) (bar)	3.3				3.3
Lambda (λ)	1	1.2	1.4	1.5	1.8
Spark Timing (deg. BTDC)	18	20	24	25	17
CoV IMEPg (%)	2.3	2.5	2.5	4	1.0
CA50 (deg. ATDC)	8	7	7	8	7
Pressure Rise Rate (bar/deg.)	1.2	1.1	1.0	0.8	1.1

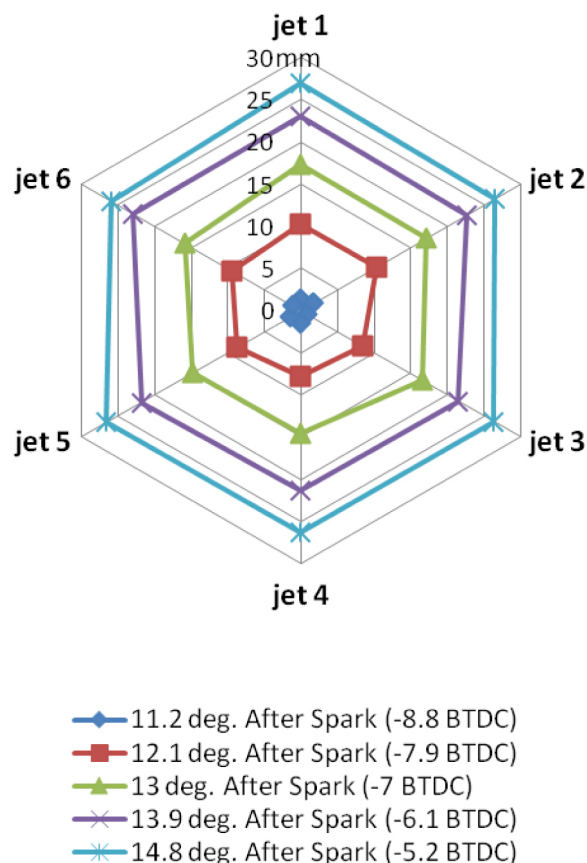


Figure 12. Ten-cycle average of Turbulent Jet Ignition jet penetration versus crank angle.

SUMMARY/CONCLUSIONS

Spark ignition natural gas combustion was tested across varying speed and load points at several air to fuel ratios varying from stoichiometric to lean in a single cylinder optical engine. In addition, a pre-chamber Turbulent Jet Ignition combustion process was also examined. Experimental results based on piston crown optical studies indicated that the rate of flame propagation was more dependent on load when compared to engine speed across the test points for stoichiometric spark ignition operation. Additionally, with lean spark ignition combustion, optical results showed significant reductions in flame propagation rates as the excess air dilution was increased across all test points. The reduction in flame propagation for lean spark ignition combustion was shown to be heavily reliant on the initial flame kernel development and resulting poorer combustion. This was highlighted by the reduced blue chemiluminescence which results from ionization in the reaction zone, with high levels of brown luminosity observed in the later stages of the lean combustion. However, with the enhanced ignition provided by the pre-chamber Turbulent Jet Ignition system, the combustion produced only visible blue chemiluminescence even at $\lambda=1.8$ due to the very rapid

combustion produced with this ignition enhancement process in lean mixtures.

Natural gas is a promising alternative fuel for current and future powertrain applications as it is abundant worldwide, with large reserves existing in the United States. However, in order for society to meet legislative and environmental CO₂ concerns, future powertrains need to be developed to enable robust lean engine operation. Although natural gas can provide a reduction in CO₂ emissions due to the lower carbon content of the fuel, conventional homogenous lean spark ignition combustion with natural gas has been shown to be problematic due to slow burn rates and high HC and CO emissions. Hence, coupling gaseous fuels such as natural gas with pre-chamber ignition systems, like the Turbulent Jet Ignition system presented in this paper, may be one way forward to enable future powertrains to achieve ultra-lean operation for both significant fuel economy and emission benefits.

ACKNOWLEDGMENTS

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DEFINITIONS/ABBREVIATIONS

BMEP

Break Mean Effective Pressure

CA

Crank Angle

CA10

Crank Angle at 10% Mass Fraction Burned

CA50

Crank Angle at 50% Mass Fraction Burned

CA₉₀
Crank Angle at 90% Mass Fraction Burned

TDC
Top Dead Center

CH
Methylidyne

TJI
Turbulent Jet Ignition

CH₂O
Formaldehyde

λ
Relative air/fuel ratio (actual AFR/stoichiometric AFR)

CO
Carbon Monoxide

CoV
Coefficient of Variation

DI
Direct Injection

EGR
Exhaust Gas Recirculation

HC
Hydrocarbon

HCCI
Homogeneous Charge Compression Ignition

IMEP_g
Gross Indicated Mean Effective Pressure

IMEP_n
Net Indicated Mean Effective Pressure

MSU
Michigan State University

NO_x
Oxides of Nitrogen

PFI
Port Fuel Injection

SI
Spark Ignition

APPENDIX

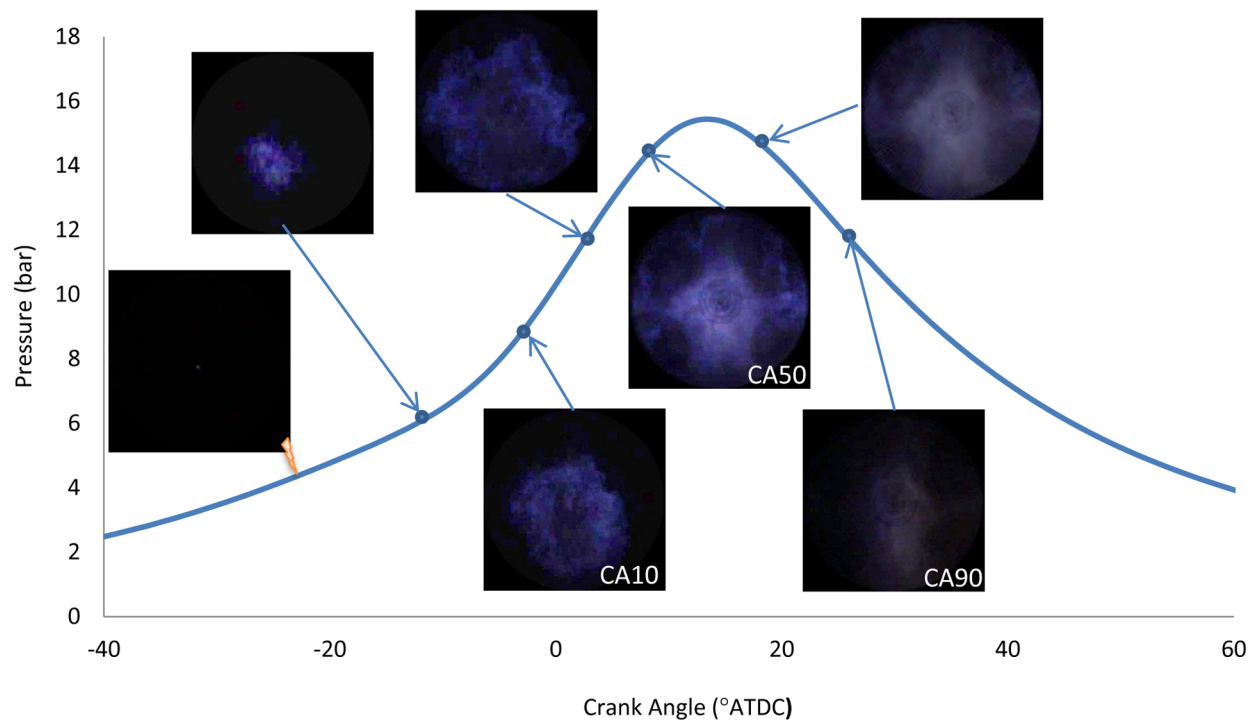


Figure A1. Spark ignition combustion pressure trace and images with CA10, CA50 and CA90 noted. $\lambda=1$, 1000 rev/min, 1.8 bar IMEPn, natural gas fuel.

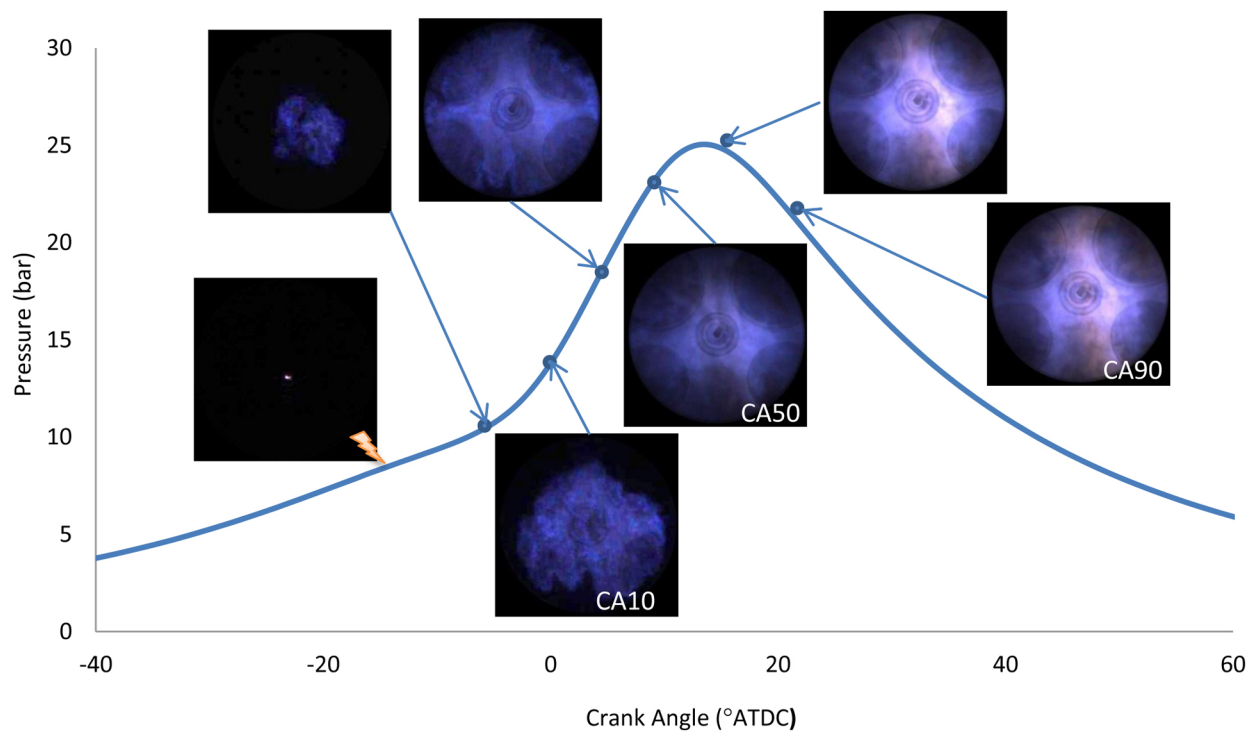


Figure A2. Spark ignition combustion pressure trace and images with CA10, CA50 and CA90 noted. $\lambda=1$, 1000 rev/min, 3.3 bar IMEPn, natural gas fuel.

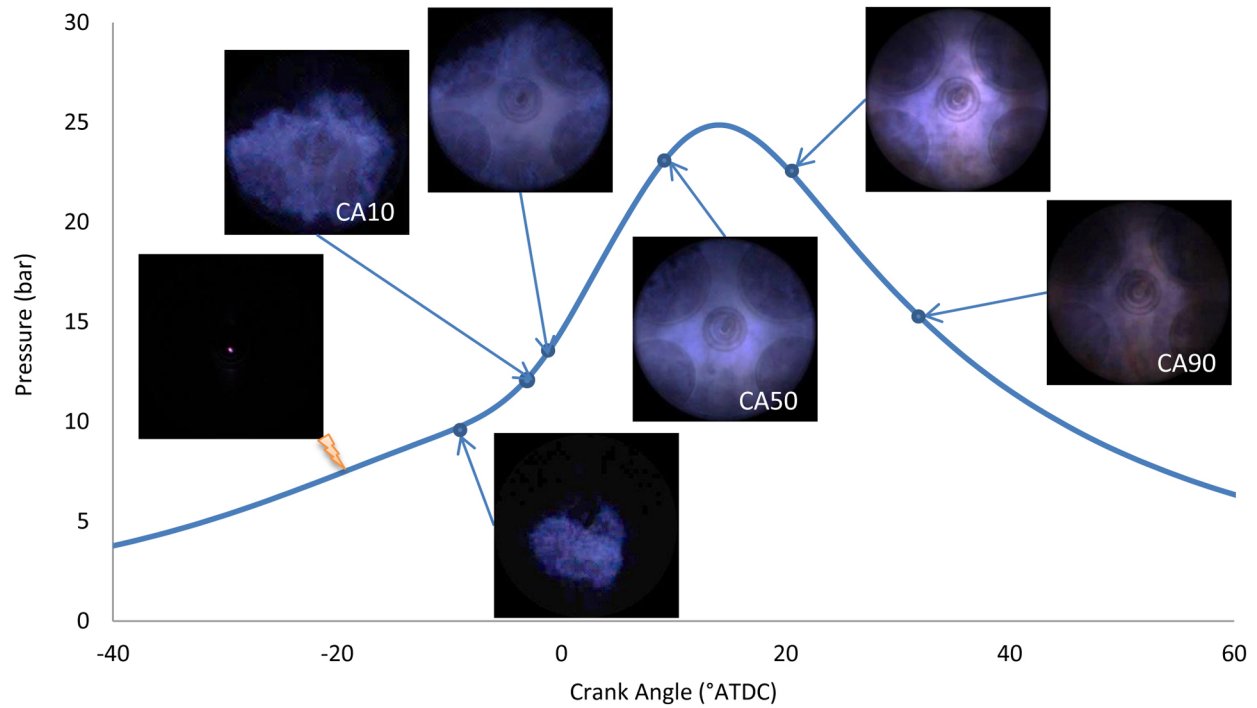


Figure A3. Spark ignition combustion pressure trace and images with CA10, CA50 and CA90 noted. $\lambda=1$, 1500 rev/min, 3.3 bar IMEPn, natural gas fuel.

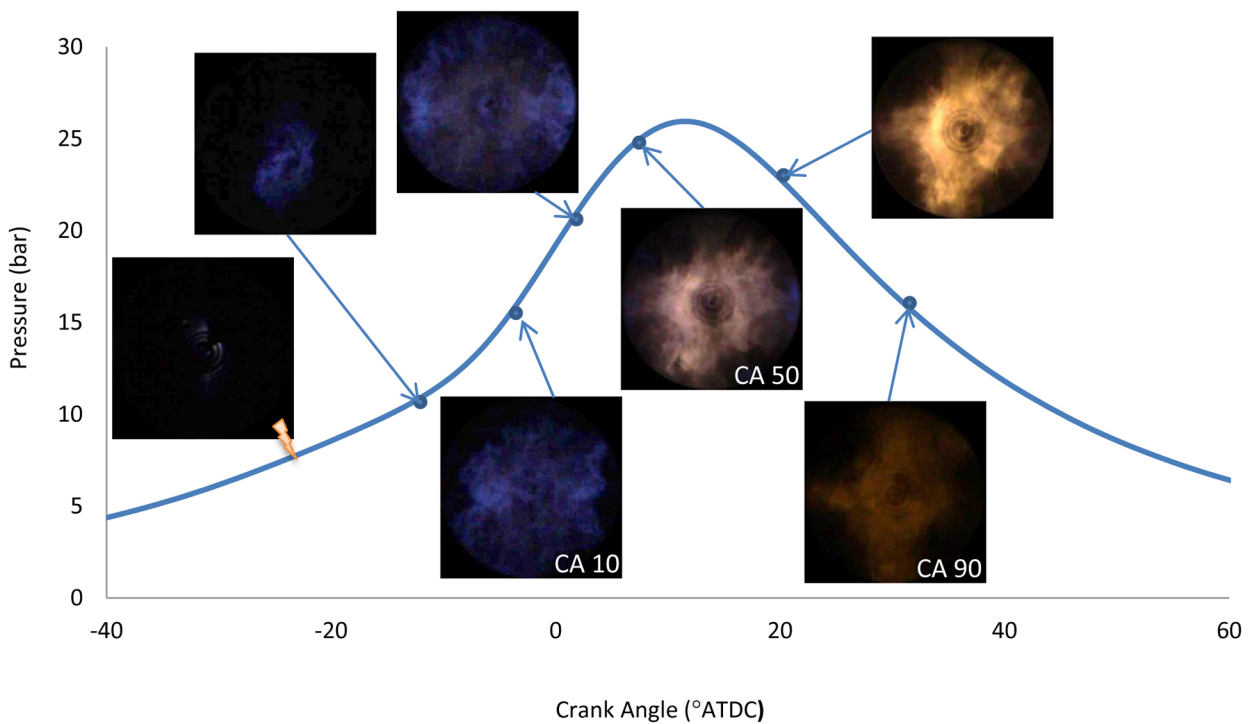


Figure A4. Spark ignition combustion pressure trace and images with CA10, CA50 and CA90 noted. $\lambda=1.4$, 1500 rev/min, 3.3 bar IMEPn, natural gas fuel.

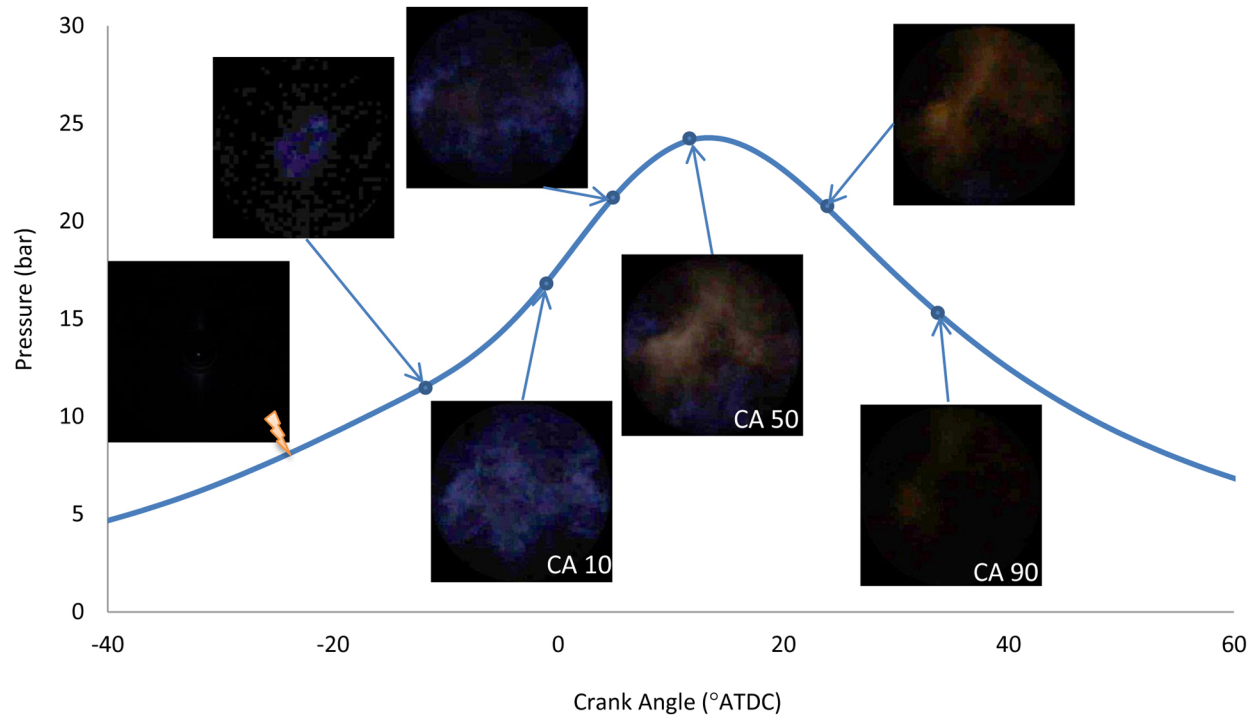


Figure A5. Spark ignition combustion pressure trace and images with CA10, CA50 and CA90 noted. $\lambda = 1.5$, 1500 rev/min, 3.3 bar IMEPn, natural gas fuel.

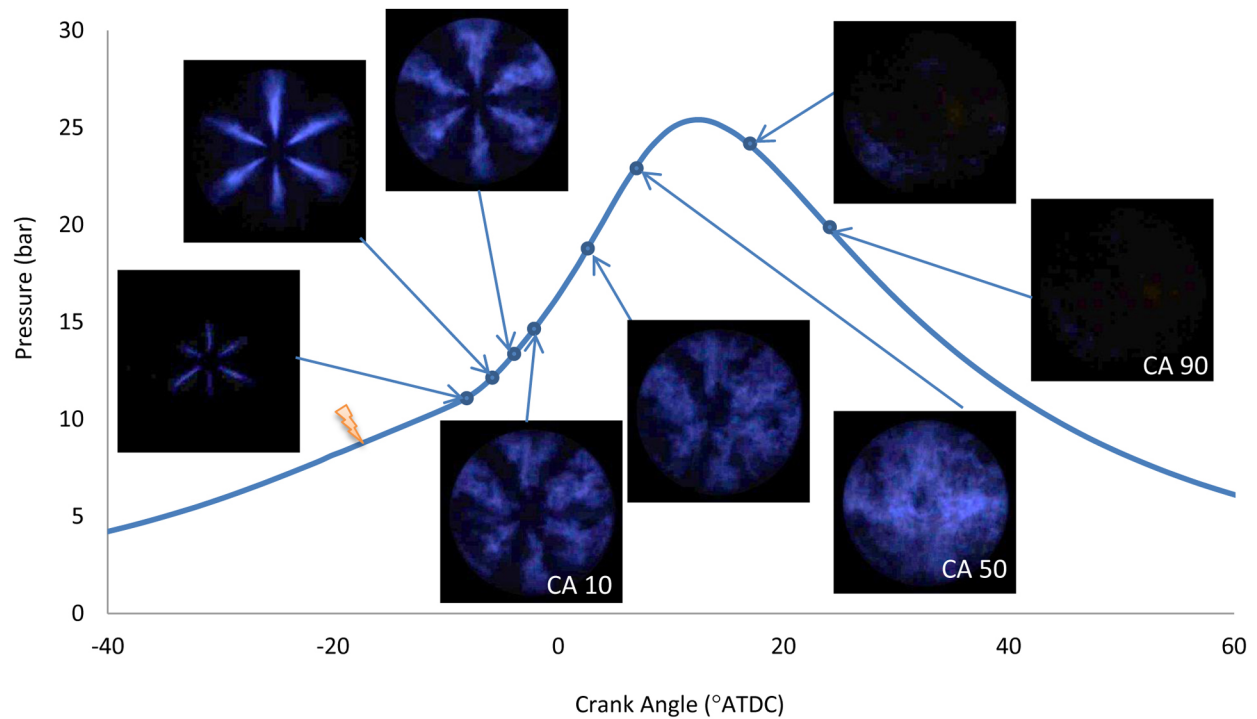


Figure A6. Pre-chamber Turbulent Jet Ignition combustion pressure trace and images with CA10, CA50 and CA90 noted. $\lambda = 1.8$, 1500 rev/min, 3.3 bar IMEPn.

The Engineering Meetings Board has approved this paper for publication. It has successfully completed SAE's peer review process under the supervision of the session organizer. This process requires a minimum of three (3) reviews by industry experts.

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