

Improving the Understanding of Intake and Charge Effects for Increasing RCCI Engine Efficiency

Derek Splitter, Martin Wissink, Dan DelVescovo, and Rolf Reitz
University of Wisconsin

ABSTRACT

The present experimental engine efficiency study explores the effects of intake pressure and temperature, and premixed and global equivalence ratios on gross thermal efficiency (GTE) using the reactivity controlled compression ignition (RCCI) combustion strategy. Experiments were conducted in a heavy-duty single-cylinder engine at constant net load (IMEP_n) of 8.45 bar, 1300 rev/min engine speed, with 0% EGR, and a 50% mass fraction burned combustion phasing (CA50) of 0.5°CA ATDC. The engine was port fueled with E85 for the low reactivity fuel and direct injected with 3.5% 2-ethylhexyl nitrate (EHN) doped into 91 anti-knock index (AKI) gasoline for the high-reactivity fuel. The resulting reactivity of the enhanced fuel corresponds to an AKI of approximately 56 and a cetane number of approximately 28.

The engine was operated with a wide range of intake pressures and temperatures, and the ratio of low- to high-reactivity fuel was adjusted to maintain a fixed speed-phasing-load condition. This allowed for the investigation of several combinations of intake temperature, intake pressure, and charge stratification at otherwise constant thermodynamic conditions. The results show that sources of engine inefficiency compete as functions of premixed and global equivalence ratios. Losses are minimized through proper balancing of intake pressure and temperature, such that the global equivalence ratio (Φ_{global}) is as lean as possible without overly lean regions of the stratified charge causing an increase in incomplete combustion. The explored speed-load-phasing combination shows that losses are minimized at conditions where approximately 2/3 of the fuel is fully premixed. The results exhibit a pathway for achieving simultaneous increases in combustion and fuel efficiency through proper fuel reactivity and initial condition management.

CITATION: Splitter, D., Wissink, M., DelVescovo, D., and Reitz, R., "Improving the Understanding of Intake and Charge Effects for Increasing RCCI Engine Efficiency," *SAE Int. J. Engines* 7(2):2014, doi:10.4271/2014-01-1325.

INTRODUCTION

Recent advances in high-efficiency advanced combustion concepts have shown that high GTE is achievable. For example recent work with gasoline compression ignition by Sellnau et al. [1] from Delphi, and Manente et al. [2] from Lund have demonstrated that gasoline compression ignition can be optimized to offer high thermal efficiencies. These strategies demonstrate methods for increasing engine efficiency through the use of gasoline fuels with diesel-like engine designs and technologies. Likewise, Southwest Research Institute has demonstrated that the conventional gasoline engine can exhibit increases in efficiency through the inclusion of technologies like cooled external EGR [3], a technology that is typically used only in diesel engines. These brief examples demonstrate how diesel and gasoline engines are evolving to be more similar in the quest for high efficiency.

RCCI illustrates this convergence of engine technologies and designs, because it operates with autoignition of an in-cylinder blend of diesel-like and gasoline-like fuels [4]. The ratio and injection schedule of the fuels can be adjusted such that definable fuel reactivity gradients initiate and control ignition [5]. These can be directly tailored to operating conditions [6], enabling direct control of pressure rise rate, and allowing combustion to be phased to near thermodynamically ideal conditions over a wide speed and load range [7], up to 18 bar brake mean effective pressure with less than 12 bar/°CA peak pressure rise rate (PPRR) [8].

Recent multi-cylinder engine work by Curran et al. [9] has demonstrated that brake thermal efficiencies of RCCI are competitive with or higher than production conventional diesel combustion (CDC). Although these studies have demonstrated the operable flexibility and performance of RCCI, they have not provided a detailed investigation of the relations between operational parameters affecting engine efficiency, such as boost, intake temperature, and charge stratification.

To analyze engine efficiency, a first and second law approach can be used. That is, the energy or exergy (availability) flows/paths can be tracked. First and foremost, the source of exergy is from the fuel itself. Fuel exergy depends on chemical composition, but for most fuels, it is roughly equal to the heating value of the fuel [10], a widely adopted approximation. Thus, the fuel path should be tracked to examine engine efficiency and the sources of losses. Several internal combustion engine exergy studies have been performed, with notable works by Caton on the exergy destroyed in combustion [11] and heat transfer [12], and on a crank angle basis by Sexana et al. [13]. In these works, it was shown that the dominant losses consisted of incomplete combustion, heat losses, exhaust losses, and work losses. Remaining losses, from flows for example, were negligible compared to these other losses.

One of the most notable findings from these works is that the exergy destroyed (an inherent unrecoverable loss due to entropy generation) increases at reduced combustion temperatures and leaner equivalence ratios. This would suggest that high combustion temperatures would increase engine efficiency; however, increases in combustion temperature also increase heat transfer potential, and heat transfer also destroys exergy [14]. This presents a paradox in optimizing engine efficiency: is it best to reduce or increase combustion temperatures?

Several researchers have examined this paradox in efficiency with notable works on quantified heat transfer losses by Rakopoulos et al. [14] and Caton [12]. Their results show that the low-temperature gradient between the cylinder internal/coolant temperatures (typically 100-150°C) and the reference state ($T_o \sim 25^\circ\text{C}$), results in approximately 100% of this exergy being destroyed. Thus, in low-temperature combustion (LTC), decreased combustion and product temperatures reduce the driving potential for heat transfer.

For example, Hendricks et al. [15] measured a 25-50% reduction in dynamic surface-averaged piston heat transfer with RCCI compared to CDC across a wide range of heavy-duty engine operating loads and speeds. An interesting finding from that study was that the reduction in heat transfer between CDC and RCCI occurred at matched global equivalence ratios (Φ_{global}), suggesting that local effects such as charge stratification are important for increasing efficiency. These experimental results are supported in RCCI and CDC modeling work by Kokjohn et al. [16], which illustrated that at matched Φ_{global} , the more homogeneous charge of RCCI can increase bulk gas temperatures more than CDC, but also can result in lower overall heat transfer. That work shows that this seeming paradox in gas temperature and heat transfer occurs because the highly stratified and turbulent nature of CDC combustion results in combustion with near-adiabatic flame temperatures

on or near the piston bowl and combustion chamber walls, increasing heat transfer potential. These cited differences in methods and magnitudes of in-cylinder heat transfer illustrate that the LTC heat transfer reduction potential might offset the increase in exergy destroyed during lean LTC combustion.

Edo and Foster [17] demonstrated this balance in exergy through a second law analysis which showed exergy destruction increases at lean conditions, but other losses such as heat transfer and exhaust losses are reduced, opening the potential for increased work extraction and efficiency. More recent general LTC overviews on the relations of sources of efficiency have been performed by Foster [18] and Edwards [19]. These studies concluded that the increased exergy destruction during the combustion of lean mixtures has the potential to be balanced or even offset by advantages in heat transfer and expansion work. The thermodynamic advantages of reduced combustion temperatures are pointed out by Lavoie et al. [20], where reduced temperatures improve the ratio of product specific heats (γ), increasing the polytropic expansion coefficient and thus expansion work.

Although there can be heat transfer loss reductions when operating LTC, at mid-to-high loads, the short combustion duration with higher energy release can result in knock with strong in-cylinder pressure oscillations [21, 22]. It is well known that avoiding knock is critical for controlling heat losses. Syrimis et al. [23] demonstrated that heat flux from knocking zones is often double that of non-knocking zones. Moreover, Lu et al. [24] saw that in-cylinder heat flux linearly increased with pressure oscillation amplitude, starting at loads as low as 5 bar. In-cylinder diagnostics techniques have been used to better understand the sources of these increases such as the use of Coherent Anti-Stokes Raman Spectroscopy by Grandin et al. [25]. Their results supported the previous researchers' findings and suggested that in knocking conditions, the thermal boundary layer thins, resulting in increased near-wall temperatures and reduced thermal resistance. Thus, when operating very lean LTC at mid-to-high loads, a tradeoff in heat transfer losses may exist between knock and dilution.

Based on these findings, it is suggested that dilute, knock-free combustion with reduced temperatures offers the potential to further increase engine efficiency as opposed to higher combustion temperatures. However, it is well known that combustion temperatures below $\sim 1500\text{ K}$ result in incomplete combustion products, mainly CO [26]. These can be commonly encountered in LTC, particularly with overly lean regions of the charge occurring from stratification [27]. Therefore, the temperature and equivalence ratio reductions with LTC must not prevent complete fuel oxidation. One common way to quantify incomplete combustion is with combustion efficiency; a term defined by normalization of incomplete combustion product energy to the input fuel energy. Note that incomplete

combustion products are discarded combustible exhaust species, and thus are still fuels which have been chemically altered compared to the input fuel(s). Thus, in LTC, there is the potential for a tradeoff in exergy usage (i.e., fuel energy) between heat transfer and combustion efficiency. This presents the opportunity for the optimization of engine operating conditions to minimize the sources of losses.

Although these studies have shown that lean boosted conditions can increase engine efficiency, evidence of fuel-pressure sensitivity in boosted LTC has been demonstrated in recent research by Dec et al. [28]. That work illustrated that intake pressure was found to have an appreciable effect on increasing the reactivity with real gasoline fuels, thereby limiting the maximum load or CA50 advance. That study also found that similar high-octane PRF fuels and ethanol do not strongly exhibit this dependency. Later, supporting work by Sjöberg and Dec [29] demonstrated that at matched conditions, ethanol displayed weak or no pressure sensitivity.

These experimental findings are supported by detailed kinetics simulation results by Mehl et al. [30]. A key finding of that study concluded that increased pressure was likely responsible for enhancing the formation of intermediates like the hydroperoxy radical (HO_2), and with reduced temperatures, the chain branching reaction of H_2O_2 into 2OH is reduced, extending the ignition delay. If sufficiently low temperatures were used, fuel radical oxidation reactions ($\text{R}+\text{O}_2$) became dominant, resulting in negative temperature coefficient behavior. Thus, a fuel with low reactivity as determined by the research and motor octane tests (ASTM 2699 RON and 2700 MON respectively), may display increased reactivity if exposed to boosted low intake temperatures conditions (i.e., LTC operating conditions). These combined findings illustrate that if lean highly boosted conditions are required for high efficiency engine operation, fuel properties may become increasingly important, and that for some LTC strategies, the reduced pressure dependency of ethanol containing fuels may enable leaner conditions at higher loads.

In mid-to-high-load RCCI operation, the high reactivity fuel is used in a small proportion relative to the low reactivity fuel [7], and thus is more responsible for the onset of heat release and ignition enhancement than as the source of main chemical heat release, as noted by Kokjohn et al. [5] and Splitter et al [31]. Therefore, the chemical composition of the high-reactivity fuel is less influential on the limits of engine operating conditions than the low-reactivity fuel, and can be modified with reduced influence on combustion than the low-reactivity fuel [7].

One interesting approach to high reactivity fuels is to add cetane number (CN) improvers to low reactivity fuels, such as demonstrated by Dempsey et al. [32]. Two common CN improvers are EHN and di-tert butyl peroxide (DTBP). Only a small percentage of either of these chemicals is required to achieve significant increase in cetane number. Regardless of the CN improver used, the effect that CN improvers have on the overall fuel oxidation has been shown to be insignificant.

Higgins et al. [33] demonstrated that the addition of less than 1% (by mass or volume) of CN improver to diesel fuel dramatically shortens the ignition delay of the fuel. However, their findings also noted that the combustion characteristics of the base fuel remained unaltered as the ignition delay decreased with the addition of CN improver.

The effect of CN improvers has been explored in RCCI by Splitter et al. [34] and later by Hanson et al. [35]. The first cited work demonstrated that 3.5-0.75% by volume of DTBP CN improved gasoline could be used as an acceptable high-reactivity fuel, with CA50 and general performance characteristics that were well matched to RCCI with #2 Ultra low-sulfur-diesel fuel as the high-reactivity fuel. The later study demonstrated similar trends with EHN, which is more common because refineries sometimes add it to commercial ULSD blends.

The cited works illustrate that sources of engine inefficiency can be a strong function of Φ_{global} (i.e., dilution) and the RCCI combustion process, but to achieve higher loads, fuel properties may become increasingly important. The effect of fuel properties on efficiency are indicated in detail in Splitter and Reitz [36]. The present study focuses on building upon these findings by experimentally exploring how initial conditions (i.e., IVC conditions) affect mid-load RCCI gross cycle¹ engine fuel efficiency and sources of fuel energy losses. A parametric analysis of intake pressure and temperature is used with fixed engine speed, CA50, fuel injection timings, and IMEP_n. To maintain CA50 and IMEP_n, the ratio of low-to-high fuel reactivity are tailored as required, thus also changing the fuel reactivity and local equivalence ratio distribution. This enables a detailed analysis of IVC pressure, temperature, and in-cylinder mixing interaction to be compared at constant engine output and cycle thermodynamics, and makes it possible to identify and optimize for operating conditions that are favorable for maximizing engine fuel efficiency.

EXPERIMENTAL PROCEDURE

Laboratory Hardware and Systems

Single-cylinder engine experiments were performed on a Caterpillar SCOTE. The engine geometry is given in Table 1, with a diagram of the engine lab in Figure 1. The experiments used a modified piston optimized for RCCI combustion. Details of the piston design and optimization can be found in [37]. The piston profile can be seen in Figure 2, where it is overlaid on the stock piston geometry. Specific details of the geometry are listed in Table 2.

1. Gross cycle is defined as a 360°CA cycle consisting of the compression and expansion strokes (BDC-TDC-BDC). This does not account for efficiency losses from the engine air handling systems.

Table 1. Stock 3401 SCOTE geometry

Displacement	2.44 (L)
Bore x Stroke	13.72 x 16.51(cm)
Connecting Rod Length	21.16 (cm)
Swirl Ratio	0.7
Number of Valves	4
Intake Valve Opening	335°CA ATDC Firing
Intake Valve Closing	-143°CA ATDC Firing
Exhaust Valve Opening	130°CA ATDC Firing
Exhaust Valve Closing	-355°CA ATDC Firing

Table 2. Piston properties

Piston Bowl Shape	Open Crater	Bathtub
Design	Stock CDC	Modified for RCCI
r_c (-)	16.1	14.9:1
Undercut Blank ²	No	Yes
Combustion Mode	CDC	RCCI
Relative reduction in surface area vs. open crater (%)	N/A	4.25

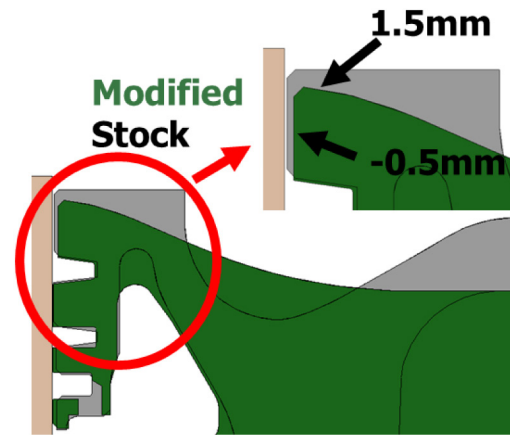


Figure 2. Cross sectional view of the stock CDC $r_c = 16.1:1$ (grey), and the optimized RCCI bathtub ($r_c = 14.9:1$ green) pistons. Note the undercutting of the top ring relative to the CDC stock piston where numbers and arrows denote the difference of the crevice squish height and piston-liner clearance in the modified geometry relative to the stock piston.

As seen in Table 2, the surface area of the RCCI piston is reduced as compared to the stock piston. For both pistons, the compression ratio was determined through examination of computer solid models coupled to the linearity fit of log indicated cylinder pressure-volume measurements.

For all tests, the loss angle of motored peak cylinder pressure was fixed at -0.4°CA ATDC . Acquired cylinder pressure traces were averaged for 500 cycles, and post-processed using a zero-phase low-pass Butterworth filter with the cutoff frequency at the first resonance mode of the respective engine, as described by Eng [21]. PPRR was calculated from the cylinder pressure ensemble averages. Combustion noise was calculated by passing the unfiltered pressure of individual cycles through an acoustic filter tuned for the human ear's frequency response using the relations described in [38], where corresponding combustion noise was reported as the ensemble average. The apparent heat release rate (AHRR) was calculated from the pressure derivative obtained by differentiating the ensemble average pressure using a first order Savitzky-Golay filter with a moving window size of nine (window of $\pm 0.5^\circ\text{CA}$ about the midpoint with 0.1°CA encoder resolution). The AHRR calculation also used a non-constant gamma approach, where the measured constant compression gamma was used prior to top dead center (TDC) and the measured constant expansion gamma was used after TDC. CA50 was characterized by the location of 50% apparent heat release, which was determined by integrating the AHRR.

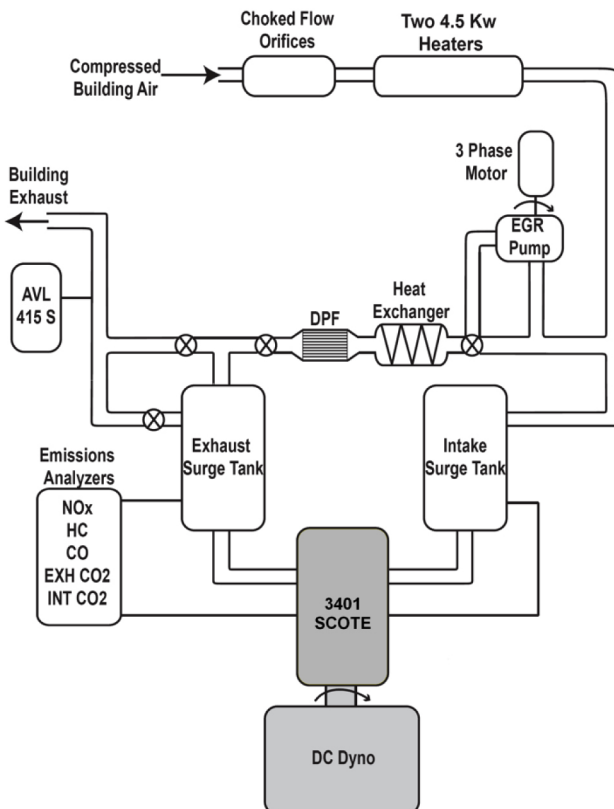


Figure 1. Diagram of the engine lab

2. RCCI pistons cut from research blanks with 1 (mm) undersized top ring lands, further detail in Splitter et al. [37]

All gaseous engine-out emissions were analyzed with a five-gas Horiba analyzer bench and smoke was measured with an AVL 415S smoke meter. HC emissions were corrected using the Kar and Cheng ethanol flame ionization detector (FID) response correlation [39].

Fuels and Fuel Delivery

The fuels used were splash blended on site. The low reactivity fuel used was E85, determined through blending a measured volume of 85% ethanol with a separate measured volume of 15% gasoline; this blend comprised the port injected fuel. The high reactivity fuel was cetane-improved gasoline, where 3.5% by volume of EHN was added to gasoline. The unblended fuel properties are listed in Tables 3, 4, 5.

Table 3. Ethanol fuel properties

Lower Heating Value (MJ/kg)	26.9
AKI (RON+MON)/2	99.15 [40] ³
H/C ratio (-)	3.0
O/C ratio (-)	0.5
Specific gravity at 60°C (-)	0.789

Table 4. 2-Ethylhexyl-Nitrate fuel properties

Lower Heating Value (MJ/kg)	27.364 [41] ⁴
H/C ratio (-)	2.125
O/C ratio (-)	0.375
Specific gravity at 15.5°C (-)	0.96

Table 5. Gasoline fuel properties

Lower Heating Value (MJ/kg)	44.301
RON (-)	94.7
MON (-)	87.8
AKI (RON+MON)/2	90.9
Ethanol Content (%)	0
H/C ratio (-)	1.88
O/C ratio (-)	0
Specific gravity at 60°C (-)	0.7009

Fuel mass flow rates were directly measured using a coriolis meter for the low-reactivity fuel and a differential pressure-based measurement for the high-reactivity fuel. The total fuel amount was verified against fuel flow rates calculated from emissions bench values. High-reactivity fuel was delivered through a common rail operated at relatively low pressure, with the injector rail pressure and injection schedule shown in Table 6.

Table 6. High reactivity fuel injector and schedule

Actuation Type	Solenoid
Number of Holes	6
Hole Diameter (μm)	250
Included Spray Angle (°)	145
Fuel Rail Pressure (bar)	500
Injection Schedule	Double
First Start of Injection Timing (°CA ATDC)	-60
First Injection DI fuel amount (%)	~65
Second Start of Injection Timing (°CA ATDC)	-35
Second Injection DI Fuel Amount (%)	~35

The low reactivity fuel was delivered through a high flow-rate standard peak-and-hold design port-fuel injector (PFI) with dimensions and injection schedule depicted in Table 7.

Table 7. Low reactivity fuel injector and schedule

Injector Style	peak and hold
Steady flow rate @ 3 bar (cc/min)	750
Included Spray Angle (°)	15
Fuel Pressure (bar)	4
Injection Schedule	Single
Start of Injection Timing	-320°CA ATDC

Fuel Selection Rationale

E85 was selected for the low-reactivity fuel, due to it being a representative market-available high-octane fuel. A high-octane fuel was desired because it expands the range of intake temperatures and pressures for the current study, and ethanol has a reduced fuel pressure sensitivity [29,30], expanding the operational window [42].

3.5% EHN doped into 91 AKI gasoline was selected as the high-reactivity fuel because the high volatility of gasoline increases vaporization processes, minimizing the potential of fuel-liner and fuel-piston impingement, which could unfavorably influence the results [42]. Recently, Dempsey et al. [32] operated the identical fuels with HCCI. That work demonstrated that the 3.5% EHN in the identical 91 AKI gasoline resulted in an AKI of ~65, which according to previous work by Kalghati [43] corresponds to a cetane number of ~28. Thus, the reactivity of the doped gasoline was less than that of #2 ultra-low sulfur diesel but was higher than that of gasoline.

3. AKI of Ethanol estimated to be 99.15 by Eyidogan et al. [40]

4. EHN properties obtained from Oxley et al. [41]