

Boosting Simulation of High Efficiency Alternative Combustion Mode Engines

2011-01-0358

Published
04/12/2011

Christopher Chadwell, Terrence Alger and Charles Roberts
Southwest Research Institute

Steven Arnold
Engine Systems Innovation Inc.

Copyright © 2011 SAE International

doi:10.4271/2011-01-0358

ABSTRACT

Four high-efficiency alternative combustion modes were modeled to determine the potential brake thermal efficiency (BTE) relative to a traditional lean burn compression ignition diesel engine with selective catalytic reduction (SCR) aftertreatment. The four combustion modes include stoichiometric pilot-ignited gasoline with EGR dilution (SwRI HEDGE technology), dual fuel premixed compression ignition (University of Wisconsin), gasoline partially premixed combustion (Lund University), and homogenous charge compression ignition (HCCI) (SwRI Clean Diesel IV). For each of the alternative combustion modes, zero-D simulation of the peak torque condition was used to show the expected BTE.

For all alternative combustion modes, simulation showed that the BTE was very dependent on dilution levels, whether air or EGR. While the gross indicated thermal efficiency (ITE) could be shown to improve as the dilution was increased, the required pumping work decreased the BTE at EGR rates above 40%. None of the alternative combustion modes was able to exceed the BTE of a traditional lean burn diesel engine with EGR when calibrated to 2.7 g/kW-h engine-out NO_x when constrained by currently available turbocharger efficiency.

INTRODUCTION

Greenhouse gas regulation, political pressure for energy security, and consumer demands due to the high cost of petroleum have motivated the engine industry to seek more efficient solutions than the standard gasoline and diesel

engines available today. Additionally, increasing demand for diesel fuel in developing nations, and the relative softening of the demand for gasoline, has led to increased research to find alternative uses for the efficient combustion of gasoline. While efficiency potential is important, the emission of criteria pollutants cannot be compromised to gain efficiency.

Future combustion technology research often takes place on research style engines that can easily be disassembled to change out components. If the combustion technology requires pressurized inlet conditions, the pressure almost always comes from an electrically driven supercharger or external air source. Turbochargers are not used because the turbocharger match is contingent on lambda (Eq. 1), EGR rate (Eq. 2), and exhaust temperature. During the early research period of new technologies, those parameters likely change often and would require constant rematching of the turbo machinery. Also, when single cylinder engines are used, turbocharging does not work well with the long period between exhaust pulses.

$$\lambda = \frac{AFR}{AFR_{stoichiometric}} \quad (1)$$

$$EGR\ Rate = \frac{\dot{m}_{EGR}}{\dot{m}_{air} + \dot{m}_{EGR}} \quad (2)$$

One of the problems with using external air sources is that the viability of the lambda and EGR requirements is hard to quantify. This can lead the research in a direction that may not be viable when the engine is required to move its own air via turbocharging or supercharging. For example, if

traditional lean-burn diesel combustion without EGR is being investigated, the researcher will observe that generally the larger the lambda value, the greater the measured ITE, while the measured exhaust temperature decreases during this exercise. One might conclude that running the engine with very high lambda values would result in optimized BTE. This conclusion fails to account for the exhaust manifold pressure that is required for the exhaust turbine to generate enough power to drive the compressor to the desired pressure ratio. In reality, there is an optimum lambda that balances the pressure ratio requirement, exhaust temperature, and map operating point (isentropic efficiency) of the compressor and turbine. With a reduced lambda, the engine may not have the highest observed ITE, but would have the optimized (highest) BTE.

In this paper, several high efficiency combustion concepts were evaluated on a model of a multi-cylinder engine at peak engine torque conditions. The modeling approach is similar to Caton's approach [1]. Caton applied a high efficiency combustion concept that was published to provide very high ITE to a model of a multi-cylinder engine to predict the BTE. He found that while the ITE can indeed be very good, the BTE remained similar to or less than current diesel technology. The analysis in this paper differs from Caton's work in engine choice and engine load. A model of a diesel engine representative of current state of the art technology was used in this analysis, and the engine load was run at 20 bar BMEP, or the stated maximum BMEP of the combustion concept if it was less than 20 bar.

REVIEW OF HIGH EFFICIENCY COMBUSTION CONCEPTS

To derive the input data for the model, a literature review of high efficiency combustion concepts was performed. Additionally, high efficiency research work at SwRI was included in the analysis. The following sections summarize the combustion concepts.

SWRI HEDGE TECHNOLOGY WITH PILOT IGNITION

The HEDGE consortium at SwRI has focused on the use of cooled EGR and high energy ignition to achieve high efficiency and low emissions from a flame propagation gasoline engine. There are two embodiments of a HEDGE engine that have been studied; a cost-effective spark ignited version intended for passenger car and truck applications [2] [3][4], and a medium duty diesel pilot ignited version intended as a medium or heavy duty diesel replacement[5]. For this paper, the diesel pilot ignited version was used because the additional EGR tolerance should produce better BTE than the spark ignited embodiment.

The research engine used to develop the HEDGE technology was based on a John Deere 4045, a 4-cylinder 4.5L medium

duty diesel engine. The engine was modified with reduced compression ratio and modified combustion chamber pistons, a flow-balanced port-injection intake manifold, and a cooled EGR loop. The engine specifications are shown in Table 1. A characteristic of this HEDGE embodiment is that the engine requires standard diesel and at least a 92 pump octane gasoline, so no special fuel formulations are required.

Table 1. Engine specifications for demonstration of the medium-duty HEDGE combustion concept

Test engine	John Deere 4045
Displacement	4.5L (inline 4)
Bore x Stroke	106 mm x 127 mm
Compression ratio	14.5
Fuel	Haltermann EEE gasoline (92.45 (R+M)/2) with diesel pilot
Ignition	Compression ignition of a diesel pilot injection
AFR	Stoichiometric (lambda = 1)
EGR (rate)	Low pressure loop, cooled (>45% for knock mitigation)
Induction	Supercharger and VG Turbocharger (series)

SWRI MULTI-CYLINDER HCCI

HCCI research has been a part of the Clean Diesel consortium at SwRI since 1998 [6]. Various HCCI embodiments have been investigated including a multi-cylinder engine with real time blending of natural gas and FT-Naphtha [7], a single-cylinder variable swirl engine with direct and port injection, and in the final iteration, a multi-cylinder engine with an engine-driven supercharger and turbocharger [8]. The final iteration multi-cylinder is the source of the data for this paper.

The engine used port fuel injection to ensure the greatest possible homogeneity. Port injection requires sufficient volatility to work properly, so a fuel with gasoline-like volatility was used. Gasoline, however, had too little low temperature heat release (LTHR) to work at low engine loads, so an HCCI specific fuel was used. The fuel was a blend of 67% Haltermann EEE gasoline (92.45 R+M/2) and 33% light straight run naphtha which resulted in an 80 octane (R+M/2) fuel. While this fuel was blended with commercially available components, the commercial viability of the fuel was not evaluated. The expectation would be that solutions based on currently available pump gasoline and diesel would be preferred.

The HCCI engine was based on a John Deere 4045, a 4-cylinder 4.5L medium duty diesel engine. The engine was modified with reduced compression ratio and modified combustion chamber pistons, a flow-balanced port-injection

intake manifold, and a cooled EGR loop. The direct injection fuel system was removed to install cylinder pressure sensors. As this engine was the culmination of the HCCI research, an air handling system was built that could provide the required air and EGR. Because of the high EGR dilution and the resulting low exhaust temperature, a combination of a supercharger and turbocharger was required. The engine specifications are shown in Table 2. A layout of the boosting system is shown in Figure 1.

Table 2. Engine specifications for the multi-cylinder HCCI engine developed at SwRI

Test Engine	John Deere 4045
Displacement	4.5L (inline 4)
Bore x Stroke	106 mm x 127 mm
Compression Ratio	14.5
Piston Style	Low squish dog-dish
Fuel	67% Haltermann EEE gasoline, 33% LSR Naphtha, 80 ON (R+M)/2
Injection Type	Multi-port fuel injection
Induction	Supercharger (Eaton M24) and turbocharger (Honeywell GT1541V)
EGR	High-pressure loop, cooled

fly” from standard diesel and pump gasoline. With this combustion concept, gasoline fuel is port injected while diesel fuel is direct injected. Since diesel fuel and gasoline have significantly different cetane numbers, the total cetane number of the mixed fuel can be adjusted by changing the relative ratio of diesel and gasoline. When the fuel is combusted with excess air and high EGR rates, the engine was shown to produce very low NOx, very low soot, and produce very high net ITE.

The Engine Research Center's research engine is a single cylinder Caterpillar 3401E. The engine specifications are shown in Table 3, and a schematic of the engine is shown in Figure 2. The source of air for the engine is an external building air source, while the EGR is circulated with an EGR pump if necessary. This setup allowed flexible independent control of the air and EGR, and presumably was quite useful while characterizing the combustion concept. A downside of this configuration is EGR can likely be circulated without a negative pressure gradient on the engine (exhaust manifold pressure greater than intake manifold pressure) which divorces the combustion concept from the reality of turbocharging requirements. Another downside is that BTE measurements were not possible, so only net ITE measurements were reported.

Table 3. Engine specifications for the Caterpillar 3401E SCOTE engine used by the Engine Research Center at the University of Wisconsin to research the RCCI combustion concept

Test Engine	Caterpillar 3401E SCOTE
Displacement	2.44L (single cylinder)
Bore x Stroke	137 mm x 165 mm
Compression Ratio	16.1
Piston Style	OEM bowl geometry
Fuel	Gasoline and Diesel
Injection Type	Multi-port gasoline injection, HEUI 315B or Bosch Gen 2 common rail DI diesel
Induction	External air source
EGR	High-pressure loop, cooled

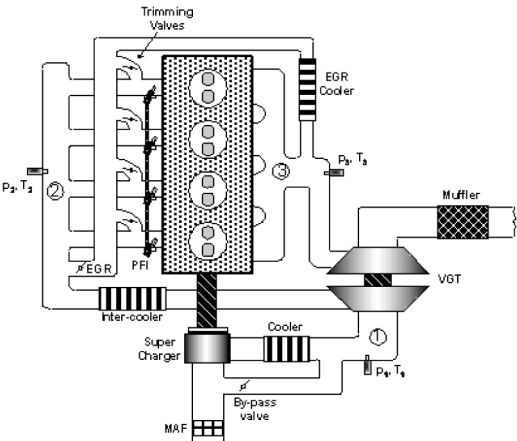


Figure 1. Schematic of the multi-cylinder HCCI engine

REACTIVITY CONTROLLED
COMPRESSION IGNITION (RCCI)
(DUAL-FUEL PCCI)

The University of Wisconsin's Engine Research Center has published several papers describing a combustion concept where gasoline and diesel fuel are mixed in cylinder [9][10]. The concept was initially called dual-fuel premixed charge compression ignition (PCCI), but has since been titled reactivity controlled compression ignition (RCCI). While the combustion concept requires fuel of a special reactivity to function, the RCCI concept is able to make the fuel “on the

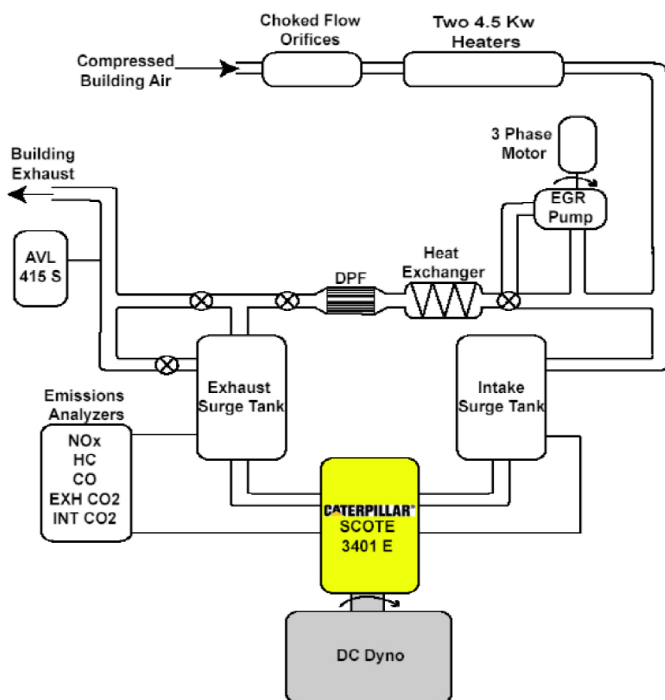


Figure 2. Schematic of the Caterpillar 3401E SCOTE engine used by the Engine Research Center at the University of Wisconsin to research the RCCI combustion concept

GASOLINE PARTIALLY PREMIXED COMBUSTION

Lund University has published several papers describing a combustion concept where gasoline-like fuels are partially premixed in a compression ignition engine [11][12]. The injection strategy at high load is not described in explicit detail, but the basic concept is to use fuels with gasoline-like volatility and partially premix some of the fuel into a highly dilute environment. The balance of the fuel is injected near TDC at typical diesel start-of-injection timing. The advantages of this type of combustion are low soot, low NO_x, and a fast burn rate which leads to very high ITE. A downside of the concept is that it appears to work best with non-conventional fuels. Unlike the HEDGE and RCCI combustion concepts that use fuels readily available at the pump, the fuels for this concept appear to be special blends of components with gasoline like volatility but varying RON and MON.

Lund University's research engine is a Scania 13L inline-6 engine modified to run as a single cylinder. The engine specifications are shown in Table 4. The engine was boosted using an external air source with a valve to control pressure. The EGR was mixed into the inlet air, and the air and EGR mixture was heated to 30C before entering the engine. Similar to the University of Wisconsin's setup, Lund

University's combustion concept is divorced from the reality of turbocharging requirements via the use of external air. BTE measurements were not possible, so only gross ITE measurements were reported.

Table 4. Engine specifications for the modified Scania D13 engine used by Lund University to research the gasoline partially premixed combustion concept

Test Engine	Scania D13 (modified to run as a single)
Displacement	2.12L (single cylinder)
Bore x Stroke	130 mm x 160 mm
Compression Ratio	17.3
Piston Style	Custom shallow bowl
Fuel	Custom gasoline-like blend
Injection Type	XPI common rail (2400 bar capable)
Induction	External air source
EGR	High-pressure loop, cooled

CONVENTIONAL DIESEL COMBUSTION WITH SCR AFTERTREATMENT

The conventional diesel engine has been the engine of choice when high efficiency, especially at high specific power, was required. Unfortunately, NO_x and PM emission constraints tend to force non-optimum injection timing and consume extra fuel for DPF regeneration. The trend has somewhat reversed with the introduction of NO_x aftertreatment. With NO_x aftertreatment, the engine's allowable NO_x has been increased which allowed more advanced start-of-injection which will improve the BTE. Also, when the SCR is located downstream of the DPF, the elevated NO_x levels passing through the DPF provide more passive regeneration which reduces the need for active regeneration and the associated fuel penalty. Since the traditional lean-burn diesel engine with aftertreatment represents the mainstream technology path for heavy-duty engines, lean-burn diesel combustion was included in this analysis.

The input data for the conventional diesel engine model came from a Caterpillar C15 that has been run in various configurations at SwRI. The engine was run to characterize the air-to-fuel ratio (AFR), EGR rate, and SOI timing required to reduce the engine out NO_x to 2.7 g/kW-h which was assumed to be an appropriate engine out NO_x concentration to meet US2010 "Not to Exceed" NO_x emission standards. This engine was also used to collect data to calibrate the FMEP and PMEP aspects of the model for use in the rest of the analysis.

Table 5. Engine specifications for the Caterpillar C15 used to represent a conventional diesel combustion engine

Test Engine	Caterpillar C15
Displacement	15.2L (inline 6)
Bore x Stroke	137 mm x 172 mm
Compression Ratio	17
Piston Style	OEM bowl geometry
Injection system	2800 bar capable MUEI-C
EGR	High-pressure loop, cooled
Heat release profile	Measured on the engine

Table 6. Caterpillar C15 engine model parameters used in the A_E code

Test Engine	Caterpillar C15
Displacement	15.2L (inline 6)
Bore x Stroke	137 mm x 172 mm
Compression Ratio	Varies by combustion concept
Piston Style	OEM bowl geometry
Induction	Isentropic turbocharger model
EGR	High-pressure loop, cooled
Heat release profile	Sourced from combustion concept

MODEL SETUP

The zero-D code, ALAMO_ENGINE (A_E), has been used at SwRI for 20 years to simulate all types of engines. The A_E code is a two-zone PC-based code with a Windows™ interface, and is used to compute power, efficiency, and NO_x emissions for diesel, spark-ignited, stoichiometric and lean-burn gasoline engines, and lean-burn natural gas engines. The A_E code is a simulation code that includes detailed gas composition data to accurately include the effects of EGR, residual gases, fuel composition, water from humidity in intake air, water in fuel-water emulsions, and water injected in-manifold or in-cylinder. The model also includes a complete chemical equilibrium code to compute chemical species in the combustion gases, and kinetics for the formation of nitric oxide. The model includes a heat transfer correlation from Woschni and a friction model based upon Chen-Flynn.

In this paper, the BTE potential of several high efficiency combustion concepts was evaluated on a common engine model using the A_E code. The purpose of using the common engine model was to evaluate the combustion concepts with the same displacement, breathing, and boosting hardware so that a proper comparison between the concepts could be performed. The engine that was modeled was a Caterpillar C15. The engine specifications are shown in Table 6. While collecting the data for the conventional diesel combustion, the Chen-Flynn friction coefficients (FMEP) and breathing (PMEP and volumetric efficiency) in the model were adjusted to match measured data. The model was run at 1250 rpm, and the input parameters were adjusted to achieve the stated maximum loads which varied from 7-bar BMEP to 20-bar BMEP depending on combustion concept.

For each of the combustion concepts, the input parameters that distinguished the concepts from each other were the compression ratio, AFR, EGR rate, and cumulative heat release profile. Also, when published, any special valve timing events were incorporated in the model. Each of the input parameters was taken from the respective publications. The heat release rate for each combustion concept, typically shown as an instantaneous heat release plot, was determined by digitizing the published heat release rate and selecting points from the curve. The in-cylinder heat transfer coefficients based on the Woschni model were left unchanged; however when cylinder pressure traces were published, a global heat transfer multiplier was used to match the model predicted and published pressure traces. An important parameter that could not be ascertained from the published data was the combustion efficiency (% of fuel that is converted to complete products of combustion). Therefore a combustion efficiency of 99.4% was used for all combustion concepts. 99.4% combustion efficiency is appropriate for diesel combustion, but may be too high when describing combustion concepts that require high EGR dilution, low combustion temperature, or a significant amount of premixing. Therefore the results for the highly dilute combustion modes may be slightly optimistic.

The turbocharger model in A_E was a simple single-stage isentropic compressor and turbine model with prescribed efficiency. The compressor isentropic efficiency was always set to a conservative 75%. The intake manifold pressure (MAP) was set in the model to achieve the desired BMEP. The model calculated the required compressor power to achieve the desired MAP, and then calculated the required exhaust manifold pressure to realize the appropriate turbine power at the model predicted turbine inlet temperature. When run like this, excess air or EGR dilution that lowered the turbine inlet temperature while increasing the MAP (compressor pressure ratio) was penalized with increased exhaust manifold pressure and the associated PMEP.

One of the desired results from this analysis was the sensitivity of engine BTE to overall turbocharger efficiency (Eq. 3), EGR rate requirement, and excess air requirement.

To generate the data for the analysis, the turbine isentropic efficiency was swept from 50% to 93.3% (40% to 70% overall isentropic turbocharger efficiency when combined with a compressor isentropic efficiency of 75% and 100% mechanical efficiency), and the EGR rate was swept around the stated EGR requirement. The AFR was held at the stated AFR since it was assumed that changing the AFR would change the fundamental operation of the combustion concept.

$$\text{Overall Turbocharger Efficiency } (\eta_{\text{turbo}}) = \eta_{\text{compressor}} \cdot \eta_{\text{turbine}} \quad (3)$$

In [Figure 3](#), the normalized cumulative heat release rates for all of the combustion concepts are shown. For most of the combustion concepts, the heat release is occurring in an optimum location. The RCCI and HCCI heat release profiles were very similar in shape; both profiles have pronounced low temperature heat release (LTHR) that is usually observed with HCCI. The standard diesel combustion is slow in comparison to the high efficiency concepts. That explains some of the ITE deficiency with diesel; the combustion phasing is less ideal than the high efficiency combustion concepts. Surprisingly the HEDGE combustion concept, which is initially slower burning than the diesel, shows what appears to be very short 50-90 burn duration.

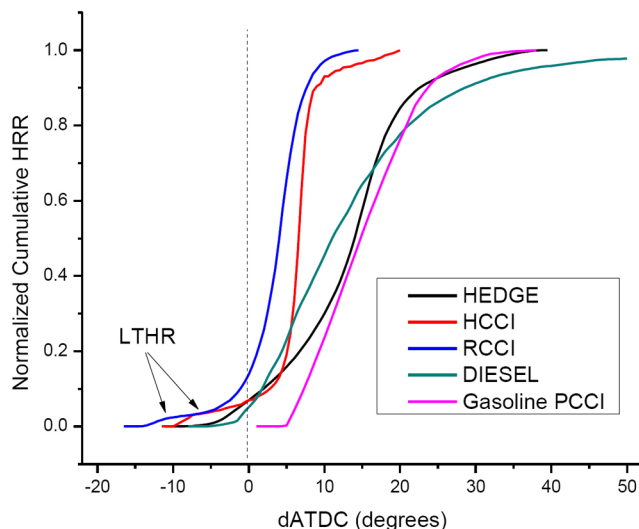


Figure 3. Comparison of the normalized cumulative heat release rate profiles from all of the combustion concepts.

ZERO-D CYCLE SIMULATION RESULTS

As stated in the previous section, the objective of the simulation was to determine the BTE potential of the combustion concepts, as well as to determine the sensitivity of BTE to the EGR rate and turbocharger efficiency. To achieve that objective, the results from the model are presented as 2-D contour plots of BTE. The x-axis for each of

the plots is the EGR rate, and the y-axis is overall isentropic turbocharger efficiency. The EGR rate was swept beyond the stated EGR requirement and the isentropic turbocharger efficiency were swept beyond reasonable values. This was done to be able to understand the BTE sensitivity to these quantities.

Rectangles were drawn in to bound reasonable isentropic turbocharger efficiency and the combustion concept published EGR rate requirement. The BTE contours present at the intersection of the rectangles show the probable BTE of the combustion concept. The bound for the possible turbocharger efficiency was chosen to be 50%. With two-stage series boosting and inter-stage cooling it is possible to exceed 50% at a single point; however it has not been shown to be easily achievable to exceed 50% across the range of operation. In order to evaluate the viability of the combustion concepts, it was decided that 50% represented an optimistic isentropic turbocharger efficiency that could be achieved at the lug curve conditions.

The model results for the RCCI combustion concept are presented differently. For the RCCI research, the highest load data that was published at the time this paper was authored was 9-bar IMEP (roughly 7 bar BMEP on a multi-cylinder). At this condition, the intake pressure was 1.74-bar, which yields an exhaust lambda of 2.0. As this lambda value is quite high, the required boost to extrapolate this operating condition to higher loads is unrealistically high. It would not be appropriate to assume how the lambda value should be reduced with increasing load; the engine must be used to determine the required lambda value at high load. Therefore the RCCI model was only evaluated at 9-bar IMEP. Instead of creating BTE contours, the model was exercised to predict the BTE at this condition at various isentropic turbocharger efficiency, and to show how the BTE would change if lambda 2.0 was required with increasing load.

SWRI HEDGE TECHNOLOGY WITH PILOT IGNITION

The pilot ignited HEDGE engine runs on 92.45 pump octane gasoline with a 14.5 compression ratio. In order to maintain optimum combustion phasing without knock, the engine requires 45 to 53% cooled EGR. Additionally, in order to minimize PM emissions, 53 to 57% cooled EGR is required. During testing, it was observed that the boosting requirements to run the highest loads precluded single stage boosting, so two stage boosting has been used. Even though the two stage boost system can efficiently make high pressure ratio to support high airflow and EGR rates, the spark ignition-like burn profile (significant heat release before top dead center) causes high cylinder pressure, so the engine's peak BMEP was limited to 18 bar.