Title: Thermally Stratified Compression Ignition: A new advanced low temperature combustion mode with load flexibility

Authors: Benjamin Lawler, Stony Brook University
Derek Splitter, Oak Ridge National Laboratory
James Szybist, Oak Ridge National Laboratory
Brian Kaul, Oak Ridge National Laboratory

Abstract

A new advanced combustion mode is introduced, called Thermally Stratified Compression Ignition (TSCI), which uses direct water injection to control both the average temperature and the temperature distribution prior to ignition, thereby providing cycle-to-cycle control over the start and rate of heat release in Low Temperature Combustion (LTC). Experiments were conducted to fundamentally understand the effects of water injection on heat release in LTC. The results show that water injection retards the start of combustion due to the latent heat of vaporization of the injected water. Furthermore, for start of water injection timings between 20 and 70 degrees before top dead center, combustion is significantly elongated compared to without water injection. The 10% to 90% burn duration with 6.6 and 9.0 mg of water per cycle was 77% and 146% longer than without water injection, respectively. Direct water injection reduces the heat release rate by local evaporative cooling that results in a forced thermal stratification.

Finally, the load limits with and without water injection were determined experimentally. Without water injection, the load range was 2.3 to 3.6 bar gross IMEP. By using water injection to control heat release, the load range in TSCI was 2.3 to 8.4 bar gross IMEP, which is a range expansion of over 350%. These results demonstrate that direct water injection can provide significant improvements to both controllability and the range of operability of LTC, thereby resolving the major challenges associated with HCCI.

Keywords

low temperature combustion; advanced combustion; HCCI; heat release; thermal stratification

Acronyms

aTDC – after Top Dead Center
bTDC – before Top Dead Center
CA50 – Crank Angle of 50% burned location
CI – Compression Ignition

34 CO – Carbon Monoxide
 35 COV – Coefficient of Variation
 36 DI – Direct Injection
 37 EI – Emissions Index
 38 GCI – Gasoline Compression Ignition
 39 GDI – Gasoline Direct Injection
 40 HCCI – Homogeneous Charge Compression Ignition
 41 IMEP – Indicated Mean Effective Pressure
 42 IVC – Intake Valve Closing
 43 LTC – Low Temperature Combustion
 44 Mass PDF – Mass probability density function
 45 MPRR – Maximum Pressure Rise Rate
 46 NO_x – Oxides of Nitrogen (NO or NO₂)
 47 NVO – Negative Valve Overlap
 48 ORNL – Oak Ridge National Laboratory
 49 PCCI – Premixed Charge Compression Ignition
 50 PFS – Partial Fuel Stratification
 51 PLIF – Planar Laser Induced Fluorescence
 52 PM – Particulate Matter (soot)
 53 RCCI – Reactivity Controlled Compression Ignition
 54 SACI – Spark Assisted Compression Ignition
 55 SI – Spark Ignition
 56 SOI – Start of Injection
 57 TSA – Thermal Stratification Analysis
 58 TDC – Top Dead Center
 59 TSCI – Thermally Stratified Compression Ignition
 60 uHC – unburned Hydrocarbon

61 **Introduction**

62 Low Temperature Combustion (LTC) is a group of advanced combustion concepts that
 63 promises simultaneous reductions in fuel consumption and pollutant emissions. Homogeneous
 64 Charge Compression Ignition (HCCI) is one of the earliest forms of LTC and possibly the most
 65 widely researched [1-6]. In HCCI, a homogeneous mixture of air and fuel is compressed until it
 66 autoignites. HCCI therefore pairs the homogenous, ultra-low soot characteristics of conventional
 67 well-mixed Spark Ignition (SI) combustion with the high efficiencies (achieved through lean,
 68 unthrottled operation) usually typical of diesel Compression Ignition (CI) combustion. Engine-
 69 out NO_x emissions are kept low through high levels of dilution with air and/or residuals. Due to
 70 these factors, HCCI has demonstrated near-zero NO_x and soot emissions at efficiencies similar
 71 to, or greater than, conventional diesel combustion [5]. However, HCCI is only achievable over a
 72 narrow, part-load operating range due to the lack of direct control over the start and rate of heat
 73 release. In order to provide control over heat release in HCCI, a better understanding of the
 74 fundamental combustion processes is required.

75 Recently, the community gained a better understanding of LTC heat release through
 76 optical chemiluminescence and planar laser-induced fluorescence (PLIF) images, which showed
 77 that while HCCI can be compositionally homogeneous, there is a significant amount of thermal
 78 stratification (i.e. a distribution of in-cylinder gas temperatures prior to ignition) that stagger the

79 ignition timing of various regions based on their local temperature [7-17]. Depending on the
80 valve strategy and the fuel injection strategy, there can also be compositional gradients of either
81 residual gases or fuel fraction, respectively, that can also stagger the autoignition timing of
82 various regions [18-23]. With this new information, an improved understanding of LTC was
83 achieved: 1) regions autoignite sequentially based on their local temperature and reactivity, and
84 2) the early-igniting regions release their energy first, compressing the remaining unburned
85 regions and causing their subsequent autoignition. Due to this second underlying phenomenon,
86 the energy release process in LTC is a positive feedback loop, which explains its relatively low
87 high-load limit.

88 In addition to the insight about thermal stratification gained through the PLIF optical
89 diagnostics by Dec and other researchers around the world, a post-processing technique was
90 previously developed and validated by Lawler et al. [24-25] that can estimate an unburned
91 temperature distribution prior to ignition in a fired, metal engine. The technique, called the
92 Thermal Stratification Analysis (TSA), uses isentropic ideal gas relations to model the possible
93 gas temperatures in the cylinder and their change over the compression and combustion
94 processes. The autoignition integral [26] and an ignition delay correlation [27] are then used to
95 predict the ignition timing of each temperature trajectory, which are finally coupled to the
96 calculated heat release characteristics to estimate a temperature distribution. The TSA approach
97 has been previously validated against the optical distributions measured by Dec et al. as well as
98 against CFD simulations [25], and has subsequently been applied to study the effects of wall
99 conditions, engine geometry [28], and operating conditions [29] on thermal stratification. The
100 results showed that wall conditions had a surprisingly small influence on the thermal
101 stratification [28]. Instead, the top dead center (TDC) temperature, combustion phasing, and
102 swirl all had a noticeable influence on the level of thermal stratification prior to ignition [29].

103 In traditional HCCI, neither the compositional gradients, nor the temperature distribution
104 prior to ignition, nor the positive feedback loop are controlled. This lack of control is responsible
105 for the narrow, part-load operating range. To address the control challenges associated with
106 HCCI, a wealth of alternative LTC modes have been proposed. Reactivity Controlled
107 Compression Ignition (RCCI) uses two distinct fuels and a specific range of direction injection
108 (DI) timings to introduce a gradient of fuel properties in the cylinder prior to ignition [30-33]. By
109 controlling the fraction of each fuel and the DI timing, RCCI has demonstrated the ability to
110 achieve higher loads than pure HCCI with better controllability. However, the requirement of
111 two distinct fuels and fuel systems detracts from RCCI's commercial viability, especially for the
112 light-duty market. Spark Assisted Compression Ignition (SACI) is a variant of HCCI where a
113 spark discharge is used to help control the start and rate of heat release [17, 34-35]. In SACI, a
114 flame front burns between 5% and 50% of the mixture, and the remaining 50% to 95% of the
115 mixture autoignites similarly to pure HCCI. SACI has shown the ability to provide some level of
116 control and help extend the load range. However, the spark discharge introduces a significant
117 amount of variability which can be a challenge for LTC. Gasoline Compression Ignition (GCI) is
118 an advanced combustion concept that uses gasoline in a diesel engine [36-37]. GCI has its
119 advantages, but its NO_x emissions and pressure rise rates pose a challenge. Dec et al. introduced
120 Partial Fuel Stratification (PFS) which uses a second injection of fuel during the compression
121 stroke to provide a gradient of equivalence ratios to stagger the ignition timing of various regions
122 when using a ϕ -sensitive fuel (i.e. a fuel whose autoignition timing is sensitive to variations in
123 equivalence ratio) [21, 38].

Most of the alternative LTC modes that attempt to provide control over the start and rate of heat release rely on a direct fuel injection event to introduce a stratification of equivalence ratio and reactivity. This approach can be effective; however, the intentional fuel-air mixture inhomogeneities present a risk of higher particulate matter (PM) and NO_x emissions due to the locally rich regions which are closer to the soot production island and have higher burned gas temperatures. In fact, recent detailed particulate emissions measurements on these equivalence-ratio-stratified advanced combustion concepts has shown that although a smoke meter measurement may be zero, the particulate emissions are simply smaller in diameter and different in composition (increased concentrations of polycyclic aromatic hydrocarbons and aldehydes) [39-45]. This result, in combination with biological research which has shown that smaller particles are more detrimental to human health [46-47], might suggest that the equivalence-ratio-stratified concepts could shift the distribution of particulate matter to a more harmful region compared to conventional diesel combustion.

Instead of attempting to use a forced fuel-air mixture stratification to control the heat release rates in LTC, we propose a new combustion mode that controls the amount of thermal stratification in LTC. This approach, termed Thermally Stratified Compression Ignition (TSCI), employs direct injection of water to control both the mean temperature and the temperature distribution in the cylinder, thereby offering control over the start and rate of heat release in LTC. This paper first presents experimental data to fundamentally understand the effects of the direct water injection event on thermal stratification in the cylinder, and second, presents a load sweep to demonstrate the load limits that are achievable with and without water injection.

Water injection in internal combustion engines has a long history [48-55]. Water injection has been used in SI combustion to mitigate knock [48]. Water injection has also been used to lower temperatures and reduce NO_x emissions in any combustion mode [49-50]. Water injection has even been previously investigated in HCCI and Premixed Charge Compression Ignition (PCCI) [51-55]. Specifically, Lund Institute of Technology port injected water and found that the evaporative cooling in the intake manifold allowed the control of start of combustion by controlling the intake valve closing (IVC) temperature [51]. New A.C.E. Institute [53] and Hokkaido University [54-55] direct injected water through a diesel fuel injector in HCCI and PCCI, respectively, and found that water injection offers the ability to extend the load range of HCCI and PCCI.

This work expands upon the prior studies utilizing water injection in HCCI combustion by applying the TSA to fundamentally understand the impact of the direct water injection event on the temperature distribution prior to ignition. The improved understanding of the effects of direct water injection on the thermal stratification allows the development of the new TSCI combustion strategy. The TSA will be used throughout this paper to gain a better understanding of the effects of direct water injection on thermal stratification. Additionally, this study was conducted on a gasoline engine platform with GDI technology, typical of current light-duty engine hardware, as opposed to the studies in the literature that were conducted on diesel engine platforms.

Experimental Configuration

Experiments were conducted at Oak Ridge National Laboratory at the Fuels, Engines, and Emissions Research Center (FEERC). The engine was a production, gasoline, SI 2.0L General Motors (type LNF) engine. A production, gasoline engine was used to demonstrate the applicability of the proposed concept to the light-duty, passenger car market and because the

added hardware costs associated with water injection are more tolerable on a gasoline engine platform compared to a diesel engine platform. Of the production four-cylinder engine, only cylinder #4 was used; the remaining three cylinders were deactivated. A port fuel injector was retrofitted to the intake manifold and used to inject the 87 AKI certification gasoline. A schematic of the test-cell is shown in Figure 1.

Figure 1. Schematic of the test cell showing the engine, dynamometer, PFI injector, and side-mounted GDI injector [56]

The production piston, which provided a compression ratio of 9.2:1, was exchanged for a custom piston that increased the compression ratio to approximately 13:1 to enable LTC. Figure 2 shows the production piston in two different angles side-by-side with the custom 13:1 compression ratio piston, which was used in this work. The shape of the custom piston was not optimized for LTC or for the direct water injection spray. It was simply designed to reduce the clearance volume to provide a higher compression ratio and enable LTC.

Figure 2. The production 9.2:1 compression ratio piston (left) was replaced with the custom 13:1 compression ratio piston (right)

The production camshafts were exchanged for custom designed negative valve overlap (NVO) camshafts, which reduced the valve lift and duration. The production cam phasers were used to phase the NVO camshafts to adjust the NVO duration, thus changing the internal residual gas fraction (RGF). The intake and exhaust valve lift profiles are shown in Figure 3 including the range of possible phasings with the production cam phasers. The maximum NVO duration was 152 crank angle (CA) degrees and the minimum NVO duration was 52 CA degrees.

Figure 3. Valve lift profile for the custom NVO camshafts showing the phasing authority with the production cam phasers

The internal RGF was used to control combustion phasing by using the cam phasers to vary the NVO duration. Additionally, the engine was equipped with an external intake heater that was used as a coarse control over combustion phasing. The pressure in the cylinder was measured with a Kistler 6125C pressure transducer. 300 consecutive cycles were collected and analyzed. The pressure data were ensemble averaged. The coefficient of variation (COV) of indicated mean effective pressure (IMEP) was less than 3% for all of the data presented in this paper. The emissions were measured with a standard five-gas emissions bench. Combustion noise was measured with an AVL 4050 combustion noise meter.

The production, side-mounted, direct injector was used to inject water. A separate, external, high pressure pump was used to supply the pressure to direct inject deionized water. Due to limits of the high pressure pump, the maximum water injection pressure was 85 bar. Additionally, there was a minimum duration, below which the fuel injector would not open. Due to these constraints, Figure 4 shows the injection pressure and duration as a function of the desired water injection amount. To increase the amount of water injected at low water injection amounts, the injection duration is held at the minimum value of approximately 11 crank angle degrees while the injection pressure is increased from 20 bar to 85 bar. Once the injection pressure reaches its maximum value of 85 bar, the injection pressure is held constant while the injection duration is increased from 11 crank angle degrees to around 37 crank angle degrees. The injection pressure and duration for all of the data presented in this paper follows the values shown in Figure 4 based on injected water amount. The injection timings varied for each of the results presented below and therefore, will be described in the discussion of each of the specific results. Due to the minimum injection duration, and the rapidly decreasing injection pressure, there was an effective minimum mass of water that could be injected, which was approximately 6 mg/cycle. All engine testing was conducted at steady state at 2000 rpm. The net IMEP for the

data presented in Figures 5 through 9 was approximately 3.6 bar. For more information about the experimental setup, please refer to the prior work [56].

Figure 4. Injection pressure and duration as a function of the desired injected water amount

Results

Effect of water injection on the start of heat release in LTC

Data was collected with and without water injection to demonstrate the effects of water injection on the start of heat release in LTC. Three cases are considered and the relevant control parameters are shown in Table 1. Case 1 and 2 demonstrate that combustion phasing can be advanced by lengthening the NVO duration. This result agrees well with the literature which has shown that longer NVO durations elevate the IVC temperature by increasing the internal RGF [34, 56-57]. In Case 3, the NVO duration was held constant from Case 2 and a direct injection of 6.7 mg/cycle of water was introduced. The direct water injection timing was 50 degrees before TDC. Simply adding water injection retarded combustion to the point of misfiring. Since 152 degrees was the maximum NVO duration, the intake temperature was increased by 14 K to avoid misfires. The in-cylinder pressures and heat release rates for the three cases are shown in Figure 5. Comparing Case 3 to Case 1, the intake temperature is higher by 14 K and the NVO duration is longer by 22 degrees; however, the CA50 phasings approximately match because the water injection compensates for the elevated intake temperature and longer NVO duration. Water injection therefore significantly retards combustion phasing. This result agrees well with the findings by Christensen [51], Iwashiro [53], and Hokkaido University [54-55]. The latent heat of vaporization of the water spray decreases the charge temperatures and retards combustion phasing. It is important to note that the present work highlights the possibility that direct water injection could be used for cycle-to-cycle control of combustion phasing in LTC.

245 *Table 1. Operating conditions for the water injection comparison*

		Case 1	Case 2	Case 3
NVO Duration	[CA deg.]	130	152	152
Water Inj. Amount	[mg/cyc]	0	0	6.7
Intake Temp.	[C]	105	105	119
CA50	[deg. aTDC]	6.4	3.8	6.8

247
248 *Figure 5. Pressures and heat release rates for the water injection comparison at 2000 rpm, 3.6*
249 *bar net IMEP, water SOI timing of 50 degrees before TDC, showing that water injection retards*
250 *combustion phasing through evaporative cooling lowering the temperatures in the cylinder.*

251 Effect of water injection on the rate of heat release in LTC: Water Injection Amount

252 Although interesting, the finding that direct water injection can control the start of heat
253 release in LTC was expected and had been demonstrated previously in the literature [51, 53-55].
254 The main focus of this paper is to fundamentally understand the effect of direct water injection
255 on the temperature distribution prior to ignition using the previously developed [24] and
256 validated [25] Thermal Stratification Analysis. To accomplish this goal, the water injection
257 timing was held constant at 60 degrees before TDC (bTDC) while three different water injection
258 amounts were studied: 0 mg/cycle (no water injection), 6.6 mg/cycle, and 9.0 mg/cycle. CA50
259 was matched through adjustments to the intake temperature and the NVO duration. It is

important to compare the burn rates for varying water injection amounts at matched CA50 phasing because the burn duration in LTC is very sensitive to combustion phasing. The mass fraction burned curves for three conditions are shown in Figure 6, and Table 2 contains the burn durations and percent changes relative to the baseline (without water injection) condition. The mass fraction burned results show that the direct water injection cases exhibited a much longer burn duration than without water injection. In fact, with 6.6 mg/cycle of water, the 25% to 75% and 10% to 90% burn durations were 39% and 77% longer than the case without direct water injection, respectively. The elongation of combustion is correlated with the amount of water injected. This is evident from the slope of the mass fraction burned curve in Figure 6, or by comparing the burn durations in Table 2. The 25% to 75% and 10% to 90% burn durations with 9.0 mg/cycle of water injection were 62% and 146% longer than the burn durations without water injection, respectively.

Table 3 presents the emissions data for the water injection amount comparison. The results show that the CO₂ and O₂ are not strongly affected by water injection. NO_x emissions decrease with the addition of water injection, with 9.0 mg/cycle resulting in a factor of 8 reduction in NO_x emissions compared to without water injection. However, the unburned hydrocarbon (uHC) and CO emissions increase by 44% and 103%, respectively, with 9.0 mg/cycle of water compared to without water injection. The decrease in NO_x emissions and increase in CO and uHC emissions can all be attributed to lower peak temperatures and the reduced burn rates with water injection. It is important to realize that the case without water injection is within the range of operability of conventional HCCI. Therefore, these results show that if the burn rate in LTC is unnecessarily elongated, the NO_x emissions will decrease while the CO and uHC emissions will increase. The effects of water injection on CO and uHC emissions will be further examined in the discussion preceding Figure 11.

Table 2. Burn durations for the water injection amount comparison at 2000 rpm, 3.6 bar net IMEP demonstrating the ability of water injection to elongate heat release in LTC

		Without Water Injection	Water: 6.6 mg/cyc, SOI = 60 deg bTDC	Water: 9.0 mg/cyc, SOI = 60 deg bTDC
25% to 75%	Burn Duration [CA degrees]	5.0	7.0	8.2
	Compared to without water injection [%]	-	+ 39	+ 62
10% to 90%	Burn Duration [CA degrees]	12.6	22.2	30.8
	Compared to without water injection [%]	-	+ 77	+ 146

Figure 6. Mass fraction burned curves for the water injection amount comparison at 2000 rpm, 3.6 bar net IMEP showing that increasing the amount of water injected decreases the rate of combustion in LTC

Table 3. Emissions data for the water injection amount comparison at 2000rpm, 3.6 bar net IMEP

		No Water	SOI = 60 deg bTDC	
			6.6 mg/cyc	9.0 mg/cyc
uHC	[g/kg-fuel]	36.8	47.1	53.1
	[g/kW-hr]	11.8	16.1	18.9
NOx	[g/kg-fuel]	0.69	0.19	0.08
	[g/kW-hr]	0.22	0.07	0.03
CO	[g/kg-fuel]	11.3	16.3	22.9
	[g/kW-hr]	4.0	6.1	8.9
CO2	[kg/kg-fuel]	2.01	1.99	1.98
	[%]	9.04	9.18	8.94
O2	[kg/kg-fuel]	1.40	1.30	1.42
	[%]	6.30	6.02	6.42

To understand this phenomenon in more detail, the TSA was used to process the experimental results and estimate an unburned temperature distribution prior to ignition for each of the three cases in Figure 6. The unburned temperature distributions at TDC are shown in Figure 7 in the form of mass probability density functions (Mass PDFs), which describe the density of mass at a given temperature in the cylinder. The results show that the temperature distribution is broadened by direct water injection at 60 degrees bTDC. The reason for the broadening of the temperature distribution with direct water injection is that the water spray evaporates causing a local cooling in the areas targeted by the spray. Regions that were not

targeted by the spray remain unaffected. At this injection timing, there is enough time for the spray to evaporate and locally lower the temperatures of the gases targeted by the spray, but there is not enough time for the local effects to mix-out with the whole cylinder charge. The result is a broader temperature distribution due to the water injection event. The amount that the temperature distribution is broadened is correlated with the amount of water injected. Note that the increase in the hot side of the distribution with water injection is attributed to the increase in intake temperature and NVO duration and is not an effect of the water injection.

Figure 7. Unburned temperature distributions prior to ignition for the water injection amount comparison at 2000 rpm, 3.6 bar net IMEP. Water injection broadens the temperature distribution due to local evaporative cooling. Increasing the amount of water injected results in a broader temperature distribution.

Effect of water injection on the rate of heat release in LTC: Water Injection Timing

To provide further insight, the water injection quantity was held constant at 6.7 mg/cycle while the water injection timing was varied from 10 to 110 degrees bTDC (start of injection timings). Very small changes were made to the NVO duration to match CA50 for the injection timing sweep. Figure 8 compares the thermal widths, defined as the temperature distance between the 10% and 90% (or 25% and 75%) locations on the temperature distribution, at each injection timing to the reference case without water injection. The results show that injection timings at, or earlier than, 80 degrees bTDC do not spread the temperature distribution. At these timings, the spray evaporates, locally cooling the mixture in the cylinder, but there is enough time for the effect on the local temperature distribution to mix-out before ignition. While water injection at these early timings are not effective at controlling the rate of heat release in LTC, they are still effective at controlling the start of combustion. If the injection timing is 10 degrees bTDC or later, there is not enough time for the spray to evaporate and the effects on the temperature distribution are minimal (see Figure 8). This result shows that it takes approximately 20 crank angle degrees for the water spray to break up and evaporate at these conditions. If the injection timing is in the range from 20 to 70 degrees bTDC, the spray has a significant effect on the temperature distribution and the thermal widths are as much as 20% larger than without

water injection. At these timings, there is sufficient time before ignition for the spray to evaporate and locally cool the mixture, but there is insufficient time for this local cooling to mix-out and become homogeneous with the cylinder contents. The result is a forced thermal stratification. It is not surprising that these injection timings are in the same range as the high reactivity fuel injection event in RCCI combustion. This is because the physical processes and the desired effect are the same; to inject a liquid at a timing such that it can evaporate and introduce a stratification, with insufficient time to mix-out and become homogeneous. These results demonstrate that a direct water injection event in the range of injection timings of 20 to 70 degrees bTDC can forcibly amplify the level of thermal stratification in the cylinder prior to ignition, thereby offering cycle-to-cycle control over the rate of heat release in LTC.

Figure 8. Thermal widths as a function of the start of water injection at 2000 rpm, 3.6 bar net IMEP with 6.7 mg of water injected per cycle. Early and late direct water injections are not able to spread the temperature distribution. However, injections in a certain window before TDC increase the thermal stratification significantly.

Load limits with and without water injection

The previous sections demonstrated that direct water injection can control both the start and rate of heat release in LTC. The TSA helped illuminate the fundamental mechanism by which this is possible; latent heat of vaporization of the spray locally cooling the mixture. If the injection occurs too early, the cooling effect becomes homogeneous with the cylinder contents, and the net effect is a lower average temperature, and a later start of heat release. If the injection timings are in a specific range before TDC, there is enough time for the spray to evaporate and locally cool the charge targeted by the spray, but there is insufficient time for this local cooling to mix-out. The net effect is a sizeable, forced thermal stratification that staggers the autoignition timing of various regions based on their temperature, and decreases the rate of heat release in LTC.

The goal of this subsection is to use the fundamental knowledge gained through the previous sections' experimental investigations and analyses with the TSA to expand the operable load range of LTC. In this section, load sweeps in LTC with and without water injection are studied. Since determining the load limits involves simultaneously varying a large number of different control parameters, these load limits results are less fundamental than the results in the previous sections. However, they help to provide insight into the range of operability in a vehicle application.

The low load limit was defined as a COV of IMEP of 3%. The high load limit was defined as either a maximum pressure rise rate (MPRR) of 5 bar/degree or an AVL noise meter reading of 95 dB. These load limits are fairly conservative especially compared to other researchers who may use 5% COV of IMEP and 10 bar/degree as the low and high load limits, respectively.

Figure 9 shows the experimentally-obtained load limits with and without water injection. To find the load limits without water injection, the fueling rate was increased while the NVO duration was adjusted to retard combustion phasing as much as possible. As can be seen in the figure, the load limits without water injection ranged from 2.3 bar to 3.6 bar IMEP on a gross basis. These load limits are slightly narrower than what has been presented in the literature and that can be explained by 1) the conservative load limits used in this work, and 2) the use of a production engine and cam phasers that introduce more variability than some experimental research setups. For these reasons, these results might more closely reflect the load limits of LTC in a production engine. As can be seen from Figure 9, the relationship between MPRR and load or AVL noise and load are linear in HCCI without water injection. The very steep slope, which leads to the narrow operating range between the low and high load limits, is due to the lack of control and the positive feedback nature of the HCCI energy release process.

When direct water injection is added, the relationship between AVL noise and load or MPRR and load is broken, and the high load limit is extended. This is because the water injection event provides the ability to control the rate of heat release. In fact, at 5.6 bar with water injection, the AVL noise was 6.3 dB lower than the noise at 3.6 bar without water injection. Since the decibel scale is logarithmic, a change of 6 dB corresponds to a factor of 4 in power ratio. Therefore, the 5.6 bar condition with water injection had 4 times lower acoustic power than the 3.6 bar condition without water. A maximum load limit of 8.4 bar gross IMEP was reached. This type of load extension was also seen in the literature [53-55], although the maximum load achieved here was slightly higher than what was previously presented. At 8.4 bar, the variability started to approach 3% COV of IMEP. At this condition the ringing limit and the COV limit were the same, which is common for high load LTC.

These results illustrated that water injection can be used to extend the load limit by inducing forced thermal stratification. It should be noted, however, that the hardware used in this investigation was not designed for this application and as a result there is likely room for considerable improvement. With the current spray targeting, it is not known how much of the water evaporates, but the side-mounted injection strategy likely resulted in significant impingement and wall-wetting, which is an inefficient use of the water. With intent-designed hardware, it is possible that the amount of water that needs to be injected to achieve similar results could be reduced significantly, which then has ripple effects on the exhaust temperature, the turbocharger efficiency, and ultimately the amount of dilution that is possible. Additionally, the water injection pump had a maximum pressure of 85 bar, which limited the amount of water that could be injected in the window that provides control over the rate of heat release. If the

injection pressure could have been increased to 150-200 bar, which are the current operating pressures of modern production GDI systems, more water could have been injected in the range that provides control over the rate of heat release and the maximum load limit might have been extended further.

The operable load range without water injection (pure HCCI) according to these results is 1.3 bar (2.3 to 3.6 bar). When operation with and without water injection is combined in what we've termed Thermally Stratified Compression Ignition, the operable load range is 6.1 bar. Therefore, TSCI has a 370% larger operating range than pure HCCI.

Figure 9. Load limits with and without water injection. Water injection is able to significantly extend the high load limit by providing control over LTC heat release.

Figure 10 shows some other relevant conditions for the load sweep. Retarding CA50 to extend the load is common in LTC. However, in pure HCCI, there is a limit on the amount to

which CA50 can be retarded before the misfire limit is reached. Without water injection, this limit occurs around 8.5 degrees after TDC (aTDC). With water injection, however, CA50 can be extended to almost 14 degrees aTDC because the lower rate of combustion allows some energy to be released earlier in the cycle. The second subplot in Figure 10 shows the intake pressure over the load sweep. Below 6 bar, the intake pressure was kept at, or just under, atmospheric pressure. However, above 6 bar, the NVO camshafts shown in Figure 3 simply could not induct any more fresh charge. At this point, the intake pressure was increased to force more fresh charge into the cylinder. To keep the results as realistic as possible, the back pressure was also increased at this point, and the calculated turbocharger efficiency was kept at 25% or below. This shows the need for a variable valve timing technology with variable duration to achieve better load flexibility at naturally aspirated conditions. The final subplot in Figure 10 shows the mass of fuel and water injected in milligrams per cycle. The fuel mass is linearly proportional to load. The first four load conditions did not have water injection. From 4.0 bar to 6.5 bar, the amount of water injected was less than the amount of fuel injected. Above 6.5 bar, the amount of water injected surpassed the amount of fuel injected. By 8.4 bar, the amount of water injected was a little less than twice the amount of fuel injected.

Figure 10. CA50, intake pressure, and injected fuel and water masses for the load sweep.

The emissions index (EI) NOx emissions, equivalence ratio, and combustion efficiency over the load sweep are shown in Figure 11. The engine-out NOx emissions in conventional, low load HCCI were ultra-low due to the low peak temperatures. With water injection and the associated load extension, the peak temperatures increased, which caused the NOx emissions to increase dramatically. In fact, the NOx emissions at 4.5 bar were approximately 7 times as much as the NOx emissions at 3.6 bar. To resolve this issue, the equivalence ratio was increased to approximately stoichiometry such that a production three-way catalyst could be used to

simultaneously reduce the NO_x, CO, and uHC emissions. The equivalence ratio over the load sweep is shown in the second subplot in Figure 11. There is still a large amount of dilution, but all of the dilution is accomplished through internal residuals by controlling the NVO duration, while the incoming fuel and air are kept at stoichiometry. This technique for reducing tailpipe NO_x emissions is effective and relatively inexpensive (three-way catalyst are used universally in production SI engines); however, there is an efficiency penalty associated with this approach because lean, air-dilute mixtures have a more favorable ratio of specific heats than residual-dilute mixtures. An alternative approach could have been to increase the water injection amount at these mid-range loads to bring the peak temperatures back down below the NO_x production threshold. At the higher load range, the amount of water is sufficient to reduce the engine-out NO_x emissions even though the load is high. The final subplot in Figure 11 shows the combustion efficiency. Combustion efficiency trends with load for the four conditions without water injection. Once the load is increased above 4 bar, the combustion efficiency is very high, approaching SI and conventional diesel combustion values. This is because the burned gas temperatures are higher at these higher loads, and the equivalence ratio is near stoichiometry. CO and uHC emissions (not shown for conciseness) follow the inverse trend as combustion efficiency. They are highest for the low load, pure HCCI conditions. Once water injection is added and the load is extended, the CO and uHC emissions decrease (see Table A1 in the Appendix). This result is significant because it proves that the increased CO and uHC emissions associated with water injection that was shown in Table 3 was simply a result of using water injection at a low load when the elongation of combustion was not necessary. The emissions data for the load sweep shows that water injection at higher loads exhibits lower CO and uHC emissions than pure HCCI. Soot emissions or smoke meter readings were not recorded for this data set, since the particulate matter emissions in premixed LTC combustion modes have been shown to be very low in comparison to conventional diesel combustion [58]. One advantage of the proposed strategy is that port injecting the fuel results in lower particulate matter than early direct injection strategies [59]. A large number of other relevant engine operating conditions and results for the load sweep are shown in Table A1 in the Appendix.

Figure 11. NO_x, equivalence ratio, and combustion efficiency for the load sweep.

Conclusions

In this paper, we propose a new advanced combustion mode, Thermally Stratified Compression Ignition (TSCI), which uses direct water injection to control the start and rate of heat release in Low Temperature Combustion. Experiments were conducted to better understand the effects of water injection on LTC, with particular focus on comparisons between operation with and without water injection, as well as the effects of water injection amount and timing. Finally, the load limits with and without water injection were determined. From these results, the following conclusions can be drawn:

- 1) Adding water injection retards combustion phasing due to the latent heat of vaporization of the water, which cools the mixture. The amount of phasing retard is proportional to the amount of water injected. Therefore, water injection can be used for cycle-to-cycle control of the start of heat release in LTC.
- 2) Direct water injection in the range of 20 to 70 degrees bTDC reduces the heat release rate by locally cooling the mixture in the areas targeted by the spray, therefore forcibly amplifying the level of thermal stratification in the cylinder. The amount of heat release elongation trends with the amount of water injected in this range of SOI timings. Direct water injection can therefore be used for cycle-to-cycle control over the rate of heat release in LTC.
- 3) By using direct water injection to control the heat release rate, the high load limit was extended from 3.6 bar to 8.4 bar gross IMEP. TSCI has a 370% larger operating range than pure HCCI combustion on this engine setup.

These results are preliminary in nature, but demonstrate the potential of water injection to provide cycle-to-cycle control over both the start and rate of heat release in LTC, thus resolving the major challenge of pure HCCI combustion.

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Appendix

The table below shows the relevant operating conditions over the load sweep for reference. The emissions results are presented as either emissions index values or as indicated specific emissions.

690 Table A1. Operating conditions for the load sweep without and with water injection

691

		Gross IMEP [bar]													
		2.3	2.8	3.3	3.6	4.0	4.5	5.0	5.6	6.1	6.5	6.8	7.4	7.8	8.4
Net IMEP	[bar]	2.11	2.62	3.11	3.42	3.84	4.32	4.83	5.43	6.06	6.33	6.50	6.98	7.30	7.67
PMEP	[bar]	0.18	0.18	0.19	0.20	0.16	0.18	0.20	0.20	0.08	0.15	0.30	0.39	0.47	0.76
Residual Rate	[%]	52.8	42.9	35.5	33.6	34.5	31.9	27.5	25.1	22.6	22.8	23.7	24.3	26.1	27.4
Equivalence Ratio	[-]	0.65	0.63	0.63	0.66	0.82	0.93	0.97	0.98	0.98	0.99	0.97	0.98	0.98	0.98
CA10	[degrees]	1.7	3.1	4.2	4.4	4.7	5.4	7.0	7.0	6.8	6.2	5.9	4.5	5.1	3.8
CA50	[degrees]	6.3	7.1	8.2	8.3	9.2	10.4	13.3	13.1	13.8	12.9	13.4	11.0	13.0	11.1
CA90	[degrees]	28.8	26.6	18.6	17.9	18.5	18.7	25.0	22.0	25.3	24.6	28.6	21.4	29.3	23.6
10% to 90% Burn Duration	[degrees]	27.2	23.5	14.4	13.5	13.9	13.3	18.0	14.9	18.5	18.4	22.7	16.9	24.2	19.8
Max. Pressure Rise Rate	[bar/degree]	2.1	3.1	4.0	5.3	3.8	3.5	2.7	3.1	3.4	3.6	3.4	4.8	3.5	4.5
AVL Noise	[dB]	86.8	90.3	92.7	94.8	91.3	90.7	89.1	88.5	90.4	91.2	89.2	91.6	89.7	91.2
Mass of Fuel	[mg/cycle]	7.2	8.2	9.6	10.5	11.8	13.4	15.3	16.8	18.9	20.5	22.0	23.8	25.5	27.3
Injection	Mass of Water	[mg/cycle]	NA	NA	NA	7.2	10.0	12.0	13.6	16.4	20.1	27.4	33.9	41.6	50.2
	Pressure	[bar]	NA	NA	NA	19.0	34.0	47.9	61.2	85.2	85.1	85.3	85.1	84.9	85.3
	Start of Injection	[degrees]	NA	NA	NA	-60	-60	-60	-60	-59	-64	-65	-66	-70	-70
	End of Injection	[degrees]	NA	NA	NA	-49	-49	-49	-49	-48	-50	-46	-42	-40	-34
Intake Pressure	[bar]	0.94	0.91	0.93	0.94	0.92	0.91	0.93	0.97	1.05	1.13	1.27	1.39	1.56	1.72
Efficiency	Turbocharger	[%]	NA	NA	NA	NA	NA	NA	NA	20.3	21.9	22.4	25.7	25.0	24.7
	Combustion	[%]	95.7	96.4	97.0	97.7	98.2	98.5	98.6	98.8	98.4	98.8	99.0	98.5	98.6
	Rel. Gross Thermal	[%]	93.4	99.4	100.0	99.5	97.3	96.2	94.0	95.6	92.7	90.4	88.2	88.2	88.1
	Rel. Net Thermal	[%]	86.0	92.9	94.1	94.0	93.3	92.3	90.3	92.2	91.6	88.4	84.3	83.6	81.9
	Rel. Net Fuel Conv.	[%]	82.3	89.6	91.3	91.8	91.6	91.0	89.0	90.8	90.4	87.0	83.3	80.6	79.0
Emissions	uHC	[g/kg-fuel]	71.1	64.9	55.0	43.2	34.8	27.9	26.7	25.0	21.0	22.3	21.4	17.2	18.5
		[g/kW-hr]	37.2	25.8	19.0	14.4	12.0	9.4	8.6	7.7	6.3	7.0	7.1	5.8	7.0
	NOx	[g/kg-fuel]	0.08	0.12	0.23	0.52	1.35	3.53	4.07	4.92	5.43	5.16	3.86	2.54	1.77
		[g/kW-hr]	0.04	0.05	0.08	0.17	0.46	1.18	1.32	1.51	1.62	1.62	1.29	0.87	0.64
	CO	[g/kg-fuel]	45.5	27.7	19.0	14.1	10.6	8.2	9.5	17.6	13.2	25.5	10.4	10.4	18.2
		[g/kW-hr]	23.8	11.0	6.5	4.7	3.6	2.7	3.1	5.4	4.0	8.0	3.5	3.5	6.6
	CO2	[kg/kg-fuel]	2.11	2.13	2.15	2.18	2.24	2.29	2.30	2.30	2.31	2.30	2.31	2.32	2.30
		[%]	9.03	8.79	8.92	9.46	11.9	13.6	14.3	14.5	14.5	14.6	14.4	14.5	14.5
	O2	[kg/kg-fuel]	1.85	2.01	1.97	1.71	0.79	0.30	0.15	0.09	0.10	0.06	0.13	0.10	0.11
		[%]	7.93	8.32	8.16	7.44	4.17	1.80	0.90	0.58	0.61	0.38	0.78	0.60	0.68

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