INVESTIGATION INTO THE EMISSIONS AND EFFICIENCY OF LOW TEMPERATURE DIESEL COMBUSTION

A Thesis

by

BRYAN MICHAEL KNIGHT

Submitted to the Office of Graduate Studies of
Texas A&M University
in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

August 2010

Major Subject: Mechanical Engineering

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Approved by:

Chair of Committee,	Timothy Jacobs
Committee Members,	Jerald Caton
	Jorge Alvarado
Head of Department,	Dennis O'Neal

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ABSTRACT

Investigation into the Emissions and Efficiency of Low Temperature Diesel Combustion. (August 2010)

Bryan Michael Knight, B.S., Texas A&M University

Chair of Advisory Committee: Dr. Timothy J. Jacobs

As global focus shifts towards the health and conservation of the planet, greater importance is placed upon the hazardous emissions of our fossil fuels, as well as their finite supply. These two areas remain intense topics of research in order to reduce green house gas emissions and increase the fuel efficiency of our vehicles. A particular solution to this problem is the diesel engine, with its inherently fuel-lean combustion, which gives rise to low CO_2 production and higher efficiencies than its gasoline counterpart. Diesel engines, however, typically exhibit higher nitrogen oxides (NOx [NOx = NO + NO_2, where NO is nitric oxide and NO_2 is nitrogen dioxide]) and soot.

There exists the possibility to simultaneously reduce both emissions with the application of low temperature diesel combustion (LTC). While exhibiting great characteristics in simultaneous reductions in nitrogen oxides and soot, LTC faces challenges with higher carbon monoxide (CO) and hydrocarbon (HC) emissions, as well as penalties in fuel efficiency.

The following study examines the characteristics of LTC which contribute to the differences in emissions and efficiency compared to typical conventional diesel combustion. More specifically, key engine parameters which are used to enable LTC, such as EGR and fuel pressure are swept through a full range to determine their effects on each combustion regime. Analysis will focus on comparing both combustion regimes to determine how exhaust gas recirculation (EGR) and fuel pressure relate to lowering NO and smoke concentrations, and how these relate to a penalty in fuel efficiency.

This study finds that the application of LTC is able to realize a 99% reduction in NO while simultaneously reducing smoke by 17% compared to the conventional combustion counterpart. Through a sweep increasing EGR, LTC is able to defeat the typical soot – NO tradeoff; however, brake fuel conversion efficiency decreases 6.8% for LTC, while conventional combustion realizes a 4% increase in efficiency. The sweep of increasing fuel pressure confirms typical increases in NO and decreases in smoke for both LTC and conventional combustion; however, brake fuel conversion efficiency increases 2.3% for LTC and drops 4% for conventional combustion.

DEDICATION

To Laura

ACKNOWLEDGEMENTS

This is one of the most important sections of this whole thesis. This section is meant to bring people into the spotlight that most deserve it, and give back to them a little bit of what they have given to me. I am so often touched and guided by others in my life, but I rarely get the chance to reflect and realize their importance. Even rarer do I get a chance to tell them what an honor it is to have them in my life.

Dr. Jacobs has been a wonderful light in my life that has developed both my professional career and my personal life. His guidance throughout my experience in graduate school has been invaluable. One of the most wonderful things is that he always makes time, especially when I would come to him in need. I cannot understate the compassion he has for his students, and it really shows in his teachings, his manner, and his friendship. Thank you Dr. Jacobs for your mentorship through this stage in my life.

Josh Bittle is a fellow graduate student who has amazing patience, an even more amazing thirst for learning, and a great passion for teaching. Through his lessons, whether it is working on fluids homework in a hotel in San Antonio, or wiring up DAQ equipment in a cabinet, Josh has helped make my graduate school experience a little easier. Even more, he has helped show me the joy of teaching others, for that I thank you.

To the other gentlemen and ladies in the research lab, thank you for your camaraderie, friendship, and support.

I honestly would not be where I am today if it were not for the guidance, support, and love from my family. To Michelle, I want to thank you for being such a wonderful little sister. This may not sound important, but you taught me some of the most valuable skills possible. Being the older brother, I wanted to be mature, fair, and equal with you and Jason. You taught me how to work with people, and it is a something that I hold so valuable in my life. I strived to be the brother that you would respect, and it made me a better person for it. To Jason, I want to thank you for always pushing my competitive side. I never wanted to let my little brother beat me in anything, especially in school. You have pushed me to work harder and accomplish more than even I thought I could do. Now, you've become my role model with your wonderful family, and someday I hope that I can follow your footsteps. I love that we are able to help each other out through the road of life. To my mom, I wish I could thank you with something more than words for the selfless love that you have given me throughout my life. You are someone who I can always trust to be there for me, someone who will always listen. While this may not make the cover of Times magazine, it is an attribute that makes you wealthier than any human being can ever become. I know it is something that I can never repay to you, but I hope the person that I have become, and the things that I accomplish can begin to show you how appreciative I am. And to my dad, you have been the North Star of my life. Your emphasis and respect for education is what has led me to where I am today, and I thank you for all of your teachings. You have let me make my own path through life, but always with the honor and integrity that you raised me with. I think the most important lesson that you taught was the unspoken one, one in which you were a perfect role model for me to follow as I grew up, and now as I progress through life. Thank you!

Finally, I want to thank the love of my life and my best friend, Laura. I have dedicated this thesis to you because of the sacrifices that you have made to make it possible. You selflessly supported my desire to go to graduate school, and you were there to remind me of my passion for learning when I felt like giving up. Your influence makes me strive to do my best and work my hardest (even when I didn't want to hear that you were disappointed on the day that I didn't work on my thesis). You are my rock in tumultuous water, and I would not be able to be the person that I am today without you.

NOMENCLATURE

dATDC	Degrees after top dead center
BMEP	Brake mean effective pressure
CO	Carbon monoxide
EGR	Exhaust gas recirculation
EMP	Exhaust manifold pressure
EMT	Exhaust manifold temperature
FMEP	Friction mean effective pressure
$IMEP_g$	Gross indicated mean effective pressure
$IMEP_n$	Net indicated mean effective pressure
IMP	Intake manifold pressure
IMT	Intake manifold temperature
HC	Hydrocarbon(s)
NO	Nitric oxide
NO_x	Nitrogen oxides
PM	Particulate matter
PMEP	Pumping mean effective pressure

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1. INTRODUCTION

1.1 Motivation

A substantial portion of the United States' fuel consumption and carbon dioxide emissions stem from our vehicles and our transportation. A recent report shows that the transportation industry represented 30% of the United States' energy consumption and is responsible for producing 31% of the carbon dioxide emissions in America [1]. With recent focus shifting towards greenhouse gasses, CO_2 emissions are becoming increasingly important.

Diesel combustion systems, which currently represent less than 3% of transportation-based energy converters, are a readily available technology that can improve our nation's energy consumption and CO_2 emission rates [2]. Diesel engines, as part of their inherit fuel-lean combustion, produce far less carbon emissions than their gasoline equivalents. With better efficiency and lower carbon emissions, greater prevalence of diesel engines in the transportation sector can help solve these current issues. However, in spite of these efficiency and CO_2 benefits, diesel engines face challenges with emissions of nitrogen oxides (NO_x), particulate matter (PM), hydrocarbons (HC), and carbon monoxide (CO).

1.2 Background: Air Quality Emissions

Nitrogen oxides, or NO_x, combined with unburned HC in the presence of sunlight react to form ground-level ozone. This ground-level ozone is the primary constituent of smog, and is responsible for both health and environmental problems. Ground-level ozone can trigger health problems that include chest pain, coughing, respiratory irritation, and congestion. Ozone reduces lung function, inflames the lining

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of the lungs, and longer exposure may scar lung tissue. Plants, vegetation, and crops can be damaged by exposure to ground-level ozone making them more susceptible to diseases and produce lower yields [3]. As of 2008, the EPA states that motor vehicles are responsible for more than half of the NO_x emissions in the United States. Of these emissions, diesel engines in heavy duty trucks and buses are responsible for 33% of the transportation emissions, even though diesel engines represent only 3% of the transportation-based energy converters [4].

PM is most well known by the black smoke that is emitted from diesel vehicles. It consists of combustion generated carbonaceous material, also known as soot, which absorbs organic compounds from the combustion process [5]. The EPA is concerned with inhalable coarse PM particles between 2.5μm and 10μm which are directly linked to a potential for health effects, and can increase respiratory symptoms, decrease lung function, and aggravate asthma. Fine PM particles such as soot smaller than 2.5 μm are responsible for visibility reductions, or haze. As of 2008, heavy-duty trucks and buses using diesel engines are responsible for 25% of the transportation PM emissions in the US.

CO, a component of motor vehicle exhaust, is a colorless and odorless gas emitted by combustion into the atmosphere. When inhaled, it can cause harmful health effects by reducing oxygen absorption in the blood stream, ultimately reducing oxygen that is delivered to the body's organs. Low levels of CO can affect those with heart disease, while higher levels can affect the nervous systems of healthy people causing vision problems, reduced cognitive abilities, and difficulty performing complex tasks. Very high levels can result in death [6]. Motor vehicles are responsible for 75% of CO emissions nationwide [3].

In order to limit the amount of hazardous emissions polluting the atmosphere, the United States Congress passed the Clean Air Act (CAA) of 1970 which was signed into action by Richard Nixon. The first vehicle exhaust emissions were established, and updates to the act have continued to reduce the emissions of harmful compounds. Even more recent, stricter emissions standards implemented by the EPA are aimed at reducing emissions from both on road and off-road diesel vehicles by more than 90% [3]. With the CAA in effect, actual national averages from 1980 to 2008 of concentrations such as: NO_2 (a primary constituent of NO_x) has decreased 46% [7], ground-level ozone has decreased 25% [8], PM (10 μm) has dropped 31% [9], and CO has decreased 79% [10]. However, the total miles traveled by people in the US has increased 178% from 1970 to 2005 and continues to increase 2% to 3% every year. There are over 210 million cars and light-duty trucks on the road in the US. As time goes by and more vehicles are on the road for longer periods of time, air quality concerns will remain very important.

1.3 Why Diesel Engines?

Rudolf Diesel, born in Paris in 1858, can be touted as the father of the diesel engine. Having a love for engine design, he created many types of heat engines before he filed for a patent in 1894 on his new invention, the diesel engine. The first successful diesel engine was completed and operated in 1897. By the next year, Rudolf Diesel was a millionaire and his engines were used extensively, rapidly replacing the competitor at the time, the steam engine [11].

In the present day, the diesel engine has seen a rise and fall of sales. Steadily climbing from 1998 to an impressive growth period from 2002 to 2007, the world demand for diesel engines has decelerated since, and is expected to grow slower at three percent per year through 2012 to a market of $160 billion [12]. Growth in the North American market is dominated by demand for diesel in the United States, where an increase in diesel engines used in light vehicles is expected to increase from the 3.6% market share in 2004 [13]. However, diesel powered automobiles have faced challenges, both from the purchasing power of the consumer and the pollutants emitted from the exhaust.

A 1988 study found that diesel car sales rose from less than 1% of new car sales in 1976 to 6% in 1981, but collapsed back to less than 1% in 1985. A survey was conducted and found that consumers relied on per gallon fuel prices, not fuel costs per mile as the indicator of money savings. The fall of the diesel came when the per gallon gas price advantage of diesel fell compared to gasoline. Ultimately fuel price and vehicle quality were found to be important drivers in the success of diesel vehicles from the consumer point of view [14].

With modern ultra-low sulfur diesel fuel implemented in 2007 for all on-road diesel applications, the price of diesel fuel has climbed higher than its gasoline counterpart due to greater refining costs. This loss in the per gallon price advantage of diesel has hurt the modern sales of diesel engines. Along with higher per gallon prices, the sticker price of diesel vehicles remain higher due to more expensive after treatment systems. In order to reduce the NO_x and smoke emissions, expensive urea injection systems, PM traps, and oxidation catalysts are being coupled to the exhaust systems. In order to make the diesel engine more competitive in the market for transportation based engines, the application of in-cylinder emissions reductions can help to reduce the cost of these after treatment systems and subsequently reduce the cost of diesel vehicles.

1.4 Purpose

A particular solution to the higher NO_x and PM can be found by implementation of low temperature diesel combustion which can realize up to 90% and 70% reduction in NO and PM, respectively, compared with conventional diesel combustion [15].

Low temperature combustion is not novel; even its implementation in engine systems dates back 30 years [16]. It is now widely demonstrated across a breadth of applications, including light-duty (e.g., passenger cars) [17] - [24] up to heavy-duty (e.g., large trucks) [25] - [30]. Significant understanding about the implication of low temperature combustion in a diesel engine is provided by Kamimoto and Bae [31], who draw a relationship among combustion temperature, combustion stoichiometry, nitric oxide formation, and carbon formation and oxidation. In their [31] study, the interest to move to high combustion temperature is motivated by the desire to decrease net soot emissions. Correspondingly, however, nitric oxide emissions increase. It is well established that nitric oxide formation and destruction is strongly coupled to the post-flame gas temperature [32]. Soot formation and oxidation are also strongly coupled to the post-flame gas temperature [33] - [35]. Consequently, net soot release – which is the difference between soot formation and soot oxidation, and eventually serves as the building block for particulate matter – conventionally possesses an inverse relationship with nitric oxide, commonly known as the “soot – NO” tradeoff. With low temperature combustion, both nitric oxide and soot formations are abated, resulting in a defeat of the soot – NO tradeoff and correspondingly low concentrations of both. Further, mixture stoichiometry no longer acts as a variable affecting in-cylinder soot formation [24].

Exactly how LTC is able to defeat the soot – NO tradeoff, and if conventional theory regarding swept parameters such as EGR and fuel pressure applies to the emissions production while operating in LTC is the primary focus of this article. This study will examine how low temperature combustion is able to reduce these in-cylinder emissions and will focus on the effects of the engine parameters by performing sweeps from minimum to maximum available EGR flow and fuel pressure throughout the engine test.

Much of the difficulty in maintaining superior fuel efficiency with low temperature combustion is ensuring combustion is properly phased [36]. Other issues, including decreased combustion efficiency (resulting from increased hydrocarbon and carbon monoxide concentrations) and increased pumping work, can also potentially affect the engine’s efficiency. This study also explores the behavior of such parameters (e.g., combustion phasing, combustion efficiency, and pumping work) in the attainment of low temperature combustion. The secondary objective of the study is to identify the major causes for changes to brake fuel conversion efficiency between conventional and low temperature combustion modes as EGR and fuel pressure are swept, and thus provide insight to combustion researchers for mitigating negative consequences of low temperature combustion on engine efficiency.

2. EXPERIMENTAL SETUP AND METHODOLOGY

2.1 Test Apparatus (Engine System)

The study follows an experimental approach using a medium-duty (4.5L) diesel engine. Details of the four-cylinder engine are included in Table 1. Most notable are the engine's use of an electronically-controlled fuel system, variable geometry turbocharger, and exhaust gas recirculation (EGR). These technologically-advanced features enable the attainment of low temperature combustion in the engine.

Table 1. Summary of engine parameters of the test apparatus used for development of low temperature combustion.

Parameter	Value
Bore (mm)	106
Stroke (mm)	127
Displacement (L)	4.5
Rated Power (kW @ rev / min)	115 @ 2400
Compression Ratio	16.57:1
Ignition	Compression
Fuel System	Electronic common rail, direct injection
Air System	Variable geometry turbocharger with EGR

The engine is loaded by a DC electric dynamometer which holds the engine speed constant and absorbs the brake power of the engine. Engine load (torque) is controlled via the fuel quantity. Full-authority control over engine parameters (i.e., fuel injection, fuel pressure, and EGR level) is made possible with the use of a third-party stand-alone engine controller unit (ECU) (Drivven, Inc., San Antonio, Texas).

2.2 Test Fuel

The test fuel of the study is commercially available Diesel #2, the properties of which are given in Table 2. A consistent batch of fuel was used throughout all engine testing of the study.

Table 2. Summary of properties of fuel (commercially available diesel #2) used in the study.

Property [ASTM Method]	**Value (Units)**
IBP [ASTM D86]	173.4 (°C)
FBP [ASTM D86]	340.5 (°C)
Lower Heating Value [ASTM D240N]	43.008 (MJ/kg)
Density [ASTM D4052s]	825.5 (g/L)
Viscosity [ASTM D445 40c]	2.247 (cSt)
Carbon Weight [ASTM D5291]	85.81 (%-weight)
Hydrogen Weight [ASTM D5291]	12.41 (%-weight)
Sulfur [ASTM D5453]	5.3 (ppm)
Cetane Number [ASTM D613]	51.3
Saturate Concentration [ASTM D1319]	74.2 (%-vol)
Olefin Concentration [ASTM D1319]	1.1 (%-vol)
Aromatic Concentration [ASTM D1319]	24.7 (%-vol)

All measured properties were conducted in a fuel testing laboratory on a sample taken from the consistent batch of Diesel #2.

2.3 Measurement Summary

Measurements and calculations are used to generate the data that support the analysis of this study. A summary of the measurements collected in this study are given in Table 3.

The data collected on engine control parameters, such as EGR valve position, common rail fuel pressure, and turbocharger speed come from sensing the stock sensors and equipment with the Drivven stand-alone ECU using custom calibration maps. The position for the VGT is set using external manual controls.

Table 3. Summary of measurements, along with their respective techniques, used in this study.

Variable	Description	Technique
CO	Carbon Monoxide	Non-dispersive infrared
CO_2	Carbon Dioxide	Non-dispersive infrared
EMP	Exhaust Manifold Pressure	Strain-gauge transducer
EMT	Exhaust Manifold Temperature	K-type thermocouple
HC	Exhaust HC concentration	Flame ionization detection on a C3 basis
IMP	Intake Manifold Pressure	Strain-gauge transducer
IMT	Intake Manifold Temperature	K-type thermocouple
P	In-cylinder pressure	Piezo electric transducer

Table 3. Continued.

Variable	Description	Technique
	Fuel Mass Flow Rate	Calculated from fuel density and volumetric flow rate measured with positive displacement flow meter
N	Engine Speed	Dyno shaft encoder
NO	Exhaust Nitric Oxide Concentration	Chemiluminescence
O_2	Oxygen	Paramagnetic
Smoke	Smoke Concentration	Reflectivity (smoke meter)
T_b	Brake Torque	Dyno-mounted load cell

NO is reported in this study and used for comparison of concentrations between the two diesel fuels. It should be noted, however, that several of the above-cited studies report NO_x.

2.3.1 Fuel to Air Ratio

– is calculated from measurements of exhaust concentrations of carbon dioxide (CO_2), carbon monoxide (CO), and oxygen (O_2). Carbon and oxygen based computations of – are provided by [16]; these correlations, however, are restricted to pure hydrocarbon fuels. A more general expression for equivalence ratio, , is provided in [37] which allows for any general fuel potentially containing oxygen and nitrogen components. The only assumptions applied in these correlations are the assumption of equilibrium between CO, CO_2, water (H_2O), and hydrogen (H_2) and the assumption of the equilibrium constant. Based on recommendation of [16], a value of 3.8 is assumed for this equilibrium constant. This study employs the computation provided in [37] since the biodiesel under study contains oxygen. Thus, F/A is determined as given by:

where – is the stoichiometric fuel-air ratio of the fuel. Measurement techniques of exhaust CO, CO_2, and O_2 are summarized above in Table 3.

2.3.2 Emissions Sampling

The analyzers used to measure gaseous species are housed in an emissions bench which, in addition to supplying the analyzers with gaseous sample, conditions the sample for temperature and humidity. The raw gaseous samples (i.e., CO, CO_2, O_2, and NO) are filtered (at 190°C) and delivered to the emissions bench in a heated line (190°C). Upon entry to the emissions bench, a portion of sample is chilled and dehumidified for analysis by the non dispersive infrared analyzers. The balance of sample is delivered in heated lines to both the flame ionization detector and chemiluminescence analyzers, which are each heated to 190°C. Further, a separate heated (190°C) sample line delivers sample to the exhaust smoke meter, which is also heated (190°C). In this thesis, exhaust NO and HC concentrations are reported on a wet-basis; CO concentrations are reported on a dry basis.

2.3.3 AVL Smokemeter

The AVL 415S smokemeter used in this study is an optical filter device which passes a known volume of exhaust, sampled after the turbocharger through a heated sampling line, and passes it through a filter paper with a known cross-sectional area. An optical reflectance detector calculates the presence of soot concentrations by reporting a difference between the darkened sample and new filter paper [38].

The filtered smoke number (FSN) is reported as accurate to a resolution of 0.01 FSN. Five samples are taken and averaged per engine condition for each exhaust smoke concentration.

2.3.4 EGR

In order to calculate the mass fraction of EGR in the intake system, the exhaust species present in the intake manifold must be sampled. A sample line from the intake manifold to the Horiba emissions bench allows for the analysis of the well mixed fresh intake air with the EGR.

To calculate the EGR level, the mass fractions of exhaust species in the intake manifold are summed:

Only major exhaust species are used in this computation (CO_2, O_2, H_2O, and N_2). The mass fractions of these species are determined by calculating a dilution ratio of CO_2 in the intake versus the exhaust system:

This dilution ratio is a volumetric ratio, necessitating the conversion from mole fraction to mass fraction. Mole fractions for H_2O are calculated as in [37] using the water-gas shift reaction for equilibrium between species:

and these mole fractions for H_2O is found by:

where K is an experimentally found constant given in Heywood to be 3.0. M/2N adjusts for the fuel composition used. Other exhaust mole fractions are found using the measured exhaust concentrations multiplied by the difference of the mole fraction of exhaust H_2O from 1. The other intake species are found using the dilution ratio multiplied by the respective exhaust mole fractions.

In order to convert from a mole basis to a mass basis, the mole fraction of each species is multiplied by its molecular weight and divided by the molecular weight of the EGR:

The sum of these mass fractions allows for calculation of the EGR mass percentage of the engine.

2.3.5 Time Averaged Measurements

Data for other engine operating conditions including temperatures and pressures are collected using in house data acquisition. This "low-speed" data acquisition system averages 100 sample points of data collected over a period of time after the engine has reached steady state. The 100 sample points are averaged and the averaged value is reported as the measured value for that operating condition.

2.3.6 Crank Angle Resolved Measurements

In-cylinder pressure is measured from cylinder #1 (the forward most cylinder) on a crank-angle resolved basis (0.2 degree resolution) using a piezo-electric pressure transducer. The ordinary calibration and fidelity checks [39] routinely occur for this measurement. In-cylinder pressure is used in the calculation of mean effective pressures and of rate of heat release, the latter of which uses standard methods and well-developed correlations [40] - [44]. Rate of heat transfer is calculated using Hohenberg's correlation [41] and assumes a wall temperature of 550 K. Also measured on a crank-angle resolved basis are the injector command and needle lift motion. The former is used to indicate "Injection Timing" in all the related plots in this article. The commanded start of injection precedes the actual start of injection by about 1.4°. All crank-angle resolved measurements are collected for 300 consecutive cycles; analysis is performed on the averages of the 300 cycles.

2.3.7 Combustion Efficiency

Combustion efficiency is reported in this thesis and calculated using standard techniques described by [37]. Measured CO, calculated H_2, and measured HC species are used in this calculation (i.e., PM is neglected). Calculation of H_2 concentrations uses the water-gas shift equilibrium assumption among CO, H_2O, CO_2, and H_2 [37] and an equilibrium constant equal to 3.8 [45]. The heating value of the HC species is taken to be the same as the fuel as recommended by [37]; the molecular weight of the HC species (needed for conversion from measured mole fractions to mass fractions) is taken to be that of propane (measurement basis of the HC analyzer).

2.4 Determination of Uncertainty

Uncertainty of measurements is rigorously evaluated using standard techniques developed for engineering practices [46]. Calibration of all instruments is routinely conducted to minimize systematic uncertainty. Random uncertainty in engine testing can be high, due to a number of extraneous ambient factors. Repeated testing of operating conditions and combustion regimes (i.e., multiple sets of measurements) is done over several days in order to capture an understanding of random uncertainty (i.e., fluctuations in daily ambient conditions and the capability to repeat the same engine operating condition), and creates a sample of data, of which statistical analysis is performed using routine techniques [47], [48]. Using methods prescribed by [48], reported uncertainty combines the instrument's precision and accuracy with the test data's standard deviation. Two standard deviations are used to give roughly 95% confidence in the reported range. Uncertainty bars in the data figures result from this analysis. Also, in the data figures, lines connecting data points are meant to illustrate the series of data, not necessarily suggest a trend between data points.

2.5 Test Methodology

2.5.1 Determining Baseline Conventional and LTC Regimes

In order to satisfy the objectives of this thesis, both a conventional combustion and a LTC strategy must be developed through the stand alone ECU. Once these combustion strategies are defined, sweeps of EGR and fuel pressure can be performed to allow considerations on emissions and efficiency.

To develop a low temperature combustion strategy on the new stand-alone ECU, a conventional combustion replication of the stock controller must be made. It was determined that a baseline condition would be established at 1400 rev/min, 50 ft-lbs torque (1.9 bar brake mean effective pressure) due to the low speed and low load that would allow for attainment of LTC. The stock calibration for this condition uses pilot injection; it is desired for this study, however, to develop low temperature combustion with single injection. Thus, the second step is to replicate the same speed and torque of the engine using a single injection. Injection timing of the single injection mode is adjusted to yield about the same torque as the multiple injection mode using the same EGR valve position, VGT position, fuel pressure, and injection duration (to keep fuel flow constant).

Thus, the conventional condition (i.e, the single injection replication mode) does not necessarily represent a "best efficiency" or "best emissions" mode; it represents the "calibrated" mode of the production engine. In other words, the injection timing is not optimized for proper phasing of combustion, rather it was chosen to try and replicate the stock calibration with a single injection.

A summary of stock calibration (the dashed black multiple injection curve) and the new conventional condition (the solid blue single injection replication curve) is provided in Figure 1, which shows in-cylinder pressure as a function of crankangle (in degrees after top dead center, or deg ATDC). Notice that reasonable replication is attainable with the single injection mode. For both cases the EGR level is 0%, fuel pressure is 816 bar, and the VGT is manually set to provide the same boost and turbocharger speed.

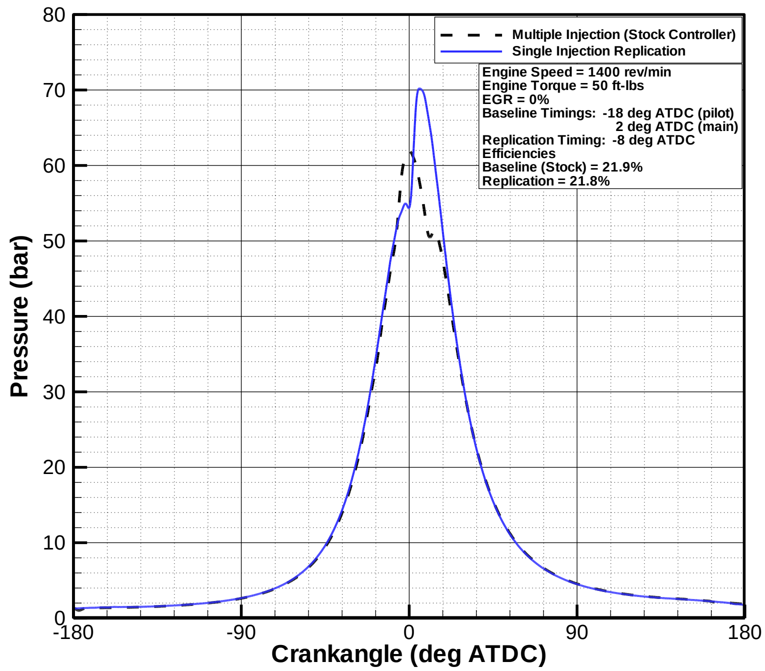

Figure 1. Pressure as a function of crankangle for 1400 rev/min and 50 ft-lbs torque using (black dashed curve) pilot injection with stock controller and (blue solid curve) single injection mode with similar torque and same fuel flow (yielding similar efficiencies).

Once the single injection replication mode is developed, low temperature combustion attainment is realized by increasing the EGR level to 56% (maximum attained level with the stand-alone ECU at the stock VGT setting and the EGR valve fully open), then retarding injection timing from -8° ATDC to 0° ATDC. Previous work [49] details confirmation of attainment of LTC at the 0° ATDC injection timing. This timing was chosen for examination in this LTC study because of significant reductions in both NO_x and smoke while still operating in a late injection strategy LTC regime.

Now that both baseline combustion regimes have been developed on the stand-alone ECU, sweeps of EGR and fuel pressure must be done in order to accomplish the stated objectives.

2.5.2 EGR Sweep

A sweep from 0% EGR valve position to 90% EGR valve position is performed for both LTC and conventional combustion to determine the effects of EGR on both combustion cases. While LTC is only realized with late injection timing and full EGR flow, the late injection timing case will continue to be called the “LTC regime” throughout the EGR sweep, even if substantial reduction in emissions are not realized.

In order to allow proper resolution for the nonlinear EGR valve position and EGR % relationship shown in Figure 2, a sweep was conducted on EGR valve position to identify locations which needed more or less resolution in valve positions. The final valve position sweep conducted in this thesis is shown in Table 4. Notice that 90% valve position is the maximum value that the EGR valve can be opened on this engine setup.

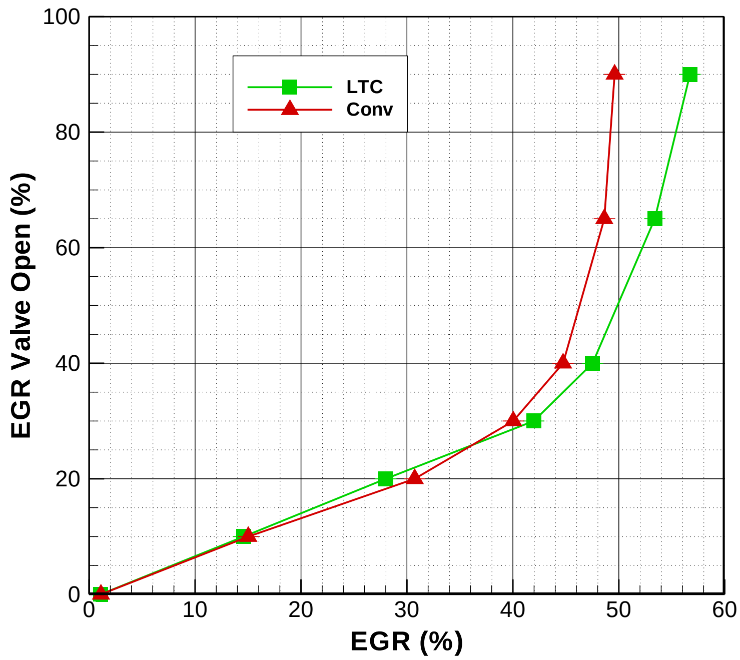

Figure 2. EGR valve position versus EGR % for LTC (■) and conventional (▲) combustion.

Table 4. Summary of EGR valve position sweep for LTC.

EGR Valve Pos	EGR	Torque
%	%	ft-lbs
0	1.1	47
10	14.6	42.5
20	28.0	44.5
30	42.0	41.5
40	47.5	38
65	53.4	30
90	56.7	29

To conduct the EGR sweep in the laboratory, the engine is first warmed up at the baseline LTC condition. After the engine fully warms up, the EGR valve is fully closed and the engine is allowed to reach steady state. Data is taken at each valve position making sure to reach steady state between different tests. Once the LTC condition has been fully swept, the injection timing is retarded to the baseline conventional case, and the same sweep is performed for conventional combustion. This sweep is rerun on a different day and used to conduct statistical uncertainties.

2.5.3 Fuel Pressure Sweep

For the fuel pressure sweep, pressures were adjusted from 816 bar used in the baseline conditions, and varied from 500 bar to 1500 bar in 250 bar increments. Table 5 summarizes the fuel pressure sweep with the resulting change in main duration and the relatively constant fuel flow rates.

Table 5. Summary of fuel pressure sweep for LTC and conventional combustion.

	Fuel Press	Main SOI	Main Dur	Fuel Flow Rate	Torque
	bar	dBTDC	CAD	(g/s)	ft-lbs
LTC	500	0	7.05	1.122	28.7
	750	0	5.775	1.132	39.2
	1000	0	5.125	1.111	40.8
	1250	0	4.65	1.125	41.4
	1500	0	4.255	1.106	37.3
Conv Combustion	500	8	7	1.118	57.7
	750	8	5.8	1.121	56.9
	1000	8	5.11	1.120	53.8
	1250	8	4.61	1.113	49.2
	1500	8	4.28	1.100	44.7

Torque was allowed to change throughout the sweep, as the goal was to maintain constant fueling between cases. The EGR valve was not adjusted from the baseline conditions, meaning that the LTC regime runs maximum EGR throughout the fuel pressure sweep, while the conventional combustion regime has zero EGR.

Similar to the EGR sweep, the laboratory test for the fuel pressure sweep is conducted with the LTC baseline condition for engine warm up. Once steady state is established, fuel pressure is reduced to 500 bar and the injection duration is set accordingly. Incremental increases in fuel pressure and reductions in injection duration are performed until the parameter sweep is complete. The injection timing is retarded for the conventional combustion case, where fuel pressure is swept and data is recorded. Re-runs of fuel pressure sweeps are conducted in order to perform statistical analysis on the data.

3. RESULTS AND DISCUSSION

The following section provides the results and discussion necessary to complete the objectives of this study. In order to more effectively do so, this section has been broken up into two sub-sections: 3.1) observation and analysis of exhaust NO, CO, and smoke concentrations as influenced by sweeps in EGR and rail pressure, and 3.2) influence of EGR and rail pressure sweeps on engine efficiency.

3.1 Emissions Considerations

In order to identify the effects that cause changes to the engine out emissions of a diesel engine, sweeps of EGR and rail pressure are conducted under both LTC and conventional combustion to allow researchers to directly compare the effects of these engine functions (namely EGR and rail pressure) on the emissions.

3.1.1 EGR Sweep

3.1.1.1 Characteristics of EGR on Emissions

EGR allows for substantial reduction in NO_x emissions in diesel engines. Recycling the exhaust gasses ultimately does this by reducing combustion temperature [50]. This is due to the fact that EGR is the re-introduction of products of combustion back into the cylinder. The exhaust gas recirculation takes up a part of the cylinder that would normally be filled with a combustible air/fuel mixture and acts as a non-reacting species. These non-reacting species absorb energy during the reaction and act to decrease the adiabatic flame temperature. Higher percentages of EGR introduce more exhaust gasses into the cylinder, thus more non-reacting species and lower adiabatic flame temperatures. EGR decreases in cylinder temperatures through oxygen dilution, increased thermal mass, and decreased dissociation mechanisms to ultimately lower NO formation [51].

As seen in Figure 3, and consistent with literature, nitric oxide concentrations in general decrease as EGR is increased. Conventional combustion is able to realize an 87% reduction in NO concentrations with 50% EGR. The LTC regime is able to realize a significant reduction in NO, up to a 98.6% reduction, or 3ppm final exhaust concentration compared to its conventional counterpart of 45ppm NO concentrations. Several factors influence this, including the ability for LTC to have higher mass percentages of EGR inside the cylinder, up to 56%, and these factors will be discussed in the following section.

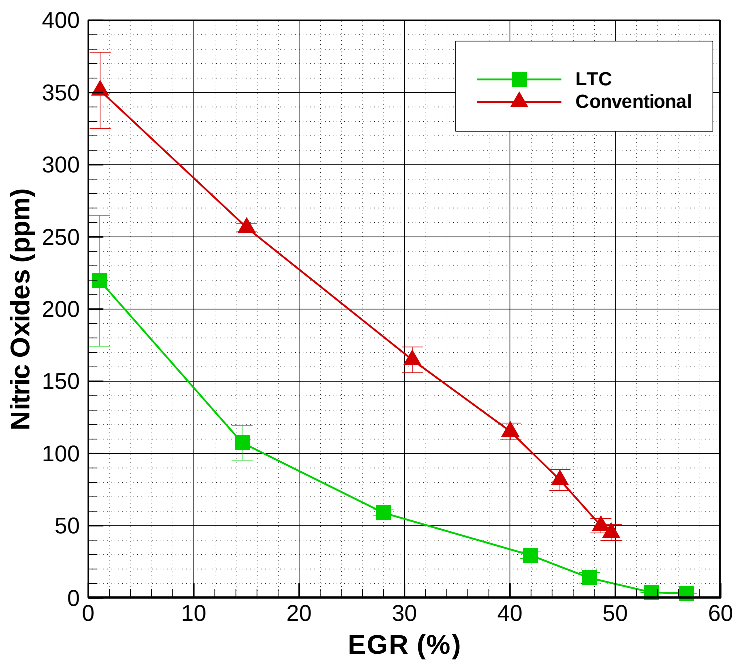

Figure 3. NO concentrations versus EGR for LTC (■) and conventional (▲) combustion.

Smoke, an indicator of exhaust soot and an ingredient of particulate matter, is shown in Figure 4. As EGR is added to the system, charge dilution acts to reduce in-cylinder temperatures. Since soot emissions are a culmination of the difference between soot formation and soot oxidation, the balance of these two forces yield the total engine out concentrations of soot. Literature [52] states that traditional thought shows soot oxidation is affected more than soot formation with charge dilution due to lower in-cylinder temperatures primarily affecting the oxidation mechanism. With very high levels of EGR (greater than 56%), [53] has shown that the flame temperatures are significantly reduced as to affect soot formation rates and prevent combustion from operating around areas where soot formation can occur.

It is assumed that this trend is visible in Figure 4 as increased levels of EGR reduce soot oxidation in the conventional combustion case and increase the amount of soot exhaust concentrations. During LTC, the late phasing of the combustion allows for substantially lower combustion temperatures, which limit soot formation processes and prevent high levels of soot being formed. If it were possible to realize more than 56% EGR, it is assumed that temperatures would be abated to lessen soot formation, and ultimately lower soot concentrations even more for both combustion cases as shown in [54].

As EGR is added, soot increases for LTC by 370% to .064 FSN (this percent increase is high mainly because of the negligible soot at the late injection timing with 0% EGR), while conventional combustion has a 1300% increase in smoke to 1.06 FSN.

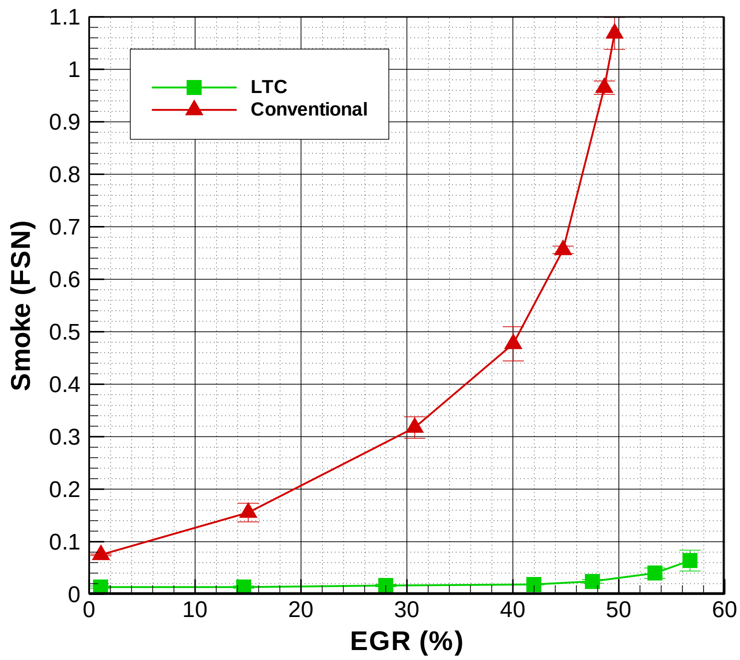

Figure 4. Smoke concentrations versus EGR for LTC (■) and conventional (▲) combustion.

The typical soot – NO tradeoff is present in the conventional combustion case in Figure 5. As EGR charge dilution is increased, lower flame temperatures reduce NO production and decrease soot oxidation, ultimately allowing more soot exhaust concentrations. The soot – NO tradeoff initially appears for LTC in Figure 6 with inverse trends visible for soot and NO with increasing EGR. However, when scaled with the conventional case as in Figure 7, it is seen that the typical tradeoff is mostly defeated with the application of LTC.

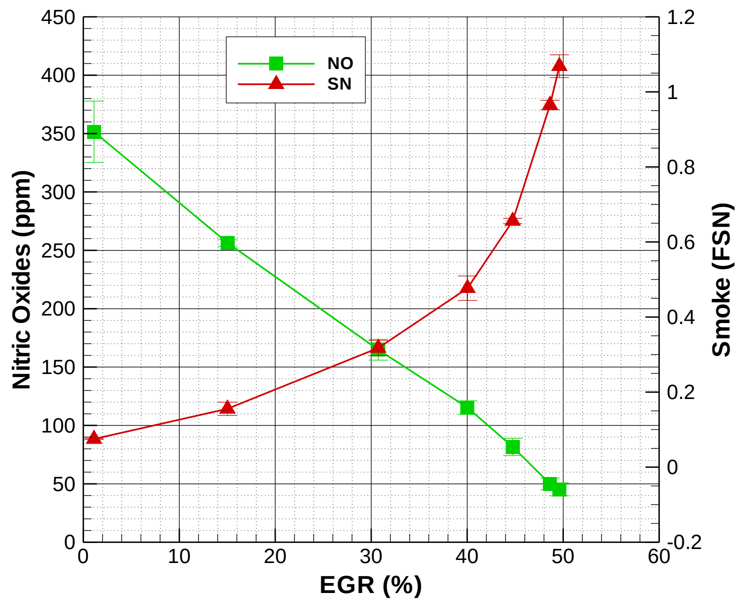

Figure 5. Typical soot - NO tradeoff versus EGR for conventional combustion.

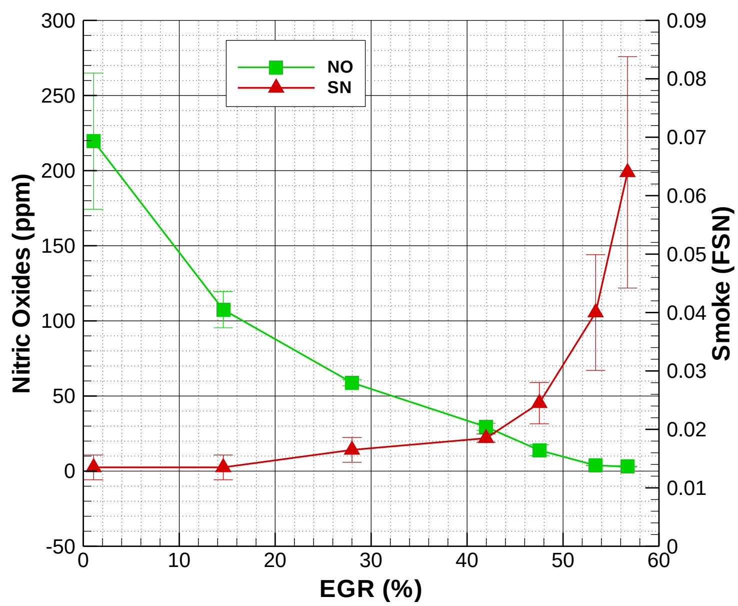

Figure 6. The soot - NO tradeoff versus EGR is reduced for LTC case.

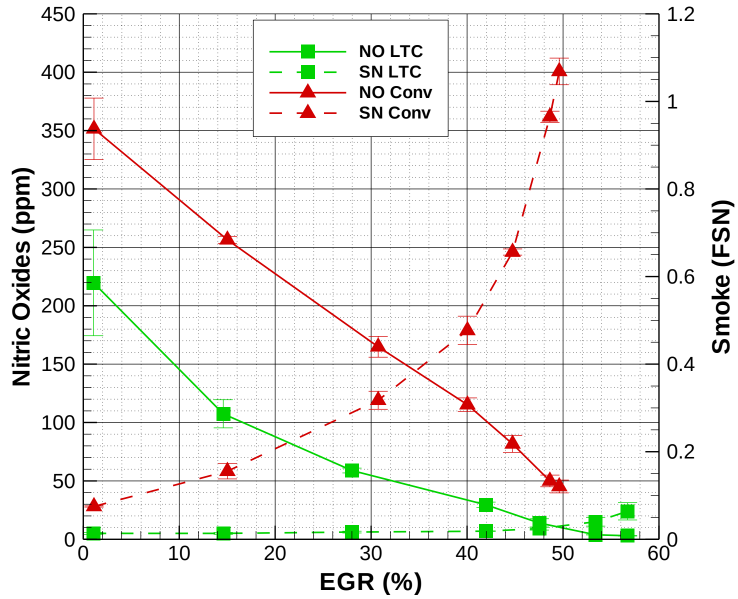

Figure 7. Defeating the typical soot - NO tradeoff versus EGR for LTC (■) and conventional (▲) combustion.

CO production typically occurs in fuel-rich regions of premixed burning [55]. During high temperature conventional diesel combustion, CO concentrations are typically small due to high oxidation of CO into CO_2. High levels of OH peak with the maximum in-cylinder temperatures, and allow oxidation of CO using the following:

where CO oxidation is very active as the charge goes through the high temperature region of combustion [52].

LTC, especially at high EGR dilution, exhibits high CO concentrations visible in Figure 8. Due to the extended ignition delay in LTC, large amounts of mixing occur to produce regions of very low equivalence ratios. These regions burn at very low temperatures, and they are too low to enable oxidation of CO with OH. It is interesting to note that it has been shown that combustion phasing has little effect on CO production, rather the dominant force is combustion temperature [56]. Highly diluted charge has been shown to be the dominant source of CO during conventional diesel combustion [55].

Kook et al. [52] goes on to state that the presence of soot in LTC implies that part of the premixed burning occurs under fuel-rich conditions. Large amounts of CO are formed during rich combustion, and within a LTC regime, the lower in-cylinder temperatures of the diluted charge imply that there is less time available before cooling from expansion quenches the combustion.

For these reasons, charge dilution leading to very lean combustion in the conventional case acts to increase CO concentrations in Figure 8. For LTC, reduced combustion temperatures lower the oxidation of CO, and a quenched reaction from expansion cooling prevents the possibility for oxidation to occur.

With the application of EGR, LTC CO exhaust concentrations increase by 400% to 4160 ppm, while conventional combustion increases a similar amount of 470% to 640 ppm.

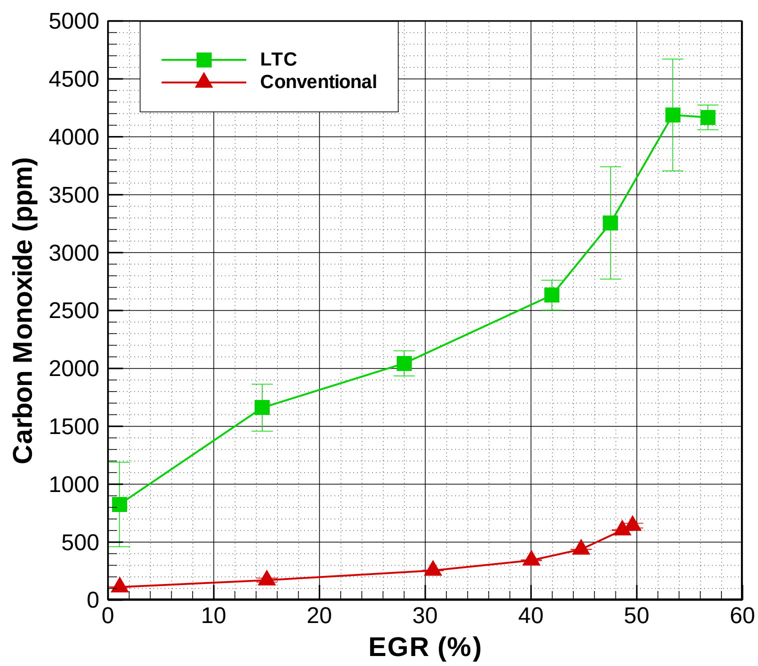

Figure 8. CO concentrations versus EGR for LTC (■) and conventional (▲) combustion.

3.1.1.2 Analysis of EGR on Emissions

In order to understand the mechanisms creating NO_x and soot formation, it is important to realize what is happening locally in the combustion region. The contour plot in Figure 9 gives insight into localized combustion and shows regions of soot and NO_x formation as equivalence ratio and adiabatic flame temperature vary [52], [57]. Experiments were originally conducted using a constant volume combustion bomb with a premixed fuel rich mixture of propane, oxygen, and inert gas. This experimental work allowed the soot formation region to be plotted on a $\varphi - T$ diagram. The NO formation region was determined with the Zeldovich equations and was plotted to give an idea of exhaust concentrations for these two constituents [31].

Soot formation occurs at high equivalence ratios and lower adiabatic flame temperatures (typically occurs during a conventional diffusive burning regime), whereas

NO_x formation occurs at lower local equivalence ratios and higher temperatures (typically occurs under more well mixed zones during premixed combustion).

As EGR is added to the cylinder, excess diluent decreases available oxygen, increases non-reacting thermal mass, and decreases dissociation mechanisms, lowering in-cylinder combustion temperatures. Figure 10 shows the maximum in-cylinder bulk-gas temperatures as EGR is swept. Increasing EGR reduces maximum combustion temperatures and subsequently, NO concentrations. NO is a minimum at the lowest in-cylinder temperatures.

Another factor which may affect NO concentrations is the EGR rate. Figure 3 illustrates that LTC is able to further reduce NO concentrations with 7% additional EGR in-cylinder mass. LTC combustion is able to flow more EGR for several reasons.

Manifold pressures for both the intake and exhaust are displayed in Figure 11. Intake manifold pressure, or IMP, remains fairly constant for both conditions, however, exhaust manifold pressure, or EMP is generally higher for conventional combustion. Since differential manifold pressures are indicative of the potential for EGR flow, Figure 12 suggests that conventional combustion has a higher potential to flow EGR. Because conventional combustion actually flows less EGR, there is more information to the air system that must be analyzed.

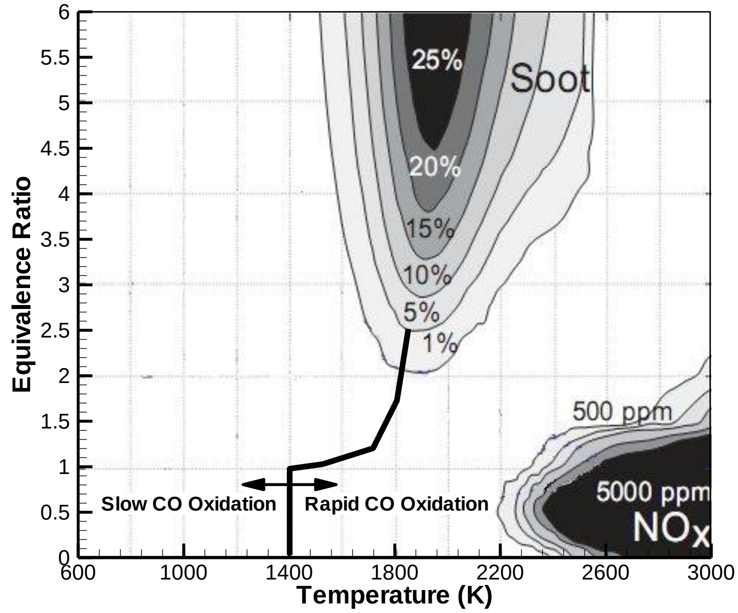

Figure 9. Equivalence ratio versus temperature showing typical regions for soot and NOx formation [52].

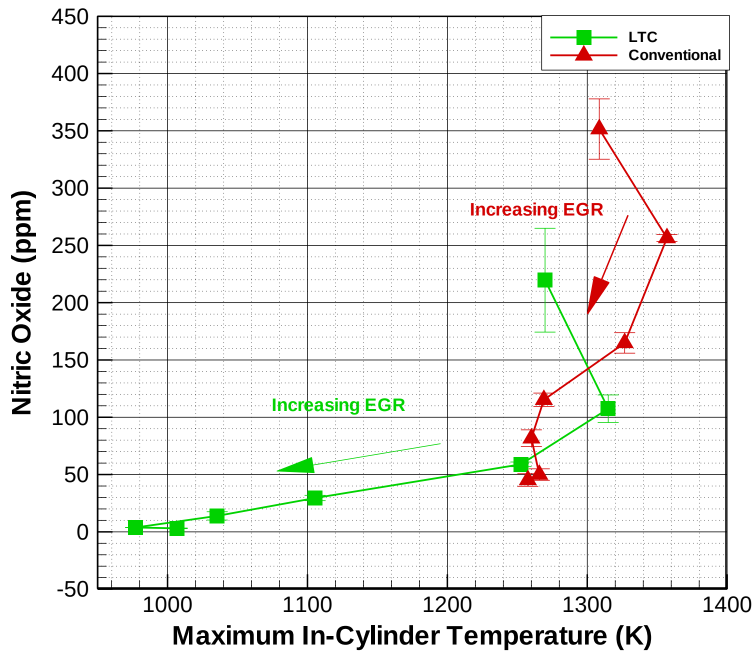

Figure 10. NO concentrations as a function of maximum in-cylinder bulk-gas temperatures as EGR is added.

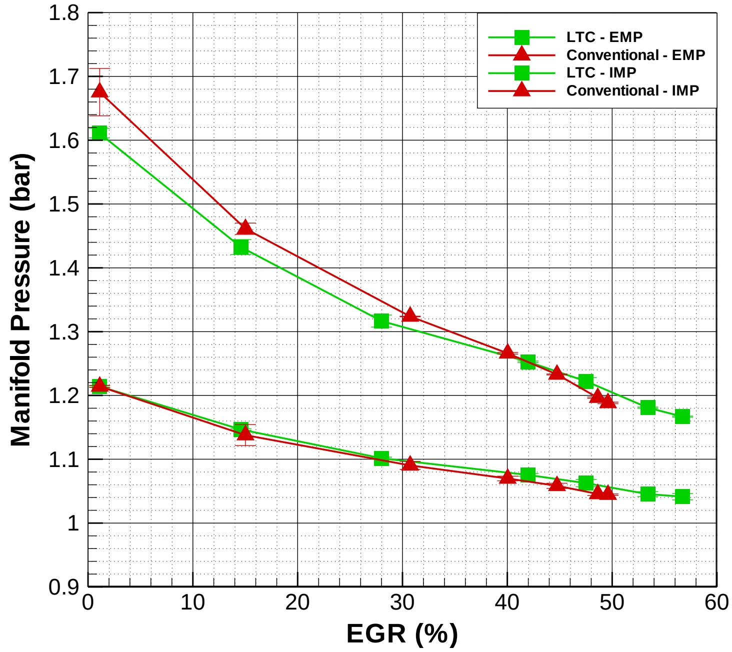

Figure 11. Manifold pressures versus EGR for LTC (■) and conventional (▲) combustion.

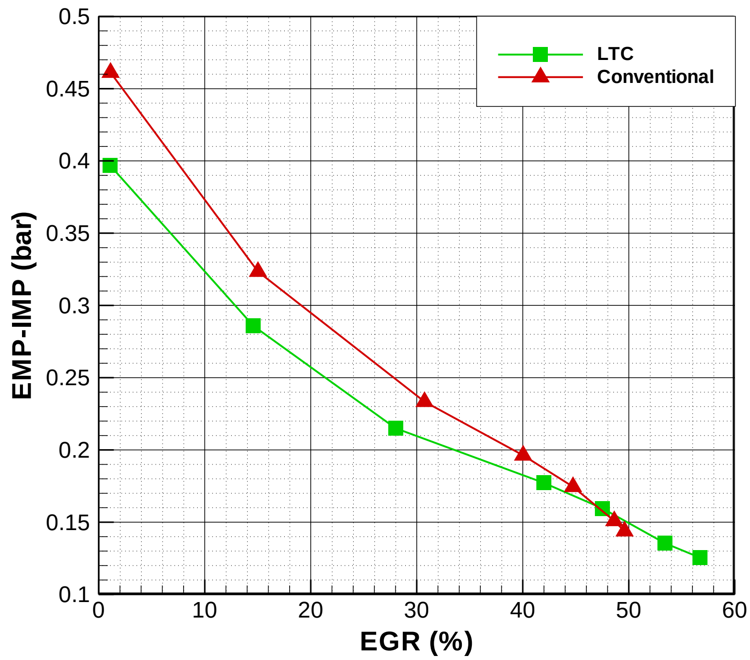

Figure 12. Differential manifold pressures versus EGR for LTC (■) and conventional (▲) combustion.

In order to understand why the conventional combustion case does not utilize the manifold pressure differential potential to flow more EGR than LTC, an in depth look must be given to the air system of the engine.

Starting with the exhaust system and working to the intake system, analysis will be performed to understand the trends of airflow within the engine. The EMP for conventional combustion is higher than LTC, but Figure 13 shows that the exhaust manifold temperature, or EMT, is greater for LTC. High EMT is indicative of exhaust energy, and extra exhaust energy is usually found when the combustion energy has not been fully extracted through the expansion stroke of the engine. It is assumed that LTC has higher exhaust temperatures because of the later phasing of the injection and subsequently the in-cylinder combustion.

To confirm this, the high-speed crank angle resolved in-cylinder pressure and calculated in-cylinder temperature is displayed in Figure 14 and Figure 15, respectively. The exhaust valve open condition, EVO, occurs at 115 dATDC and it is at this location when the calculation for in-cylinder temperature stops as the system becomes open, and explains why calculated in-cylinder temperature drops off at this location. These calculated in-cylinder temperatures are used for instructive purposes. They are not indicative of absolute temperatures, and can only be used to compare relative temperature differences between the two combustion cases.

On average, LTC exhibits slightly higher in-cylinder pressures and higher in-cylinder temperatures at the end of the expansion stroke when the exhaust valve opens. This explains the higher EMT for LTC, but before analyzing the rest of the engine air system, let's first examine the fundamental reason LTC has higher in-cylinder pressures and temperatures at EVO.

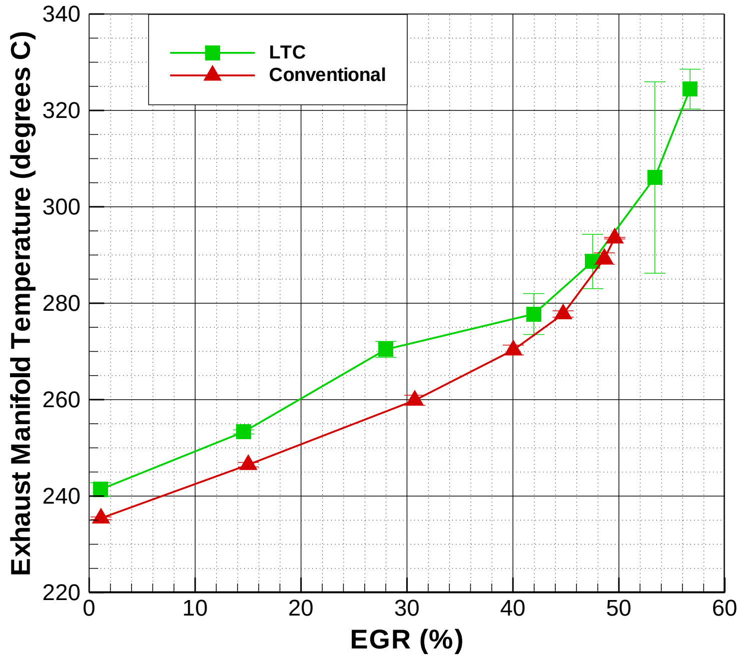

Figure 13. Exhaust manifold temperature versus EGR for LTC (■) and conventional (▲) combustion.

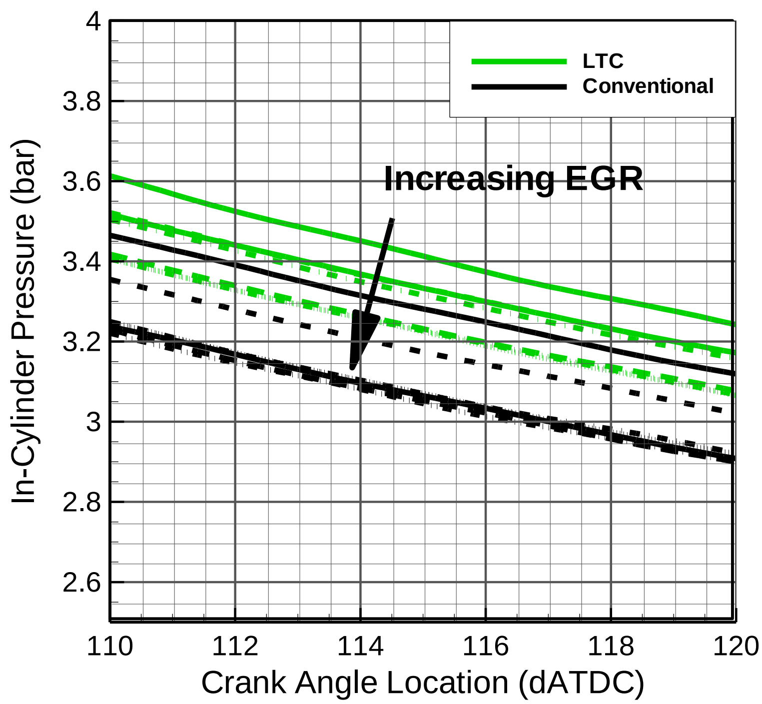

Figure 14. In-cylinder pressure as a function of cranke angle during exhaust valve opening.

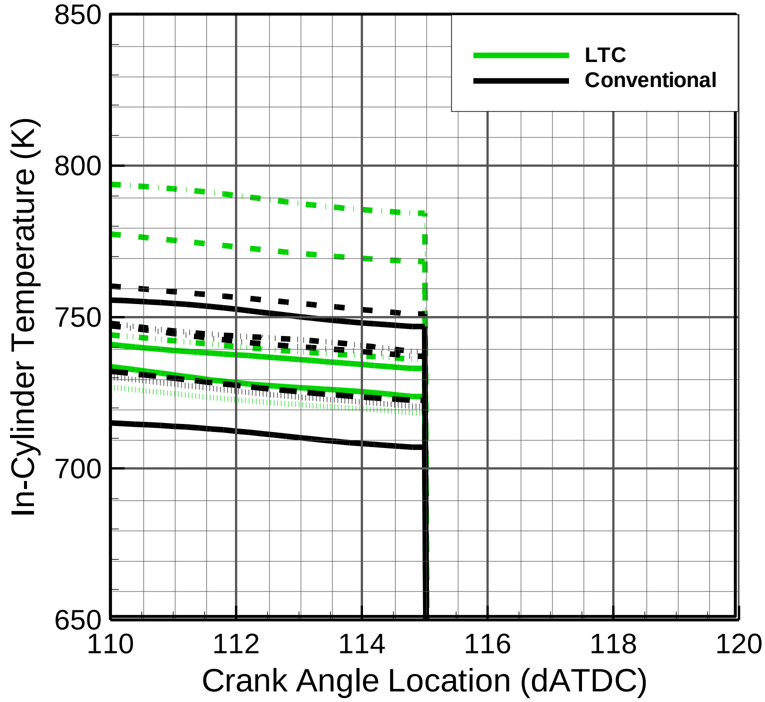

Figure 15. In-cylinder temperature as a function of crank angle during exhaust valve opening.

Literature is rich with discussion on LTC. In general, EGR slows the rate of combustion and increases the combustion duration. Figure 16 shows the rate of heat release, or ROHR, calculated in-cylinder temperatures, and in-cylinder pressures as a function of crank angle location for LTC. As EGR is added to the charge, ROHR is significantly reduced, and heat release occurs over a much longer period of the stroke. For the cases with large amounts of EGR, combustion is still commencing at up to 40 dATDC, shown by positive rates of heat release. This late combustion phasing necessarily means that the energy from the combustion is not sufficiently expanded over the piston expansion stroke, allowing for higher exhaust temperatures.

During the conventional combustion case, EGR reduces ROHR, but not to the same effect as in the LTC case, evident in Figure 17. The reason for this is the combustion phasing. As ignition delay is increased and combustion is retarded in the conventional case with the increased levels of EGR, combustion still occurs near TDC. At this location there are still significant in-cylinder temperatures and pressures to allow for rapid mixing of the fuel injection and allow for a large, quick release of thermal energy. Through the sweep of EGR, conventional combustion temperatures drop 100K and in-cylinder pressures during conventional combustion drop 15 bar. Compare this to LTC with a drop of 300K in in-cylinder temperatures and an in-cylinder pressure drop of 25 bar.

Another interesting characteristic to note is the in-cylinder pressure trace of each regime before combustion occurs. Here, the pressure trace of the engine can be seen as the piston compresses the cylinder charge up to TDC. As EGR is added to the system, the in-cylinder pressures generally decreases. Several factors influence this.

As EGR is added to the system, the in-cylinder pressures will be lower for several reasons including: 1) the increased heat capacity of the diluent gases in the charge, 2) the reduced boost from the turbocharger, and 3) the thermal throttling due to the increased IMT.

The diluents added by increased EGR flow act to decrease the in-cylinder temperatures, which do several things: 1) it decreases the in-cylinder pressures, and 2) it decreases the heat transfer to the cylinder walls. Less heat transfer will take place to the fresh mixture charge on the next intake stroke, also lowering the in-cylinder temperatures and pressures of the mixture.

As EGR is added back into the intake manifold, less exhaust energy will pass through the turbocharger, resulting in lower turbine speeds, Figure 18, and lower boost pressures seen in the intake manifold, Figure 11. Lowering the pressure, and ultimately the density of the intake manifold will reduce the mass of the intake charge, and subsequently reduce the in-cylinder pressures throughout the compression stroke, visible in the in-cylinder pressure trace.

The last, and perhaps most dominating effect of these on in-cylinder pressures, is the effect of thermal throttling due to the increased IMT with additional EGR flow, Figure 19. Ladommatos et al. [51] describes that an increase in inlet charge temperature reduces the inlet charge mass because of thermal throttling. The lower inlet density will act to reduce the in-cylinder mixture density, and this can especially be seen on the in-cylinder pressure trace before combustion occurs. The thermal throttling also affects the EGR mass flow, as will be analyzed in the next few pages.

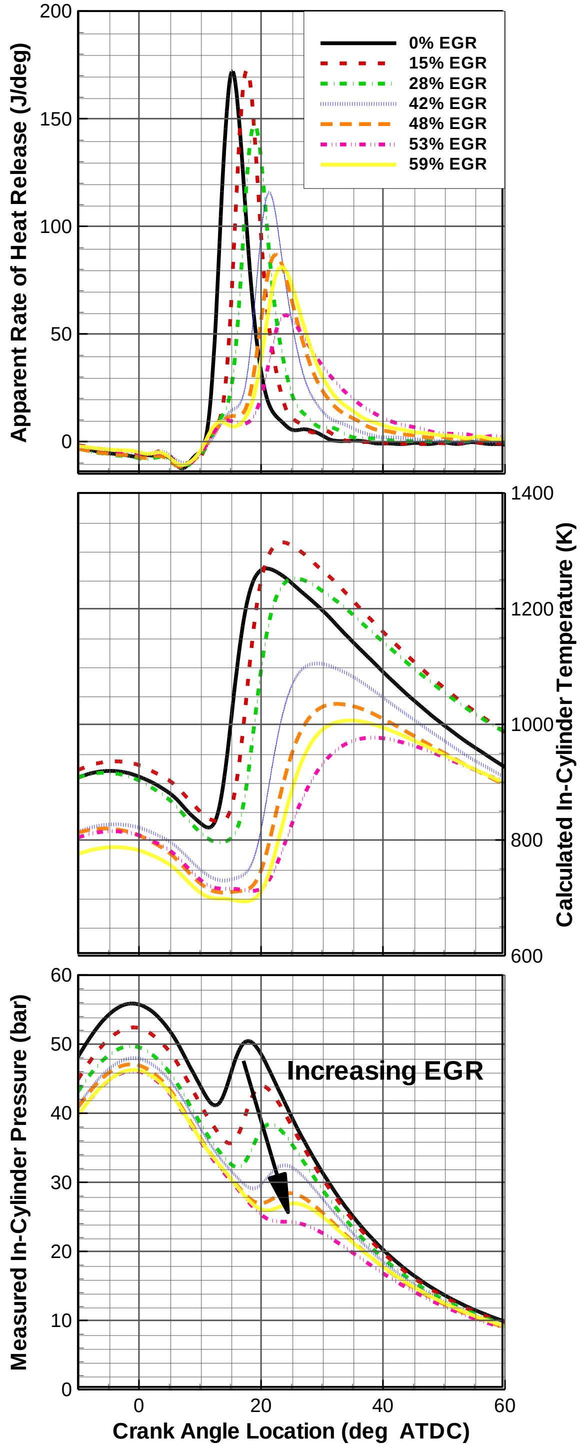

Figure 16. Rate of heat release, in cylinder temperature, and in-cylinder pressure versus crank angle for LTC.

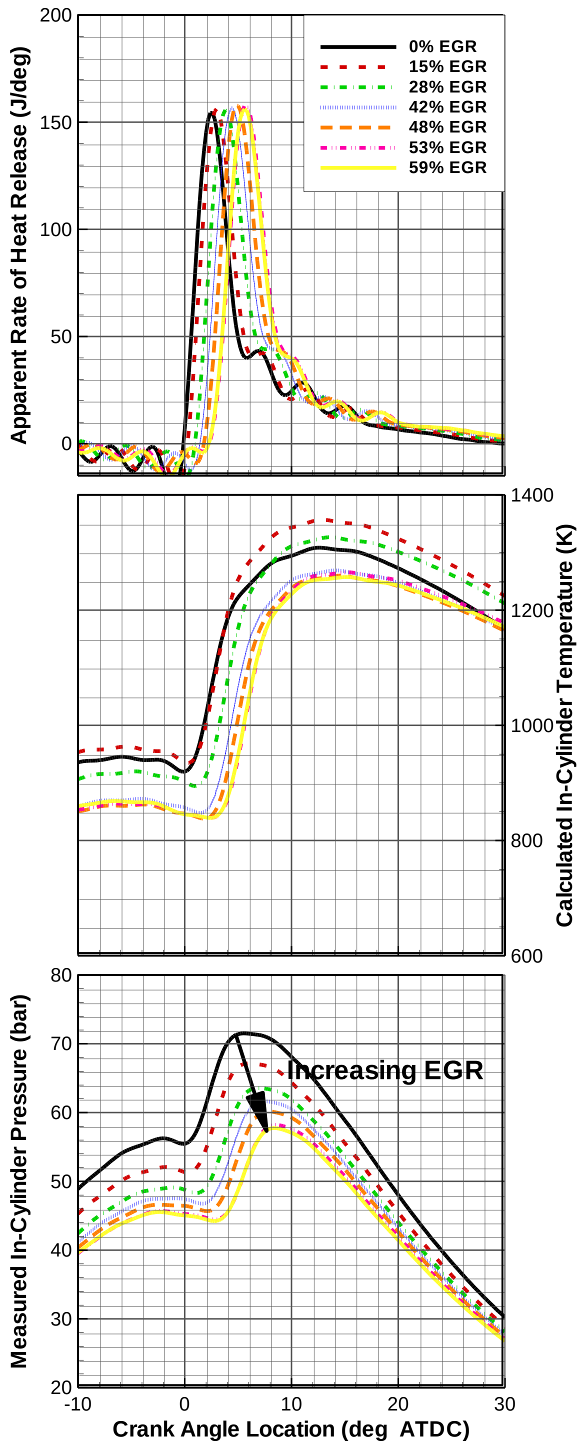

Figure 17. Rate of heat release, in cylinder temperature, and in-cylinder pressure versus crank angle for conventional.

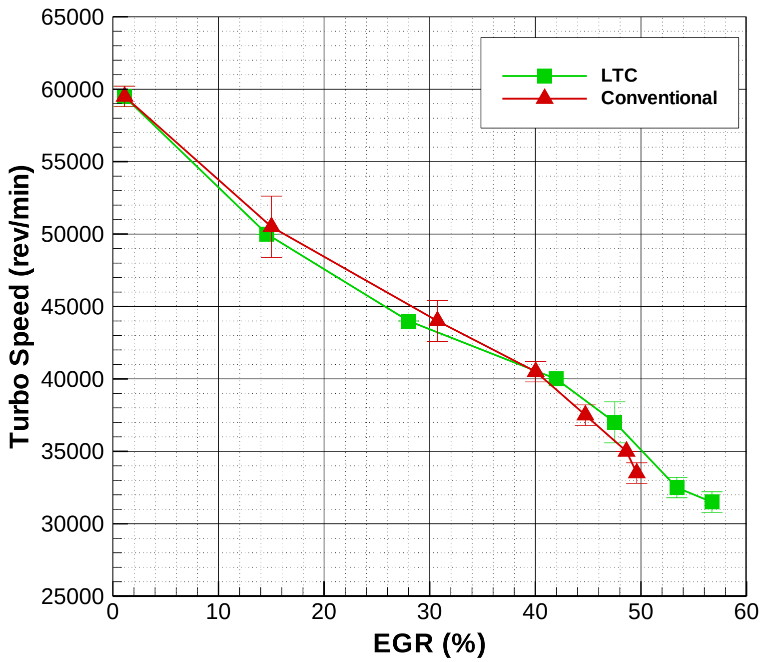

Figure 18. Turbo speed versus EGR for LTC (■) and conventional (▲) combustion.

Resuming the analysis of the air system, we next move to the intake manifold. Figure 19 reveals unexpected results with respect to the earlier discussion on exhaust manifold temperatures. So far, this investigation into the higher EGR levels for LTC has revealed that the LTC case has higher exhaust temperatures. Contrary to what would be predicted with more EGR at higher temperatures, examination of the intake manifold temperatures, or IMT, reveals that the LTC IMT is lower than conventional combustion. Perhaps an explanation could be that the EGR cooler equipped on this engine is able to remove the extra thermal energy in the LTC exhaust before it is mixed in the intake and prevent the extra heating of the intake charge, but this is unlikely. The only other explanation that would account for the addition of heat to the intake manifold could come from the fresh intake air itself.

The fresh air entering the engine is passed through a laminar flow element, and it is here that the intake air temperature is measured. Figure 20 confirms that the additional IMT for conventional combustion stems from the ambient external temperature. It can be seen that the ambient air is warmer at all test points for conventional combustion, and it can be seen that the ambient temperature rises as EGR is added during conventional combustion testing. This is due to the fact that as the EGR was swept, actual test cell conditions warmed up throughout the day. LTC does not exhibit this warming because of the unusually cool weather during testing.

Affecting not only the temperature of the inlet air before the compressor, but also the temperature of the cooling air through the intercooler, the ambient air has a large influence on the quality of the intake charge. The difference in intake air temperature represents systematic error in the test sequence conducted for comparison between the two combustion regimes.

Nevertheless, this gives an interesting chance to analyze the effects of ambient air temperature on an internal combustion engine operating in different combustion regimes.

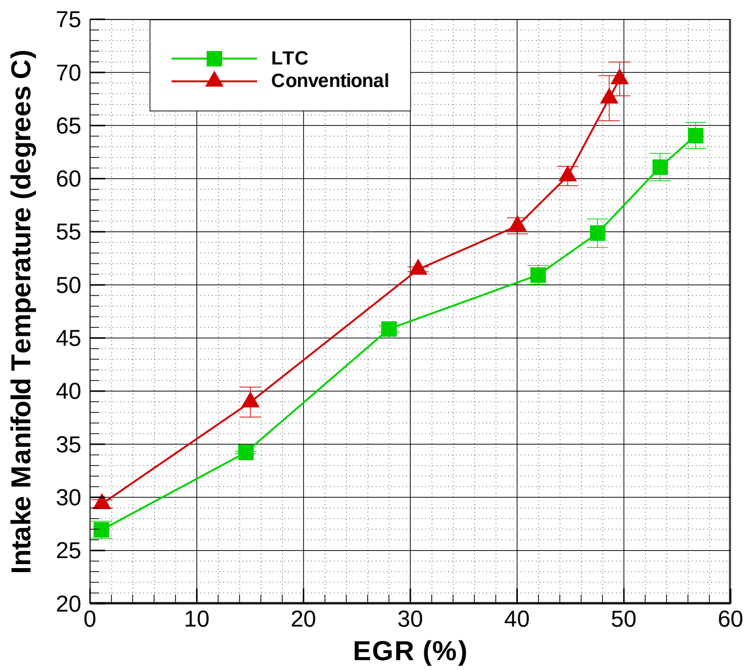

Figure 19. Intake manifold temperature versus EGR for LTC (■) and conventional (▲) combustion.

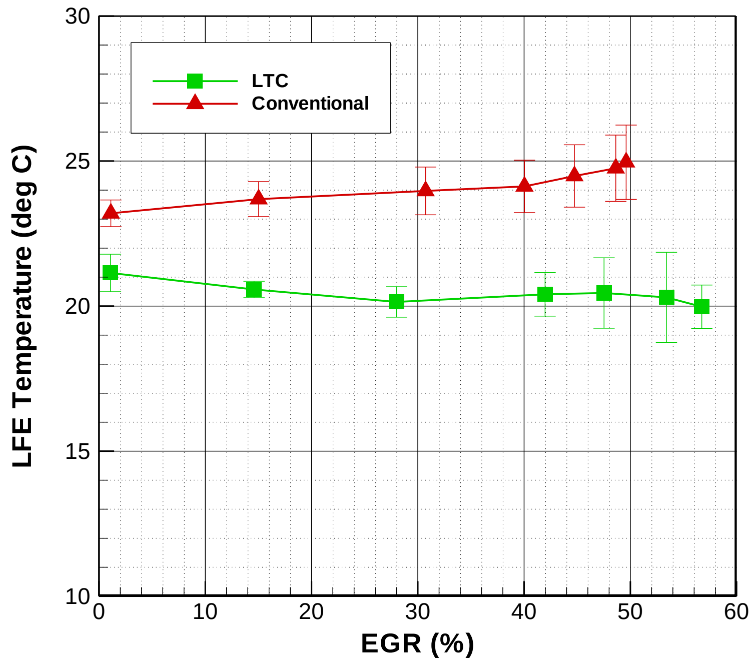

Figure 20. Laminar flow element intake temperature versus EGR for LTC (■) and conventional (▲) combustion.

In order to complete our earlier objective and characterize why LTC is able to flow more EGR than conventional combustion, it is important to use the newly discovered information on ambient air temperatures to analyze the intake air density.

The intake manifold of the engine is a fixed volume with a mixture of fresh air and re-circulated exhaust gases (when EGR is flowing). If it is assumed that this mass of intake air is an ideal gas, the ideal gas equation can be applied:

Dividing by mass yields:

And solving for specific volume yields:

Density is found by:

With these relationships, it is known from Figure 19 that intake air temperature is higher for conventional combustion. This acts to increase the specific volume of the fluid. It is also known that there is no significant difference in intake manifold pressure, which results in no change on the specific volume.

Using these trends to compare our two cases, conventional combustion has a higher specific volume than LTC. Knowing density is the inverse of specific volume, LTC exhibits a higher intake air density.

For a given fixed volume intake (the manifold), LTC is able to induct a larger intake mass of mixed intake charge. Figure 21 illustrates this with LTC having a higher intake air flow rate, especially at higher EGR percentages, where the intake air for conventional combustion is hotter and less dense.

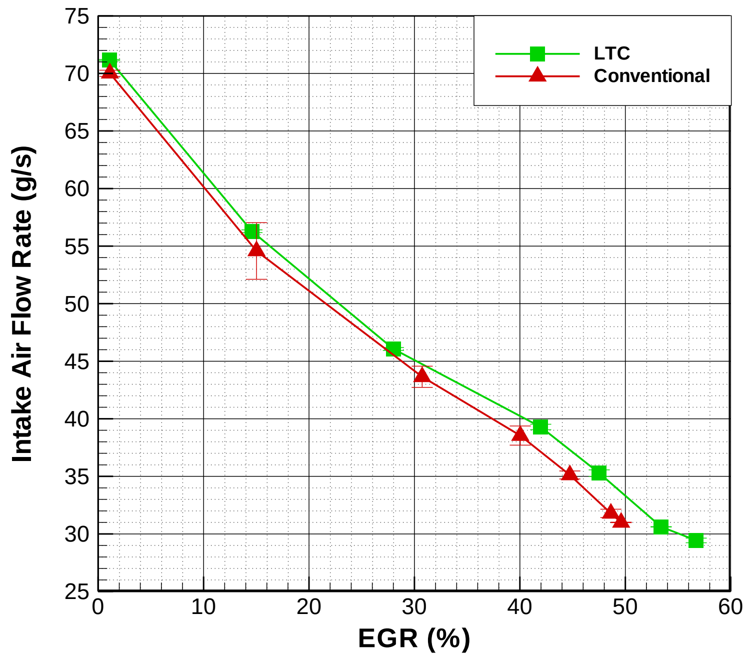

Figure 21. Intake air flow rate versus EGR for LTC (■) and conventional (▲) combustion.

This has been investigated by Ladommatos, et al. [51] in which the effect of increased inlet charge temperature lowers the inlet charge mass due to thermal throttling. In their study, the application of hot EGR was used for thermal throttling; however increased intake air temperature, as in our study, provides the same effect. A consequence of thermal throttling is a reduction in oxygen of the inlet air charge due to the lower charge density. Ladommatos, et al. [58] found that NO_x increased due to a reduction in inlet charge mass, and CO, HC, and smoke increased substantially due to the reduction in the trapped oxygen. Thermal throttling also plays a role in the flow of EGR.

LTC is able to flow more EGR than conventional combustion because of thermal throttling due to the lower air charge density in the conventional regime. The lower charge density in the fixed intake manifold volume allows for less induction of intake air mixture, which includes both mass flows from fresh air and EGR. For these reasons, conventional combustion has lower EGR flow.

Another parameter that is influenced by the EGR is the ignition delay. What seems contrary to conventional thought is that the ignition delay shortens during LTC with increased levels of EGR, Figure 22. Error is large for this combustion regime due to combustion instabilities, but the general trend is that of shorter ignition delay with increased EGR. However, this does not go against conventional wisdom, it merely requires a more in-depth look at how ignition delay is defined.

A previous study [59] compared ignition delay, defined as the location of minimum heat release, to engine ignition delay, defined as the location of 1% mass fraction burned. If the engine ignition delay of LTC is observed in Figure 23, increasing EGR shows a general trend of increasing engine ignition delay – what would originally be predicted with LTC. A possible cause for this abnormality would be cool flame reactions with increased EGR in LTC. These cool flame reactions occur with late injection timings and high EGR levels, where small amounts of combustion actually occur before the main combustion event [60]. It is evident in the ROHR curves of Figure 16 that a small amount of heat release occurs early in the cases of large EGR flows. This heat release is prior to commencement of premixed combustion, and calls into question the conventional definition of ignition delay. If ignition delay is defined as the location of minimum heat release prior to combustion, then the cool flame reactions actually commence earlier combustion, and EGR addition actually decreases the ignition delay. However, if ignition delay is defined as the time until the main combustion event occurs, then a definition such as engine ignition delay is more appropriate. Investigation is currently underway in the lab regarding cool flame reactions by other researchers.

Increased ignition delay in conventional combustion is evident in Figure 24 with increased levels of EGR. Increased diluent in the cylinder lowers available oxygen in the fresh air charge, and requires longer mixing times before combustion can occur. Also as EGR is added, the in-cylinder temperature at the time of injection is seen to drop. The drop in temperature is a function of the lower intake charge density discussed earlier resulting from lower boosting and thermal throttling with EGR. Notice that the discrepancy between engine ignition delay and ignition delay is absent for convention combustion, in Figure 23 and Figure 24, respectively. This is because the cool flame reactions do not take place with the earlier phased combustion.

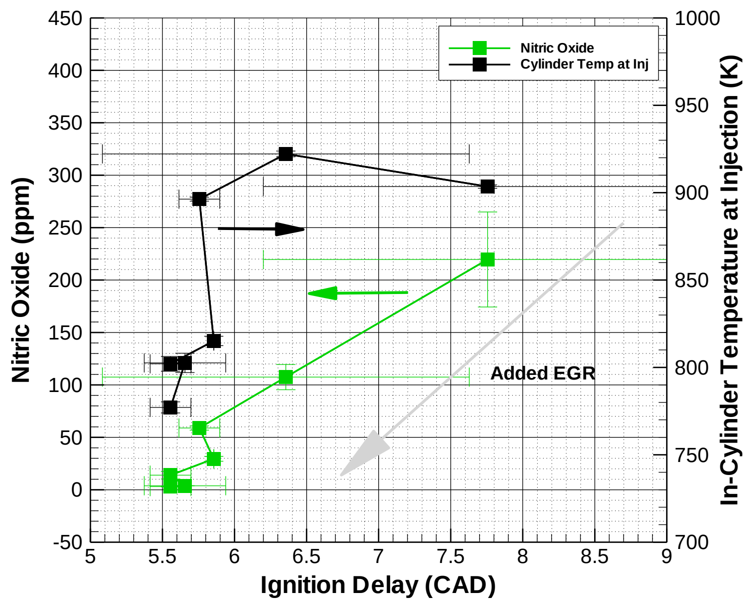

Figure 22. Nitric oxide concentration and in-cylinder temperature at the time of injection versus ignition delay for conventional combustion.

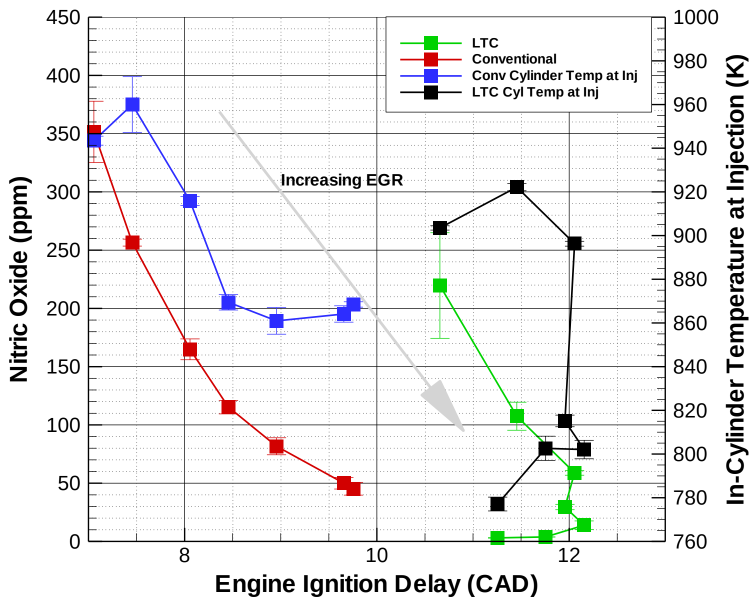

Figure 23. Nitric oxide concentration and in-cylinder temperature at the time of injection versus engine ignition delay for conventional combustion.

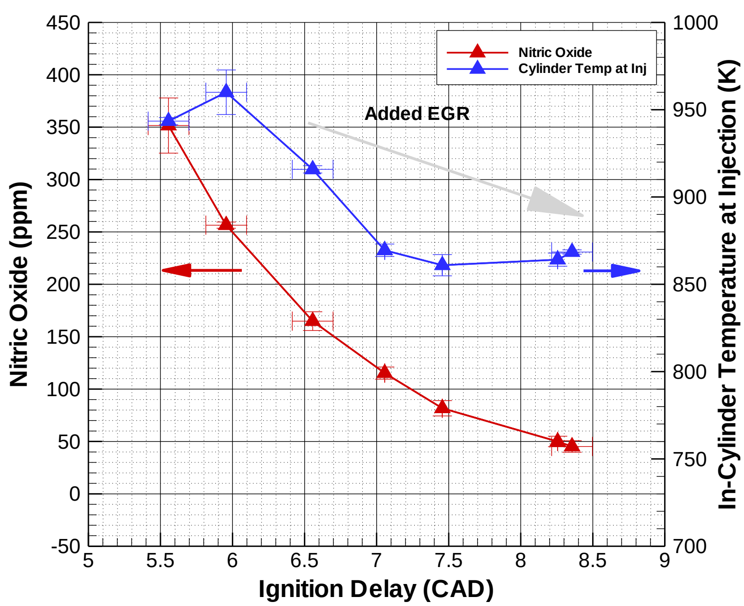

Figure 24. Nitric oxide concentration and in-cylinder temperature at the time of injection versus ignition delay for conventional combustion.

Figure 25 confirms the earlier discussion relating higher in-cylinder temperatures influencing complete combustion. As EGR is added in the conventional and LTC regimes, the maximum in-cylinder temperature is reduced and CO exhaust concentrations tend to increase.

The final cause for the increased concentrations of CO and HC stem from a reduction in combustion efficiency from the increased levels of EGR. Combustion efficiency is calculated by considering the unreacted fuel species in the exhaust, and is plotted in Figure 26 for LTC along with CO and HC species. As combustion efficiency degrades to around 88%, CO and HC production necessarily increase as extreme low temperature combustion occurs. Figure 27 shows the combustion efficiency for conventional combustion with the same range as shown for LTC to illustrate the lack combustion degradation, and the lower resultant products of incomplete combustion.

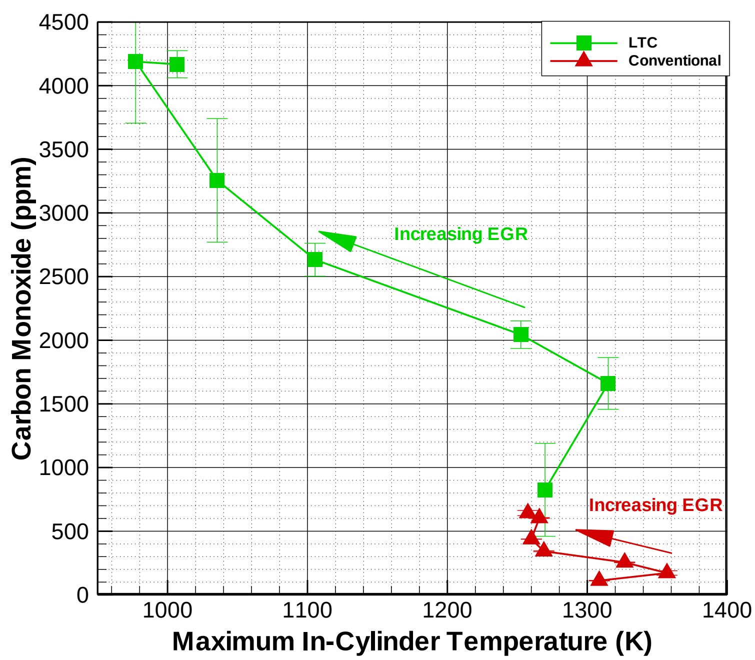

Figure 25. Carbon monoxide versus maximum in-cylinder temperatures as EGR is added.

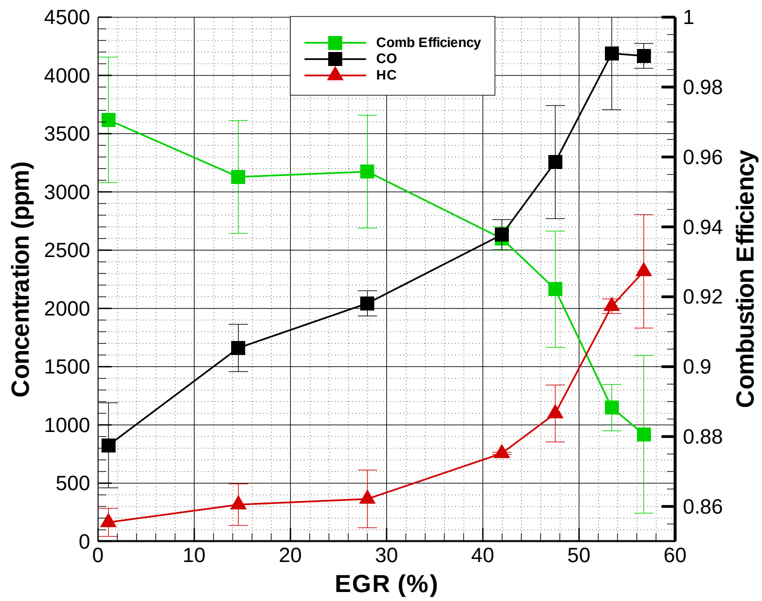

Figure 26. Exhaust CO and HC concentrations and combustion efficiency versus EGR for LTC.

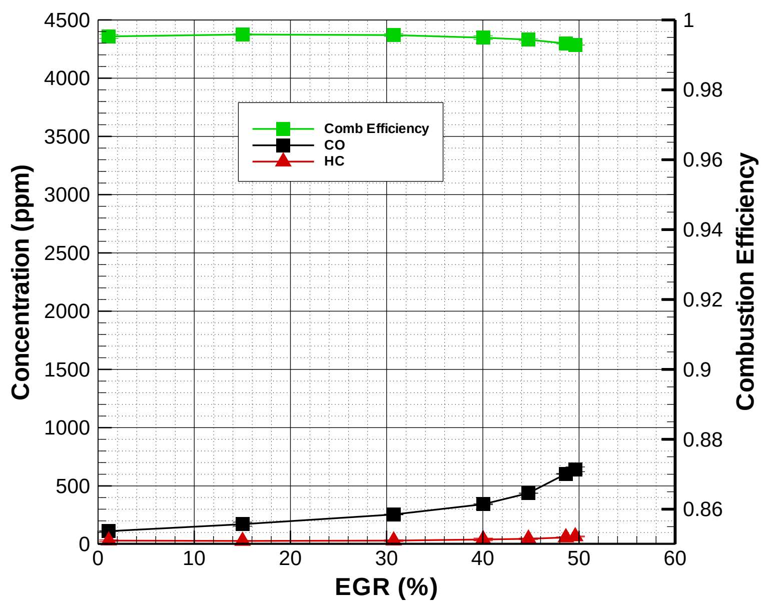

Figure 27. Exhaust CO and HC concentrations and combustion efficiency versus EGR for conventional combustion.

3.1.2 Rail Pressure Sweep

3.1.2.1 Characteristics of Rail Pressure on Emissions

Increases in fuel pressure are used to aid in mixture formation and preparation for combustion [61]. Higher injection pressures create higher jet velocities which enhance atomization and vaporization of the spray. Faster wall jets enhance turbulence for faster mixing and evaporation. All of these effects at higher fuel pressures contribute to reductions in the smoke emissions produced in cylinder [62] - [63]. At the same time, better mixing due to higher fuel pressure increases the rate of heat release due to premixed combustion. This increase in heat release will likely cause an increase in NO concentrations [54].

Consistent with literature, increasing fuel pressure increases the exhaust NO concentrations of conventional combustion in Figure 28. NO concentrations increase 160% with increasing fuel pressure. LTC appears to defeat the typical trend with this scale; however, it still exhibits an increase from 2.6 ppm to 11.1 ppm, a 360% increase in exhaust concentrations. Due to combustion instabilities though, the measurement of NO concentrations has a 4 ppm standard deviation at the highest fuel pressure, making this percent increase somewhat relative.

Also consistent with literature, smoke concentrations are dramatically reduced with increased fuel pressure for both conventional and LTC combustion regimes. Figure 29 shows similar trends between the two combustion cases as the increase in fuel pressure aids in better mixture preparation and faster evaporation of the fuel. Conventional combustion has a 94% reduction in exhaust smoke concentrations, and LTC has a similar 96% reduction in exhaust smoke concentrations.

With only small increases in NO concentrations as fuel pressure is increased, LTC is mostly able to defeat the typical NO – Soot tradeoff seen with conventional combustion in Figure 30. Increases in fuel pressure for LTC dramatically reduce smoke while keeping NO at a minimum.

CO emissions decrease for both conventional combustion and LTC as fuel preparation is enhanced from higher fuel pressures in Figure 31. Conventional combustion CO concentrations are reduced by 18%. This is not easily seen because of the scale required for the LTC CO concentrations. There is not much statistical difference between the reductions in CO for LTC, however an average decrease of 24% is seen as fuel pressure is increased from 500 to 1500 bar.

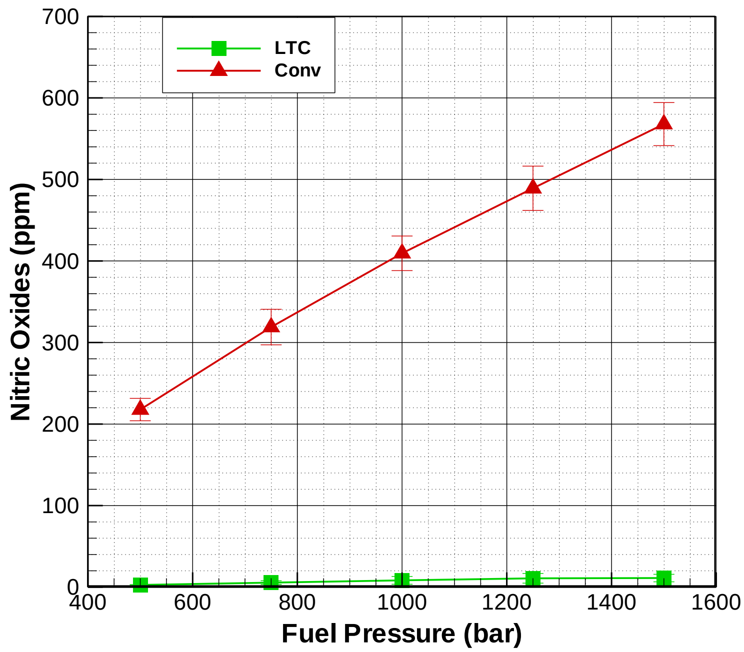

Figure 28. Nitric oxide versus fuel pressure for LTC (■) and conventional (▲) combustion.

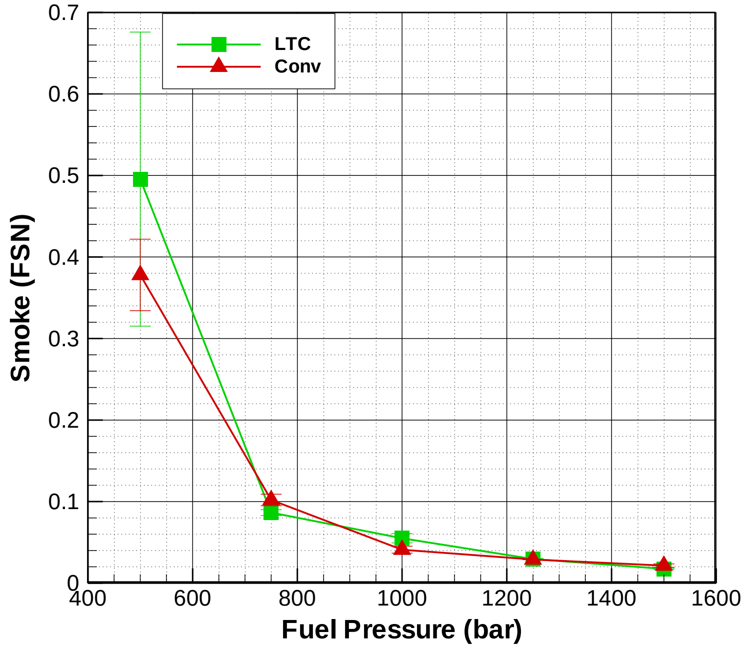

Figure 29. Smoke concentrations versus fuel pressure for LTC (■) and conventional (▲) combustion.

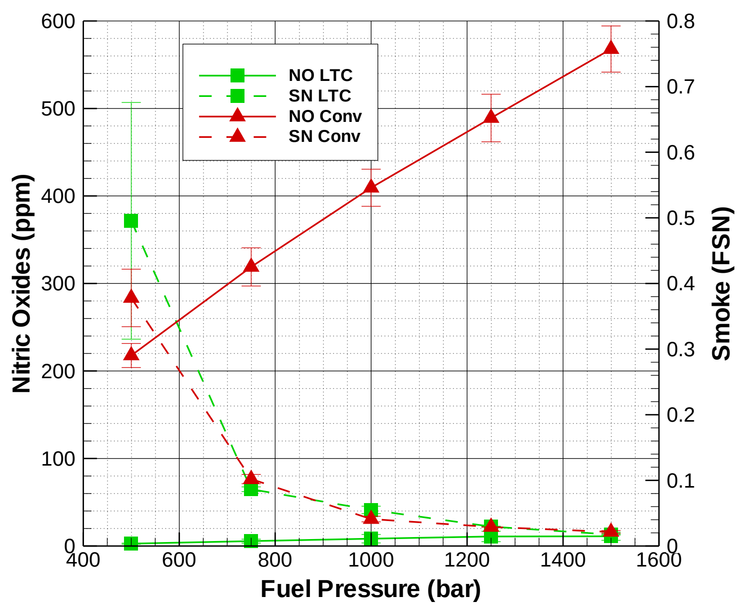

Figure 30. NO - Soot tradeoff versus fuel pressure for LTC (■) and conventional (▲) combustion.

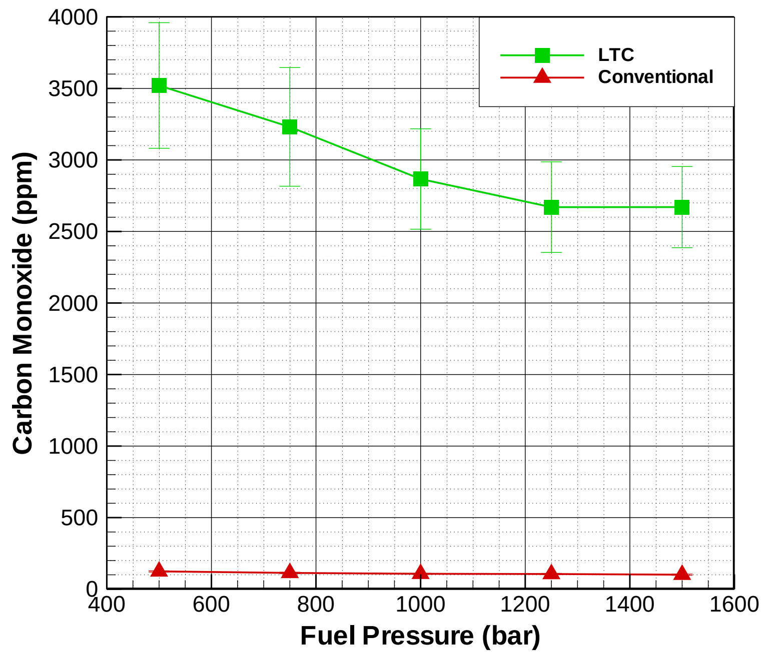

Figure 31. Carbon monoxide versus fuel pressure for LTC (■) and conventional (▲) combustion.

3.1.2.2 Analysis of Rail Pressure on Emissions

In order to verify the previous discussion, let us examine exactly how modifying the fuel pressure can affect the emissions. As fuel pressure is increased, in order to maintain the same injection quantity, the physical pulsewidth of the injector must be decreased. Doing so increases the velocities of the fuel jet, increases atomization (decreases droplet size), and increases in-cylinder mixing to allow for faster preparation for combustion. In alignment with this discussion, Figure 32 shows a trend of decreasing pulsewidth with increased fuel pressure, and a subsequent reduction in ignition delay.

It is interesting to note that the ignition delay changes more for conventional combustion, and both conventional and LTC regimes appear to have similar ignition delays, even with the differences of combustion phasing. However, if the engine ignition delay is examined in Figure 33, it can be seen that by defining the engine ignition delay as the location of 1% mass fraction burned, the engine ignition delay is significantly longer for LTC, allowing more time for mixing before combustion ensues. In alignment with earlier discussion, the discrepancy between ignition delay and engine ignition delay stems from the cool flame reactions that occur in LTC before combustion actually begins and can be seen in Figure 34 as a small amount of heat release occurs before the premixed combustion begins.

Increasing the fuel pressure allows better mixing of the fuel and increases the rate of heat release due to premixed combustion. This is evident for LTC and conventional combustion, shown in Figure 34 and in Figure 35, respectively. As ignition delay is reduced with increasing fuel pressure, the minimum heat release before combustion is advanced, or shifts to the left. Because of better mixing with higher fuel pressure, the heat release due to premixed combustion is increased, and this is seen as an increase in the maximum rate of heat release.

An increase in heat release due to higher premixed combustion yields higher in-cylinder temperatures, and it is this relationship between the increased fuel pressure, heat release, and in-cylinder temperature that causes an increase in NO exhaust concentrations for conventional combustion. Even though the increasing in-cylinder temperatures are present in LTC, the lower adiabatic flame temperature, and overall lower in-cylinder temperatures, Figure 34b, of LTC act to inhibit NO formation [52].

It is also important to note the phasing of heat release for both combustion cases and its relation to NO formation. LTC has an extremely late start of combustion, around 7° ATDC, whereas conventional combustion occurs around -1° ATDC. This combustion phasing is going to necessarily reduce in-cyilnder pressures and temperatures for LTC, which allows for the reduction in NO formation, even with elevated fuel pressures and increased rates of premixed combustion.

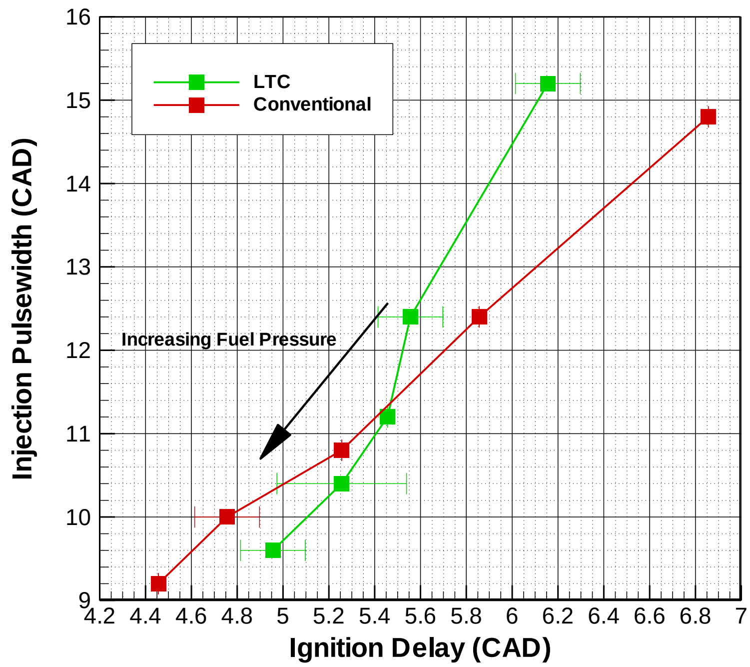

Figure 32. Injection pulsewidth versus ignition delay with increasing fuel pressure.

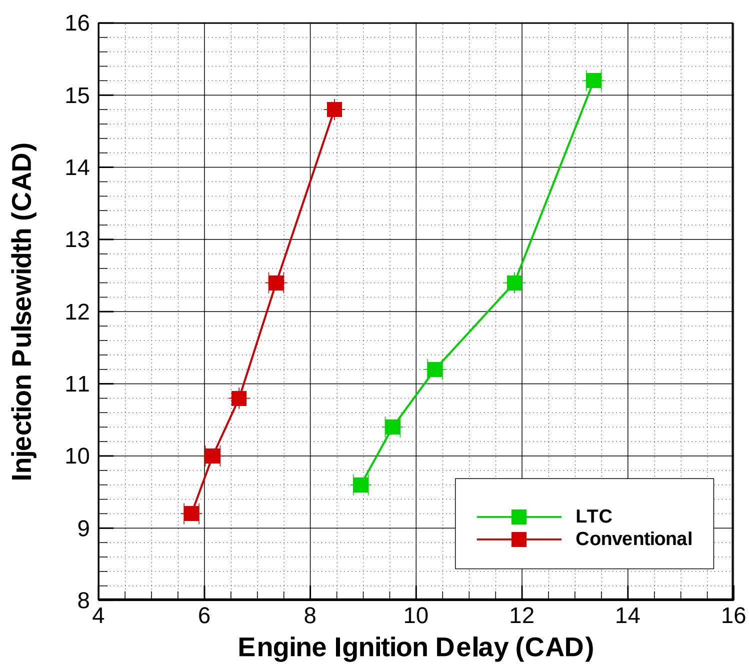

Figure 33. Injection pulsewidth versus engine ignition delay with increasing fuel pressure.

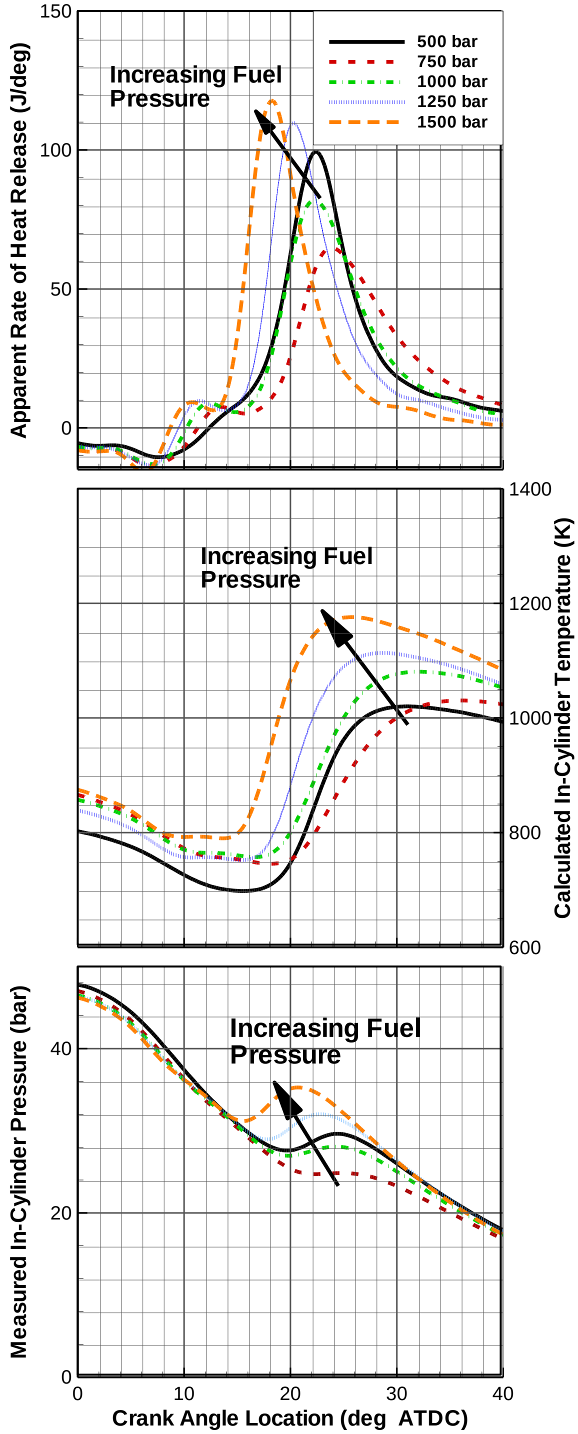

Figure 34. Rate of heat release, in-cylinder temperature, and in-cylinder pressure versus crank angle for increasing fuel pressures in LTC.

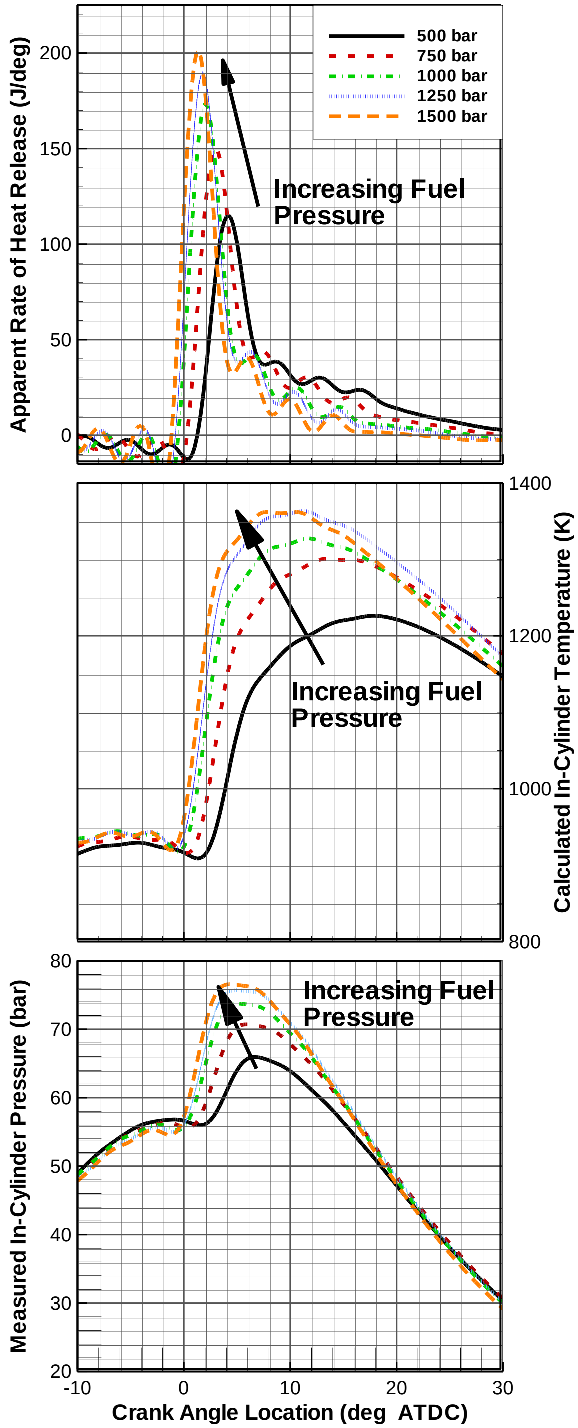

Figure 35. Rate of heat release, in-cylinder temperature, and in-cylinder pressure versus crank angle for increasing fuel pressures in conventional combustion.

The manifold pressures (intake and exhaust, as well as differential manifold pressures) in Figure 36 are drastically different between LTC and conventional combustion. The primary cause for this is that LTC is flowing maximum amounts of EGR (compared to no EGR for conventional combustion during the fuel pressure sweep), reducing the EMP. As exhaust gasses are routed back to the intake manifold, energy that could be extracted as work through the turbocharger is lost. Subsequently, energy that could be added to the fresh intake charge through the compressor is lost, also resulting in a lower IMP. Another influence of manifold pressures could be the in-cylinder pressure at EVO due to differences in combustion phasing.

In-cylinder pressure at EVO timing, Figure 37, shows negligible differences between combustion regimes, which differs from earlier analysis during the EGR sweep in 3.1.1.2, seen in Figure 14. This leads to the assumption that it is mostly the EGR flow rate that affects manifold pressure differences, rather than an effect of combustion phasing and in-cylinder pressure differences between the two combustion regimes.

In Figure 37, as fuel pressure is increased, notice the trend of decreasing in-cylinder pressures at EVO. As fuel is injected at higher pressures, the mixture is better prepared, ignition delay decreases, Figure 32, combustion occurs more rapidly, Figure 34 and Figure 35, and more work is extracted through the expansion stroke, creating lower in-cylinder pressures after expansion has taken place.

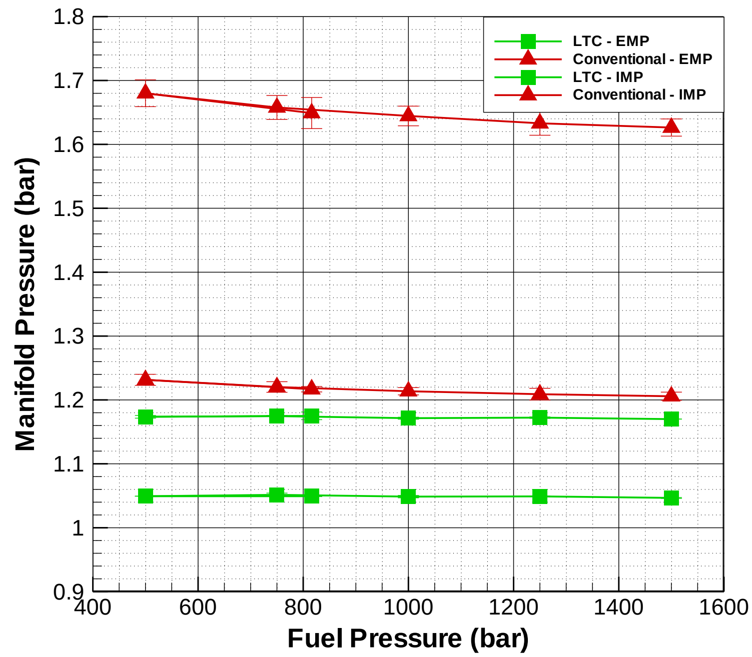

Figure 36. Manifold pressures versus fuel pressure for LTC (■) and conventional (▲) combustion.

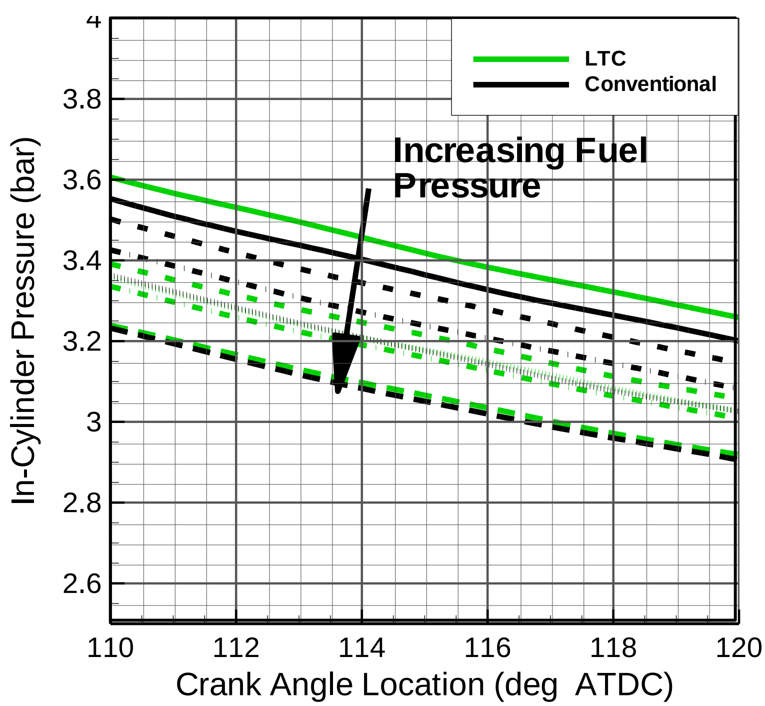

Figure 37. In-cylinder pressure versus crank angle at exhaust valve opening for various fuel pressures.

As fuel pressure is increased, the ignition delay is shortened for each combustion case. With a shorter ignition delay, combustion commences at different crank angle locations depending on the fuel pressure. Figure 38 illustrates that the calculated in-cylinder temperature at the time of injection does not change significantly for conventional combustion, and only slightly increases for LTC.

As such, NO production is more evident from maximum calculated in-cylinder temperatures, rather than the temperature at the time of injection. Figure 39 shows a good correlation between increasing maximum in-cylinder temperatures and increasing NO concentrations for conventional combustion. A 135K increase in maximum in-cylinder temperature produces a 160% increase in NO concentrations. For LTC, a 155K increase in maximum temperature produces a 360% increase in NO exhaust concentrations. As discussed earlier, this percent increase is calculated with NO concentrations that have high standard deviations due to combustion stability problems, and may not be an accurate reflection of the true increase in NO in LTC.

These maximum in-cylinder temperatures are instructive; however they do not give the full scope of the conditions within the cylinder, and more specifically, the adiabatic flame temperatures. Striations in the mixture equivalence ratios will cause different types of combustion to occur. Even though LTC aims to have a completely homogeneous premixed charge, areas of high equivalence ratio will yield regions of smoke production, and areas of high adiabatic flame temperatures will yield regions of NO production.

Also in alignment with an earlier discussion, as ignition delay is shortened, combustion duration is shortened due to better mixture preparation. Figure 40 illustrates the data from the engine test to confirm this. Note that combustion duration is longer for LTC, primarily because the in-cylinder temperatures are lower and combustion commences while the piston moves down, slowing the reaction. Combustion duration is also longer for LTC because of the cool flame reactions taking place, and delaying the main combustion event.

As fuel pressure is increased, shortening combustion duration, more premixed burning creates higher in-cylinder temperatures, and more NO formation. Figure 41 confirms this, showing the relationship between combustion duration – caused by premixed combustion, in-cylinder temperatures, and NO formation.

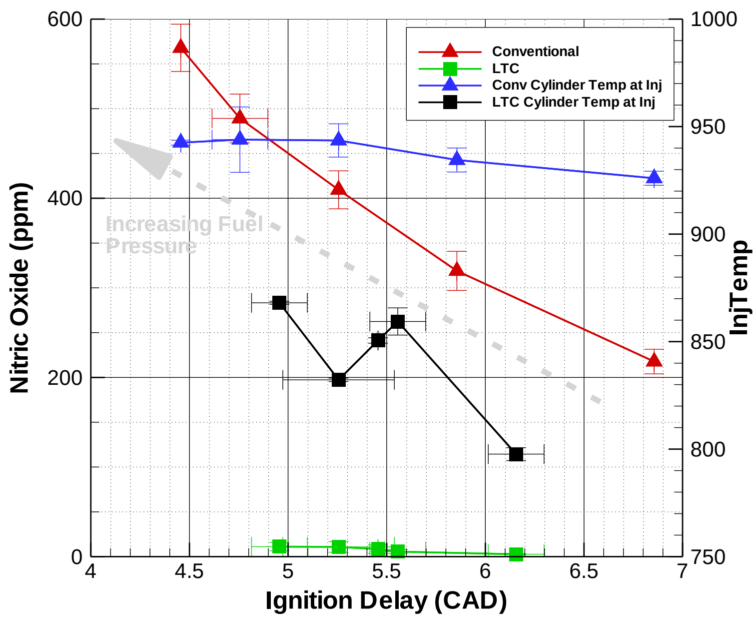

Figure 38. NO concentration and in-cylinder temperature at the time of injection versus ignition delay for increasing fuel pressure.

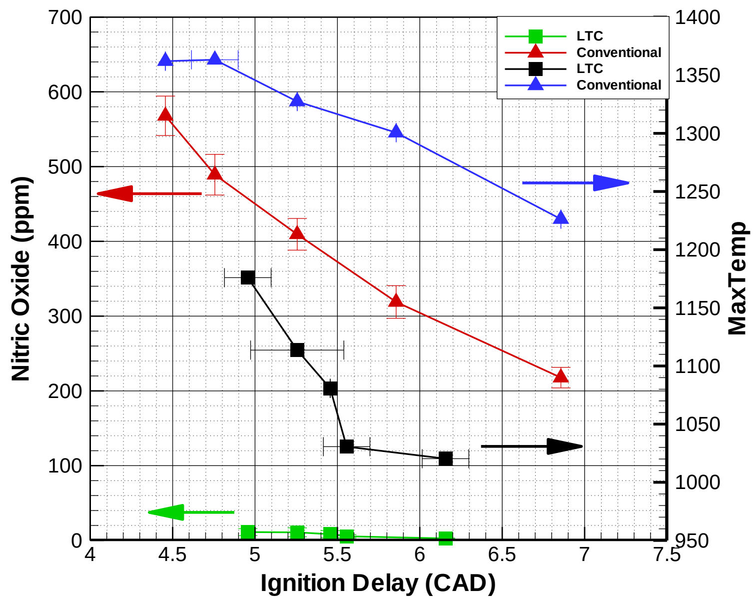

Figure 39. NO concentration and maximum in-cylinder temperature versus ignition delay for increasing fuel pressure.

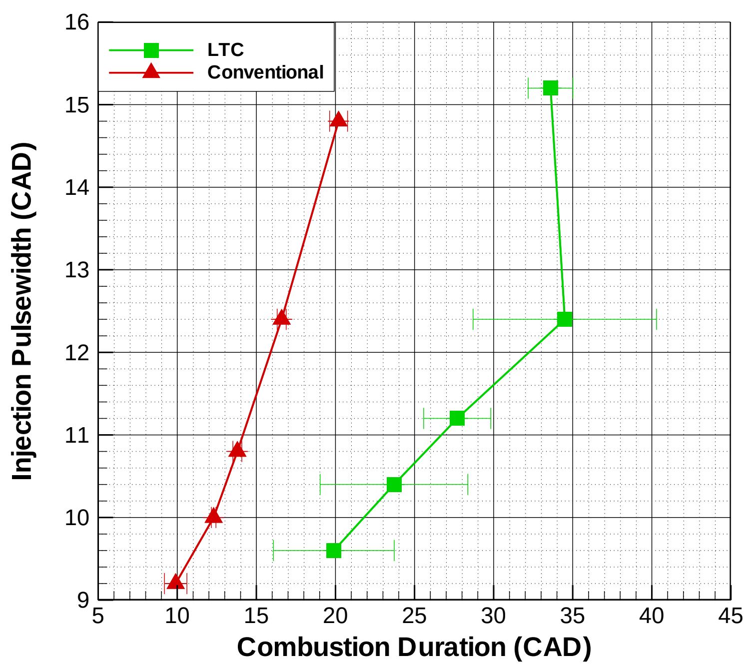

Figure 40. Injection pulsewidth versus combustion duration for increasing fuel pressure.

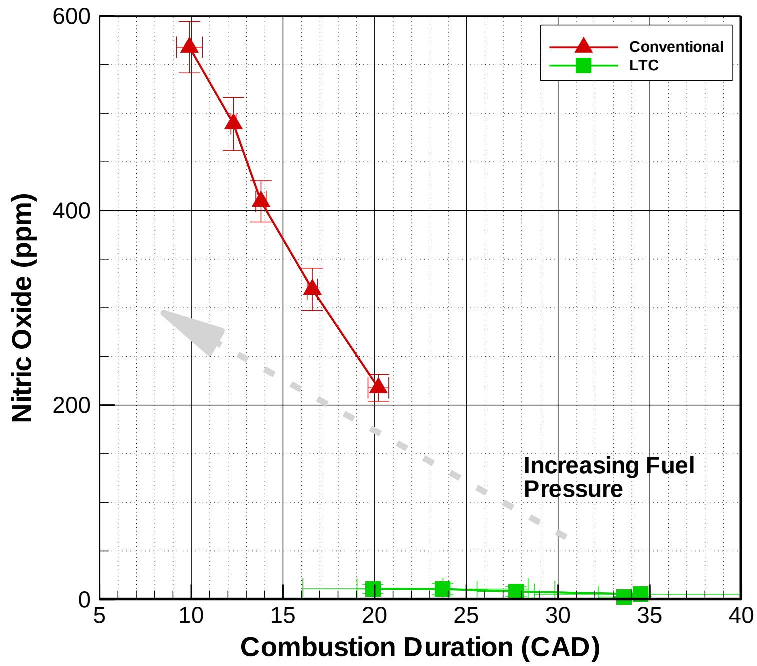

Figure 41. NO concentration versus combustion duration for increasing fuel pressure.

Differences between the EGR activated LTC regime and the conventional combustion regime will cause thermal throttling in LTC due to increased IMT, Figure 42. A consequence of thermal throttling as examined earlier is a reduction in inlet oxygen due to the lower charge density. Ladommatos, et al. found that NO_x increased due to reduction in inlet charge mass, and CO, HC, and smoke increased substantially due to the reduction in the trapped oxygen [52].

No induced systematic error with respect to inlet air temperature, Figure 43 is found for the fuel pressure sweep as was seen earlier in 3.1.1.2. The inlet air temperatures remain relatively the same thorough the fuel pressure sweep and between combustion cases.

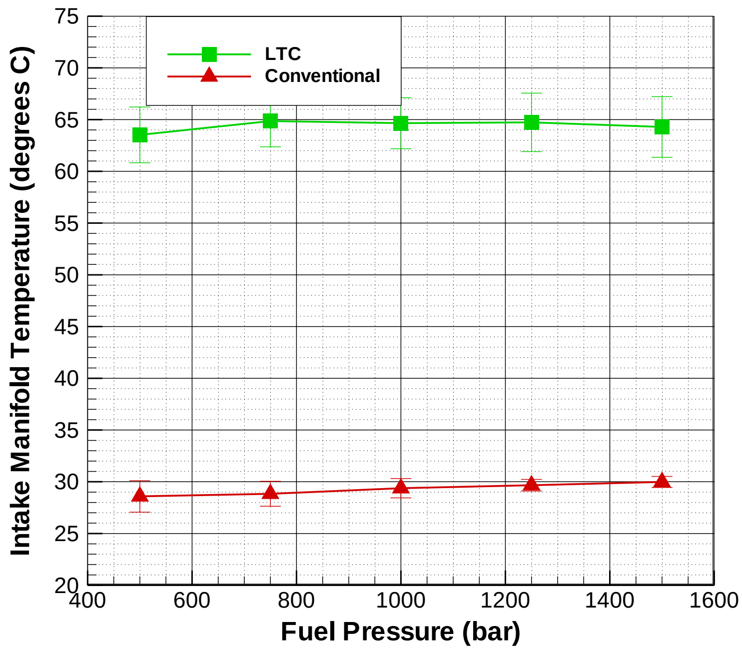

Figure 42. IMT versus fuel pressure for LTC (■) and conventional (▲) combustion.

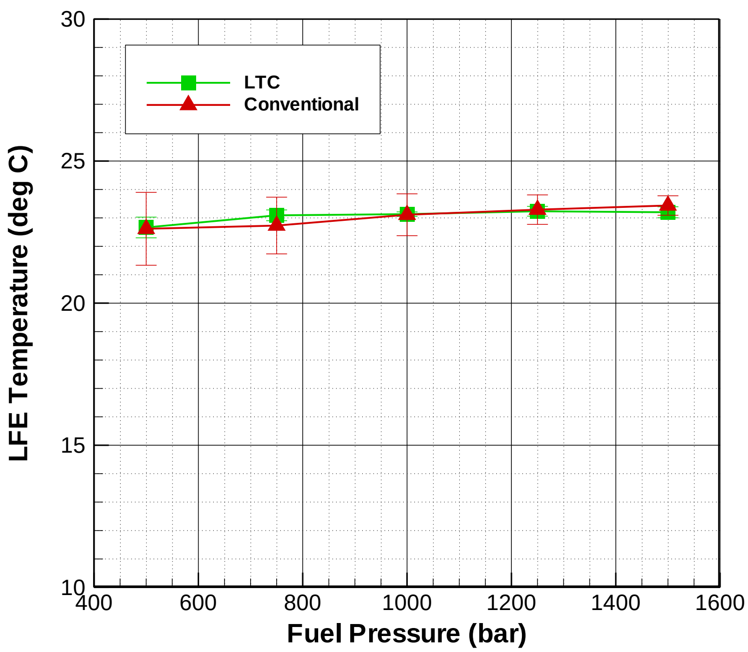

Figure 43. LFE intake air temperature versus fuel pressure for LTC (■) and conventional (▲) combustion.

3.2 Efficiency Considerations

In order to identify the effects that cause changes to the efficiency of a diesel engine, sweeps of EGR and rail pressure are conducted under both LTC and conventional combustion to allow researchers to classify the effects of these engine parameters on the efficiency of a diesel engine.

3.2.1 EGR Sweep

3.2.1.1 Characteristics of EGR on Efficiency

While EGR allows for a substantial reduction in simultaneous NO_x and smoke emissions in diesel engines, the aggressive quantities of EGR required for LTC lower the typical efficiencies seen in conventional combustion diesel engines. Literature [64]-[66], [67]-[73] reports higher indicated specific and brake fuel consumptions and lower thermal and fuel conversion efficiencies at extreme LTC regimes where significant reductions in NO and smoke exhaust concentrations are realized.

In order to identify the effects of EGR on the brake fuel conversion efficiency, let us first examine the general trends in Figure 44. As EGR flows into the intake charge, brake fuel conversion efficiency increases 4% for conventional combustion and efficiency decreases 6.8% for LTC at the onset of extreme low temperature combustion. This does not necessarily suggest that the application of EGR increases the brake fuel conversion efficiency in a conventional diesel combustion engine, and will be discussed in the analysis section.

Brake fuel conversion efficiency is calculated using the power of the engine, the mass flow rate of the fuel, and the lower heating value of the fuel:

Since power is a term derived from the engine torque and speed, the engine torque versus EGR should directly correlate from the brake fuel efficiency graph if the fuel flow rate remains the same. Figure 45 shows engine torque as a function of EGR with nominal torque values around 68 N-m, and correlates well to the brake fuel conversion efficiency graph since the fuel flow rate remains the same throughout the EGR sweep, Figure 46.

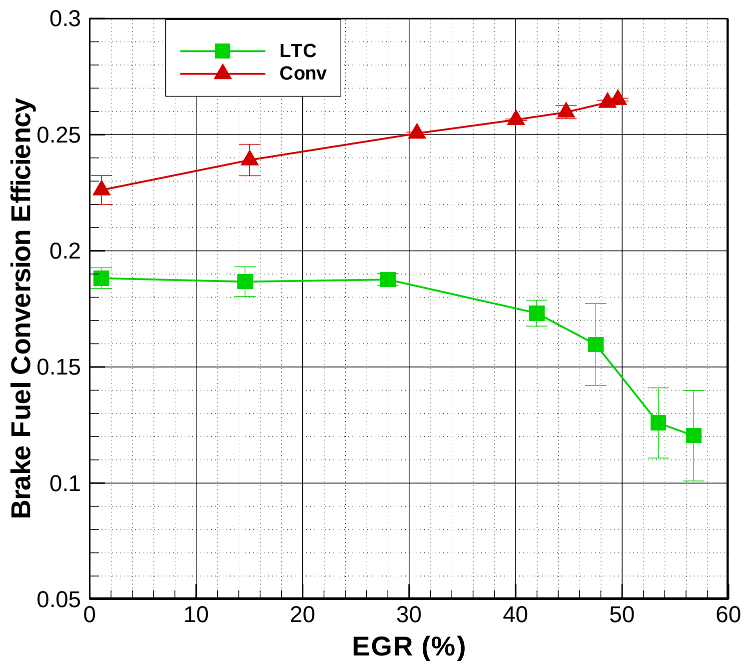

Figure 44. Brake fuel conversion efficiency versus EGR for LTC (■) and conventional (▲) combustion.

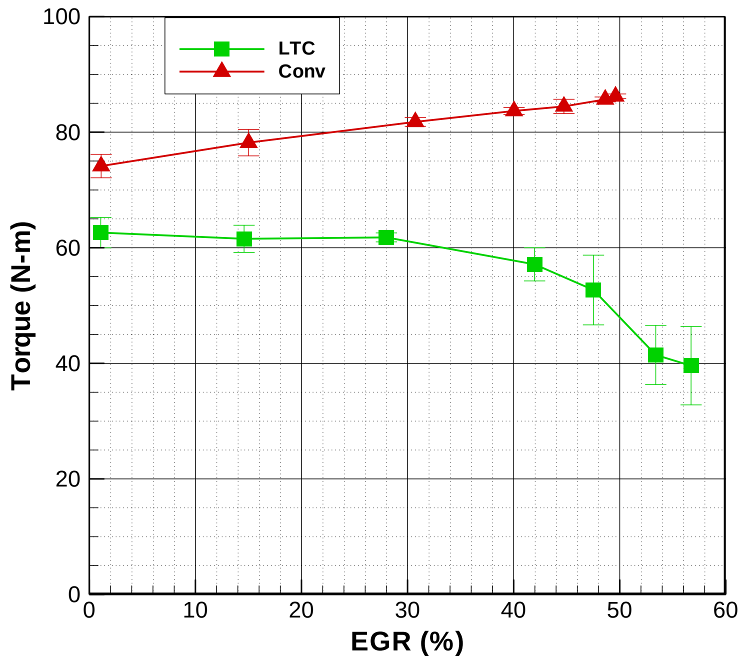

Figure 45. Engine torque versus EGR for LTC (■) and conventional (▲) combustion.

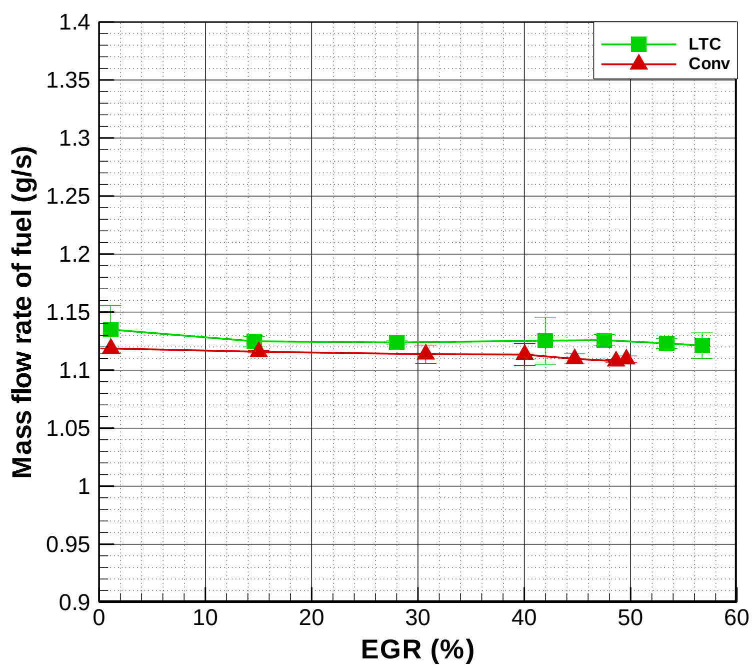

Figure 46. Mass flow rate of fuel versus EGR for LTC (■) and conventional (▲) combustion.

3.2.1.2 Analysis of EGR on Efficiency

In order to analyze the general behavior of Figure 44, a general simple approach, similar to that used in a previous study [49], will be applied. This approach to analyzing brake fuel conversion efficiency will determine if the trends stem from:

- Decreases in gross indicated mean effective pressure, $IMEP_g$
- Increases in pumping mean effective pressure, PMEP[†]
- Increases in friction mean effective pressure, FMEP[‡]

or a combination of the three effects. Figure 47 and Figure 48 show the various mean effective pressures for LTC and conventional combustion, respectively.

First, notice the similar trends of BMEP to the brake fuel conversion efficiency. Since brake work defines BMEP and fuel flow rates remain constant, these two should be, and are similar.

Secondly, the BMEP is the resultant parameter after the diminishing effects of the combustion work are added. In other words, BMEP is the sum of the gross indicated work produced in the cylinder, $IMEP_g$, reduced by the pumping work, PMEP, and the friction work, FMEP.

Overall, the addition of EGR will reduce the PMEP with no turbocharger intervention; however, cases do arise [74] where the addition of EGR increases PMEP due to the calibration of the system overdriving the turbocharger to increase boost. The addition of EGR ultimately reduces PMEP by a reduction in manifold pressure differentials, Figure 12, and tends to promote an increase in brake fuel conversion efficiency. Because of this, brake fuel conversion efficiency should be increased for both LTC and conventional combustion due to a reduction in PMEP. Since LTC exhibits decreasing brake fuel conversion efficiency with increased EGR, there are other forces negating the effects of the reduced PMEP.

[†] Calculated as the difference between $IMEP_g$ and net indicated mean effective pressure, or $IMEP_n$
[‡] Calculated as the difference between $IMEP_n$ and brake mean effective pressure, or BMEP

FMEP tends to increase as EGR is added to the system for both LTC and conventional combustion. This seems to go against what is expected, since frictional losses created by combustion [75] should be reduced with lowered in-cylinder pressures and later combustion phasing, Figure 16 and Figure 17, with increased EGR. Another factor which affects FMEP is drag from engine accessories, such as the common rail fuel pump, alternator, etc, and do not change with the EGR sweep. Perhaps this error stems from the calculation of FMEP. All of the indicated parameters are reported for only one cylinder (only cylinder #1 is instrumented with an in-cylinder pressure transducer); whereas BMEP is an engine parameter. It is possible that the instrumented cylinder behaves differently than the average of the four cylinders, resulting in differences between the net indicated and brake pressures. Whichever the case, FMEP generally increases, which acts to decrease BMEP and subsequently reduces brake fuel conversion efficiency.

The last contributor to BMEP is the available work from all of the internal processes in a combustion engine, the $IMEP_g$. It is influenced by the engine's thermal efficiency, combustion efficiency, fuel-air ratio, air density, volumetric efficiency, and fuel heating value. Some of these values stay constant, but most important to influencing the $IMEP_g$ is the thermal efficiency, combustion efficiency, fuel-air ratio, and air density.

Thermal efficiency is influenced by the phasing of combustion, rate of heat transfer, and the mixture properties. In this application of LTC through a sweep of EGR, phasing of combustion and heat transfer rates are the primary sources which diminish the thermal efficiency. Through the work of a previous study [49], it was found that in the case of late injection timing LTC, degradations of thermal efficiency were found to be the primary factor for the decrease in brake fuel conversion efficiency. Even more, as EGR is swept during LTC, the combustion becomes phased even later in the expansion stroke, leading to a further loss in thermal efficiency, a decrease in $IMEP_g$, and ultimately causes the degradation in brake fuel conversion efficiency with increasing EGR. Presumably as EGR is added to the conventional case, the later phasing of combustion (for the earlier injection timings) aids in increasing the thermal efficiency, increasing the $IMEP_g$, and results in the increasing brake fuel conversion efficiency.

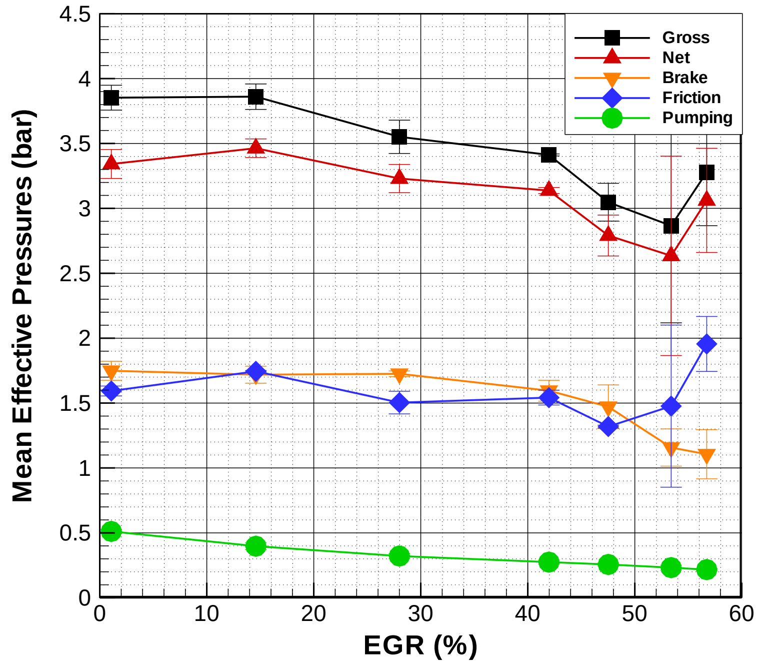

Figure 47. Gross, net, pumping, friction, and brake mean effective pressures versus EGR for LTC.

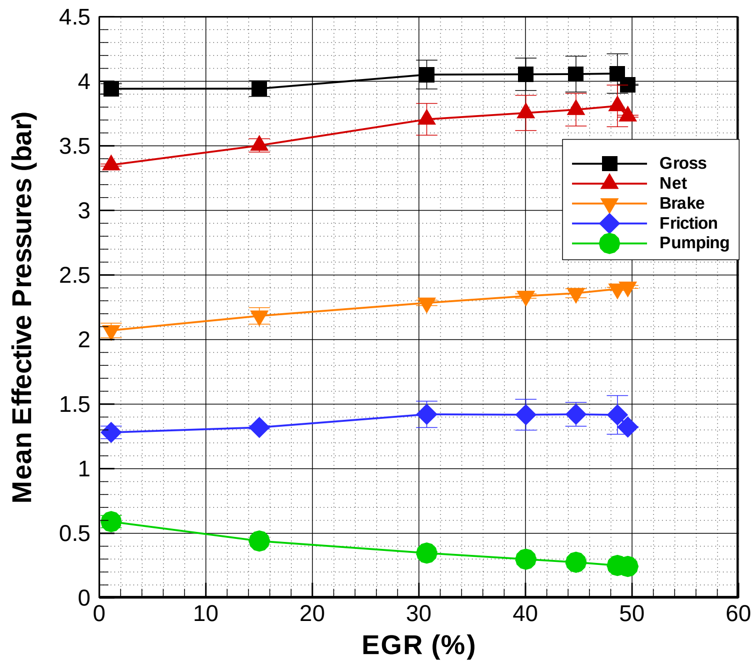

Figure 48. Gross, net, pumping, friction, and brake mean effective pressures versus EGR for conventional combustion.

Described earlier in the emissions section with Figure 26, combustion efficiency, Figure 49, is calculated considering unreacted fuel species in the exhaust. As EGR is added to the system, the combustion efficiency deteriorates for the extreme LTC cases, and CO and HC concentrations dramatically increase. The dramatic degradation of combustion efficiency fails to fully utilize the energy released by the fuel, and will act to reduce the brake fuel conversion efficiency. However, simulations from [75] have shown that combustion phasing is the dominant parameter which dictates the maximum attainable fuel efficiency of the engine. Also, from [49], the later phasing of combustion plays a larger role in the deteriorated brake fuel conversion efficiency than the loss in combustion efficiency.

Also influencing $IMEP_g$ is the fuel-air ratio. The inverse of this relationship, the air/fuel ratio is diminished in the EGR sweep, Figure 50, as diluents are added back into the cylinder, ultimately displacing fresh intake air. Because the mass flow rate of fuel is constant during the sweep, the increase in fuel-air ratio stem from the loss of fresh air in the charge. The increase in the fuel-air ratio will likely result in a decrease in thermal efficiency and brake fuel conversion efficiency.

As previously investigated in 3.1.1.2, the intake air density is reduced with increased levels of EGR due to the effects of thermal throttling. Reductions in air intake density act to reduce the $IMEP_g$ for both combustion modes, with a stronger emphasis on conventional combustion because of the increased intake air temperature caused by systematic experimentation error. However, this effect does not appear in the trends in Figure 48.

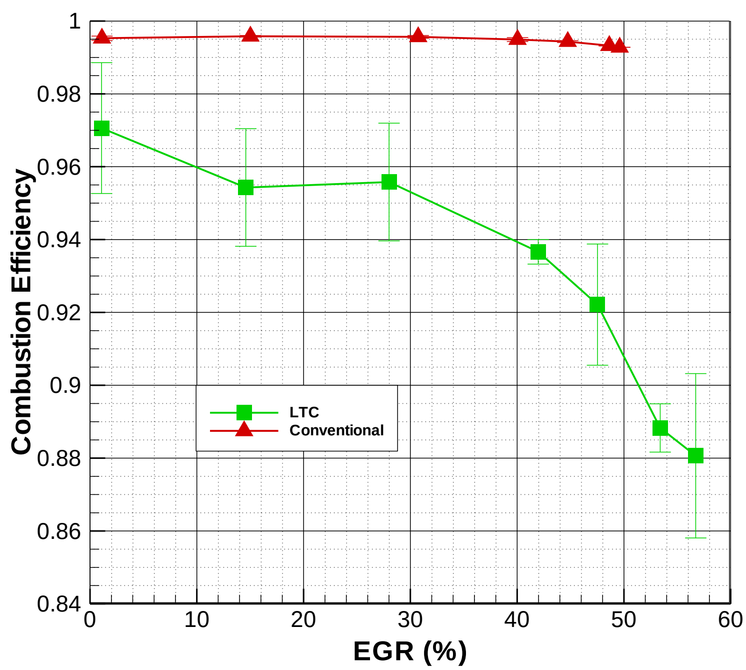

Figure 49. Combustion efficiency versus EGR for LTC (■) and conventional (▲) combustion.

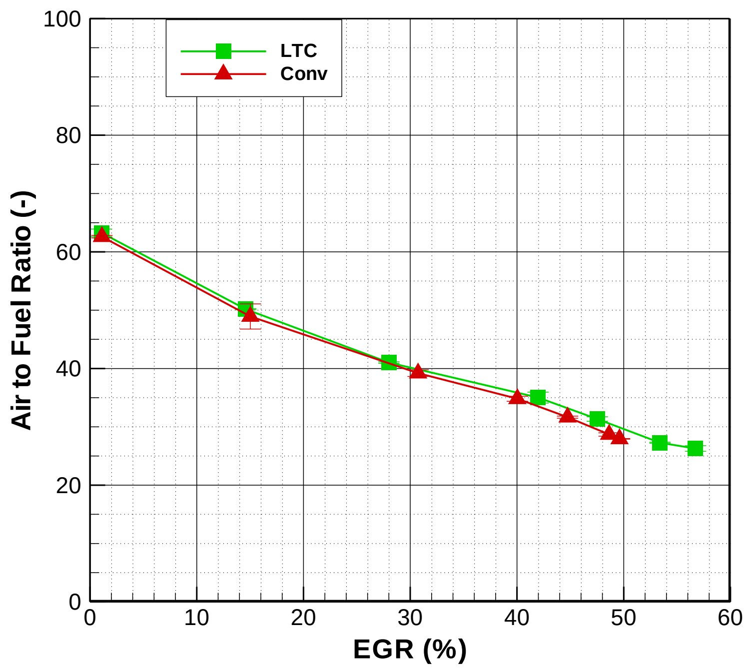

Figure 50. Air to fuel ratio versus EGR for LTC (■) and conventional (▲) combustion.

As EGR is added to the conventional combustion system, brake fuel conversion efficiency is seen to increase. Since the mass flow rate of fuel is constant throughout the EGR sweep, the engine power must be influencing the efficiency. Figure 51 details the engine torque versus the location of 50% mass fraction burned. As EGR is added to the system, the 50% mass fraction burned location is retarded later in the expansion stroke for both combustion regimes. Since the injection timings were not originally optimized for brake fuel conversion efficiency, it is not appropriate to state that EGR enhances brake fuel conversion efficiency, and subsequently engine torque, for conventional combustion; however, literature [52] has shown maximum fuel conversion efficiencies at moderate charge dilution levels and slightly retarded injection timings.

As EGR is added to the LTC system, brake fuel conversion efficiency decreases. Previous work [49] details that the primary cause for the reduction in brake fuel conversion efficiency in a high EGR, retarded LTC regime, is the reduction in thermal efficiency caused by later phased combustion. As EGR is added in Figure 51, the 50%

mass fraction burned location is retarded to as much as 28° ATDC. Ultimately, the late phasing of the combustion fails to take full advantage of the expansion stroke, lowering brake fuel conversion efficiency and producing less torque.

Also important in determining the proper phasing of the combustion is the EMT, displayed earlier in Figure 13. Conventional combustion is able to extract more energy out of the combustion mixture through the expansion stroke. As a result, the EMT for conventional combustion is generally lower than LTC.

Additionally, as more EGR flows from the exhaust to the intake manifold, the amount of energy captured by the turbo is reduced. This necessarily relates to a lower turbine speed, Figure 18, and a lower boosted IMP, visible earlier in Figure 11.

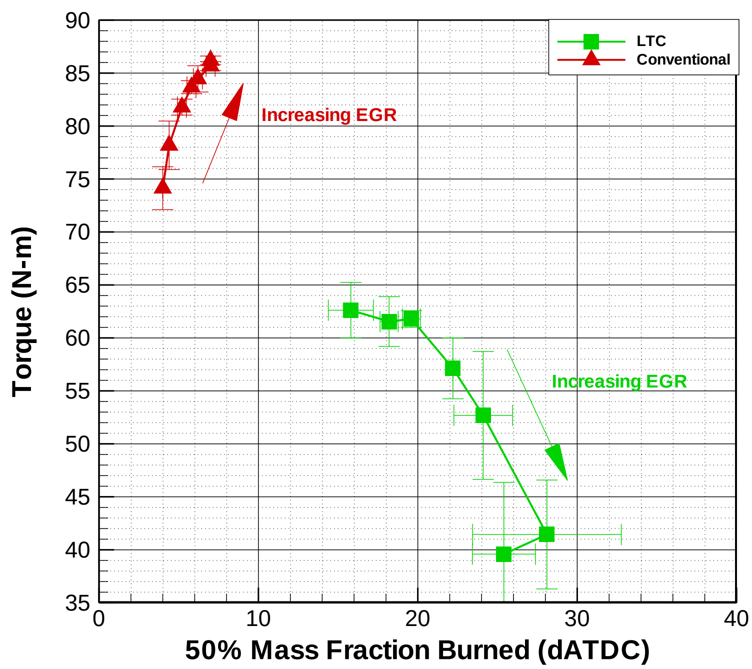

Figure 51. Torque versus 50% mass fraction burned with increased EGR for LTC (■) and conventional (▲) combustion.

3.2.2 Rail Pressure Sweep

3.2.2.1 Characteristics of Rail Pressure on Efficiency

As fuel pressure is increased, brake fuel conversion efficiency, Figure 52, drops 4% in conventional combustion, while LTC reveals a maximum brake fuel conversion efficiency at 1250 bar fuel pressure, an efficiency increase of 2.3%.

Torque, Figure 53, follows the same trend for brake fuel conversion efficiency because fuel flow rate, Figure 54, is kept constant throughout the fuel pressure sweep by modifying the injection pulsewidths.

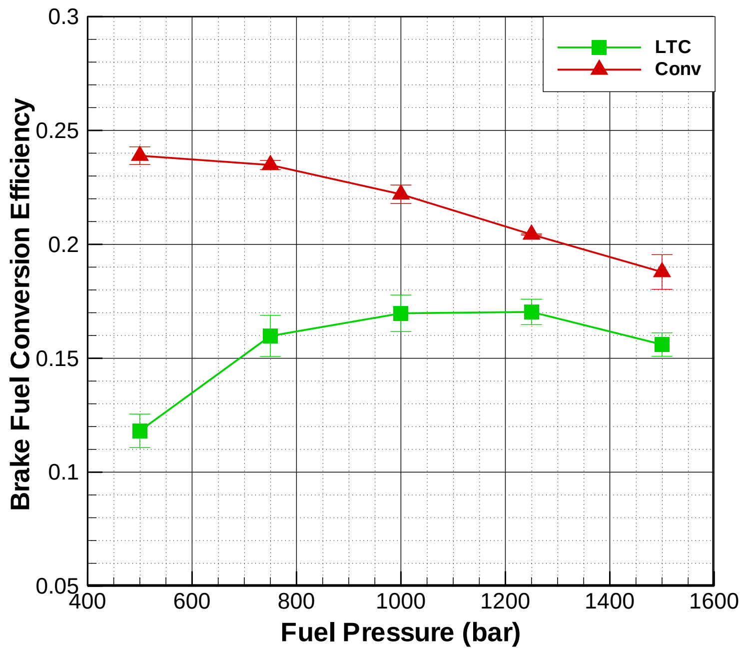

Figure 52. Brake fuel conversion efficiency versus fuel pressure for LTC (■) and conventional (▲) combustion.

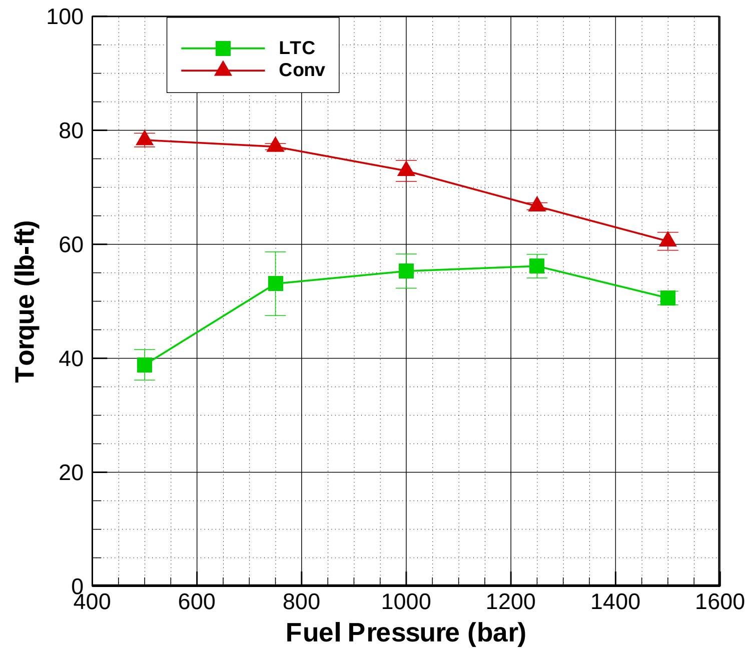

Figure 53. Torque versus fuel pressure for LTC (■) and conventional (▲) combustion.

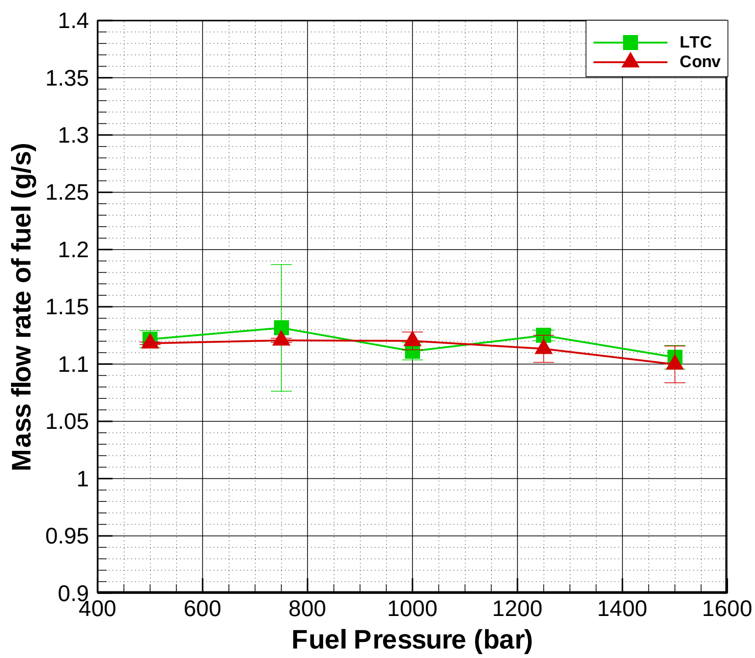

Figure 54. Mass flow rate of fuel versus fuel pressure for LTC (■) and conventional (▲) combustion.

3.2.2.2 Analysis of Rail Pressure on Efficiency

Following a similar analysis from 3.2.1.2, brake fuel conversion efficiency will be analyzed based upon a simple approach of the mean effective pressures. This approach to analyzing brake fuel conversion efficiency will determine if the trends stem from: decreases in $IMEP_g$, increases in PMEP, increases in FMEP, or a combination of the three effects. Figure 55 and Figure 56 show the various mean effective pressures for LTC and conventional combustion, respectively, as a sweep of the fuel pressure is performed.

BMEP and brake fuel conversion efficiency share similar trends since the fuel flow rates remain constant. Because of this, these two should be, and are similar

Since no other engine parameters change with the sweep of fuel pressure, including EGR, VGT, etc, PMEP should stay the same with increasing fuel pressure. Figure 57 shows that the differential manifold pressures, an indicator of pumping work, remain relatively constant as fuel pressure is swept. It is interesting to note that LTC has much lower differential pressures due to the high EGR flow between the manifolds, which acts to substantially decrease PMEP over the pressure sweep. This will act to increase BMEP and increase the fuel conversion efficiency for LTC.

FMEP tends to increase as fuel pressure is increased for both LTC (except for the first decrease as fuel pressure increases from 500 bar to 750 bar) and conventional combustion. This is consistent with the better mixing and increased combustion rates that occur with increased fuel pressure, since frictional losses created by combustion [75] should be increased with higher in-cylinder pressures and earlier combustion phasing, Figure 34 and Figure 35. Another factor which affects FMEP is drag from engine accessories, such as the common rail fuel pump, alternator, etc. Increases in fuel pressure from 500 bar to 1500 bar create a noticeably higher load on the engine, especially prevalent while in the engine test cell as the fuel pressure is increased. This will act to increase the FMEP as fuel pressure is increased, which acts to decrease BMEP and subsequently reduce brake fuel conversion efficiency.

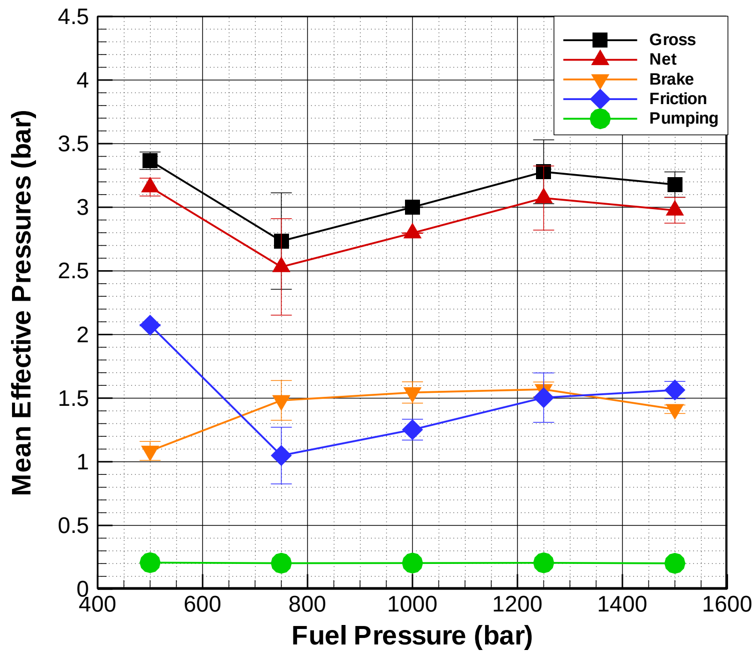

Figure 55. Gross, net, pumping, friction, and brake mean effective pressures versus fuel pressure for LTC.

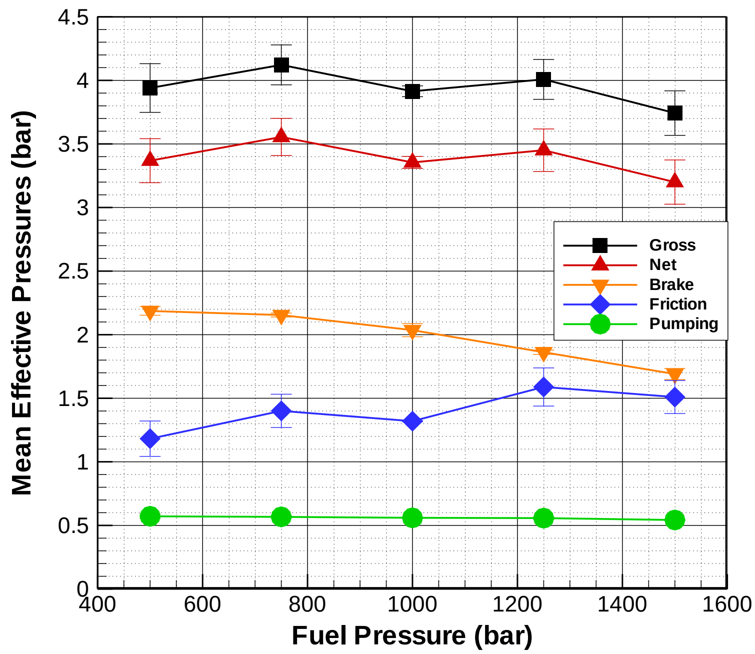

Figure 56. Gross, net, pumping, friction, and brake mean effective pressures versus fuel pressure for conventional combustion.

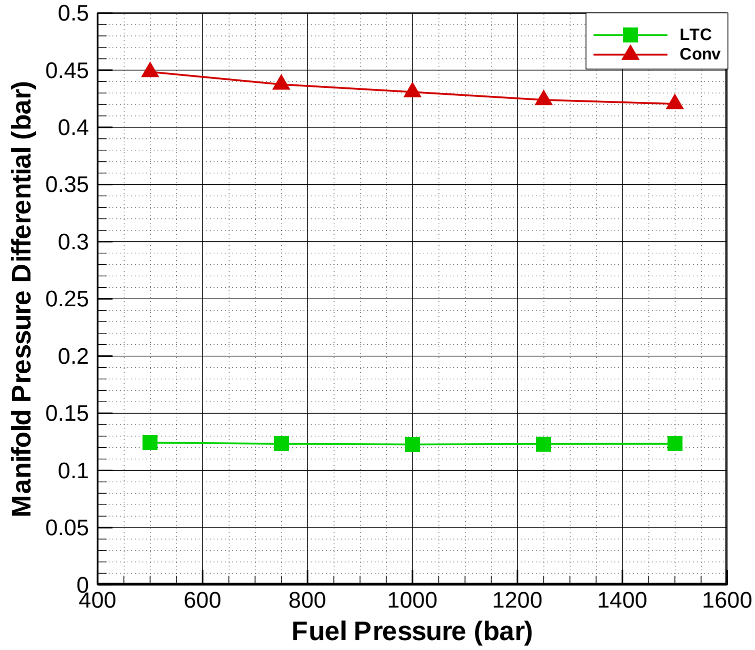

Figure 57. Manifold pressure differential versus fuel pressure for LTC (■) and conventional (▲) combustion.

The last contributor to BMEP is the $IMEP_g$. It is influenced by the engine's thermal efficiency, combustion efficiency, fuel-air ratio, air density, volumetric efficiency, and fuel heating value. Some of these values stay constant, but most important to influencing the $IMEP_g$ in a sweep of fuel pressure is the thermal efficiency, combustion efficiency, and air density.

The thermal efficiency is influenced by the phasing of combustion, rate of heat transfer, and the mixture properties. Phasing of combustion and heat transfer rates are the primary sources which affect the thermal efficiency through the sweep of fuel pressure. Through the work of a previous study [49], it was found that in the case of late injection timing for LTC, degradations of thermal efficiency were found to be the primary factor for the lower brake fuel conversion efficiency (compared to conventional combustion) because of the late phasing of combustion. This explains why thermal efficiency, $IMEP_g$, and brake fuel conversion efficiency are all generally lower for LTC.

However, as a sweep of fuel pressure is conducted and fuel pressure is increased, the combustion is seen to advance, Figure 58, due to quicker fuel mixture preparation. Advancing the combustion for LTC increases the torque output as proper combustion phasing is achieved. Since the injection timings were originally not optimized for brake fuel conversion efficiency, it is not appropriate to state that fuel pressure directly enhances brake fuel conversion efficiency in LTC, and subsequently engine torque; however, the advanced combustion timing helps mitigate the extremely late combustion phasing, and allows for earlier combustion. This aids in torque production as more of the combustion work is fully extracted over the expansion stroke. Ultimately, increasing fuel pressure advances combustion phasing, increases engine torque output, increases thermal efficiency, increases $IMEP_g$, and increases the brake fuel conversion efficiency. This is the primary factor for the trend in increasing brake fuel conversion efficiency in LTC.

The opposite trend is visible for conventional combustion. As combustion is advanced, phasing of the 50% mass fraction burned location is moved away from peak torque production, reducing thermal efficiency, reducing $IMEP_g$, and reducing brake fuel conversion efficiency with increased fuel pressure.

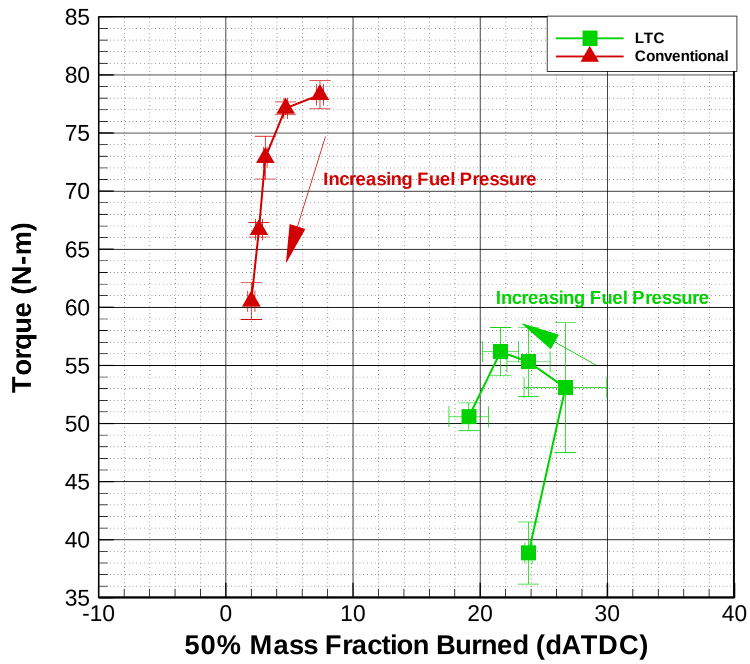

Figure 58. Torque versus 50% mass fraction burned with increased fuel pressure for LTC (■) and conventional (▲) combustion.

Also affecting $IMEP_g$ is the combustion efficiency, Figure 59, and is calculated considering unreacted fuel species in the exhaust. As fuel pressure is increased, the combustion efficiency increases almost 11% for LTC, and CO and HC concentrations decrease 35% and 76%, respectively. The increase in combustion efficiency stems from better mixture preparation, allowing fuel to find available oxygen within the cylinder full of exhaust diluent, and increasing the energy released by the fuel. The increase in combustion efficiency will act to increase the brake fuel conversion efficiency. Combustion efficiency remains the same throughout the fuel pressure sweep for conventional combustion and results in no change of brake fuel conversion efficiency.

Also influencing $IMEP_g$ is the fuel-air ratio. The inverse of this relationship, the air/fuel ratio, Figure 60, is constant throughout the fuel pressure sweep, and will not affect the brake fuel conversion efficiency as fuel pressure is swept. However, differences in air-fuel ratio are seen between the two combustion modes, and analysis can be performed between the two. As the fresh intake oxygen is reduced with LTC, it will tend to have higher fuel-air ratios than conventional combustion. An increase in fuel-air ratio will tend to decrease the thermal efficiency and decrease the brake fuel conversion efficiency compared to conventional combustion.

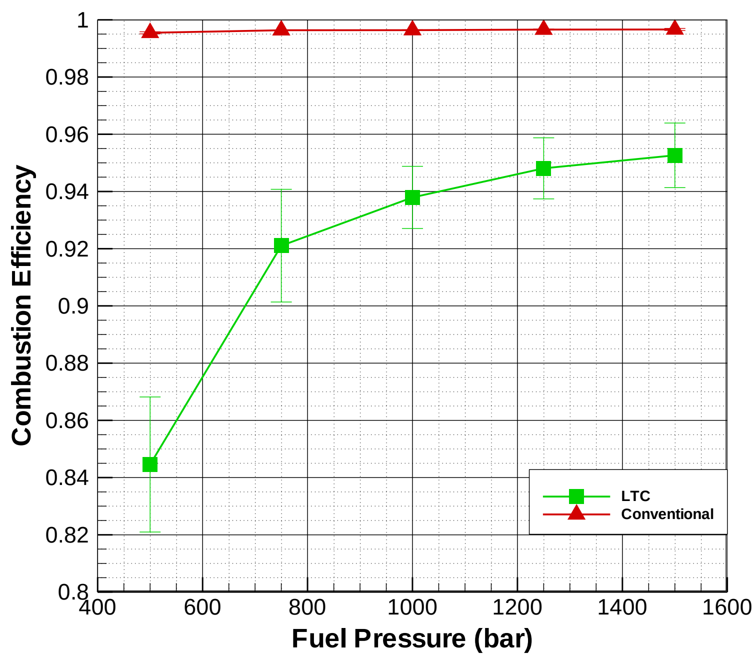

Figure 59. Combustion efficiency versus fuel pressure for LTC (■) and conventional (▲) combustion.

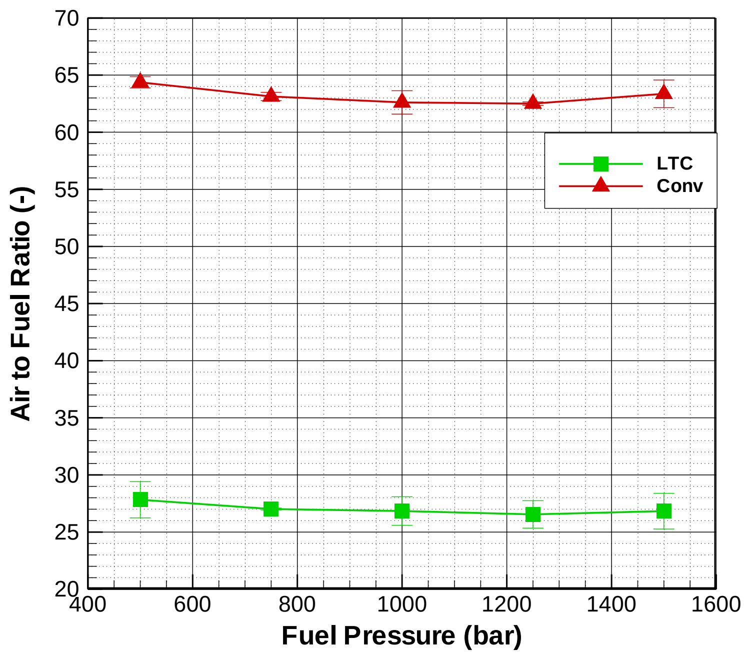

Figure 60. Air to fuel ratio versus fuel pressure for LTC (■) and conventional (▲) combustion.

As fuel pressure is increased for LTC, brake fuel conversion efficiency is seen to increase to a relative maximum in Figure 52. Since the mass flow rate of fuel is constant throughout the fuel pressure sweep, the engine power must be influencing the efficiency. Figure 58 details the engine torque versus the location of 50% mass fraction burned. As fuel pressure is increased, the 50% mass fraction burned location is advanced. The opposite trend holds true for conventional combustion. The increase in fuel pressure advances combustion and is detrimental to torque production because the combustion is phased too early. The effect of the increasing fuel pressure is to increase mixing of the fuel, decrease the mixing time, and advance the combustion timing. Because LTC and conventional combustion are not properly phased, the advancement in timing aids brake fuel conversion efficiency for LTC and reduces brake fuel conversion efficiency for conventional combustion.

Also important in determining the proper phasing of the combustion is the EMT, displayed in Figure 61. Conventional combustion is able to extract more energy out of the combustion mixture through the expansion stroke. As a result, the EMT for conventional combustion is generally lower than LTC, and conventional combustion should be able to utilize more of the extracted energy to produce more power, resulting in a higher brake fuel conversion efficiency.

Additionally for LTC, as EGR flows from the exhaust to the intake manifold, the amount of energy captured by the turbo is reduced. This necessarily relates to a lower turbine speed, Figure 62, and a lower boosted IMP, visible earlier in Figure 36.

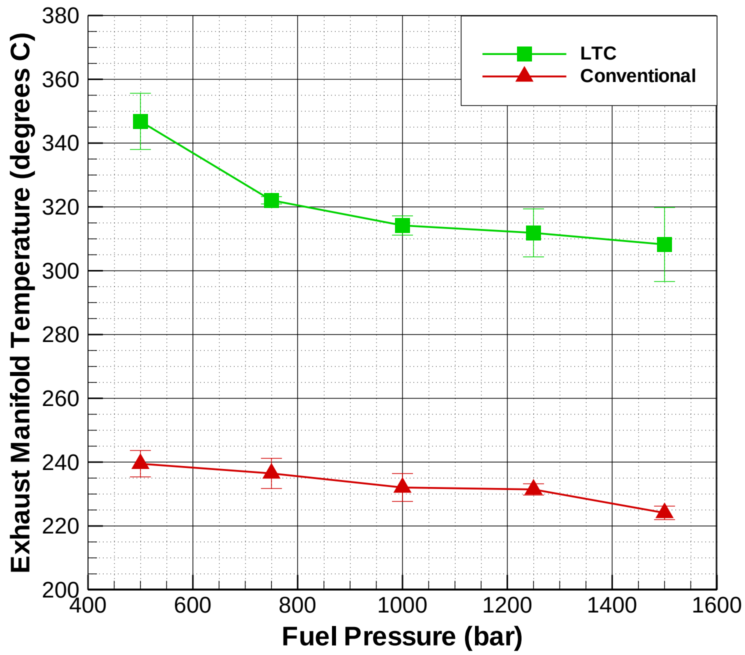

Figure 61. Exhaust manifold temperature versus fuel pressure for LTC (■) and conventional (▲) combustion.

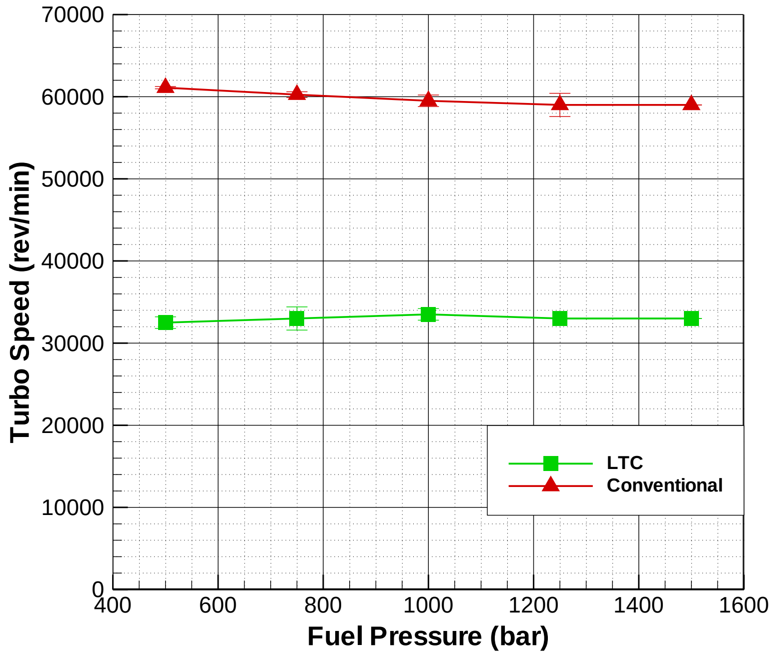

Figure 62. Turbo speed versus fuel pressure for LTC (■) and conventional (▲) combustion.

4. SUMMARY AND CONCLUSIONS

4.1 Summary

The use of diesel engines can satisfy the increased demand for an internal combustion engine with a lower carbon footprint and increased efficiency. However, the demand for lower NO_x and smoke emissions drives investigation into combustion regimes which limit the production of these emissions while simultaneously limiting penalties in fuel efficiency. Literature has shown that LTC is a suitable method for simultaneously reducing NO_x and soot production.

This thesis explores the emissions and the efficiency considerations of an experimental development of low temperature combustion in a medium-duty diesel engine. Discussion is performed on the effects that are caused by each swept parameter, and a resulting analysis is performed to investigate the reason why the effects are seen. Analysis focuses on the adiabatic flame temperatures of the in-cylinder combustion to characterize emissions production; however, some anomalies arise in the data, and care has been taken to examine the exact root cause.

Although literature exists to classify the effects of EGR and fuel pressure on LTC, no single work exists which uses literature to classify and compare these effects between LTC and conventional diesel combustion in such detail.

4.2 Conclusions

In conclusion, the objectives of this research study have been satisfied and the emissions and efficiency of an experimental application of low temperature combustion have been characterized with respect to variations in EGR and fuel pressure. More specifically, this study has contributed the following to the engineering community:

- The application of LTC is able to realize a 99% reduction in NO while simultaneously reducing smoke by 17% compared to the conventional combustion counterpart.
- Increasing EGR levels allow LTC to defeat the typical soot – NO tradeoff; however, brake fuel conversion efficiency decreases 6.8% for LTC, while conventional combustion realizes a 4% increase in efficiency.
- Increasing fuel pressure shows typical increases in NO and decreases in smoke for both LTC and conventional combustion; however, brake fuel conversion efficiency increases 2.3% for LTC and drops 4% for conventional combustion.

4.3 Recommendations for Continued Study

The approach to classifying the emissions and efficiency of an experimental development of low temperature combustion through sweeps of engine parameters allows opportunities for expansion of research. The following recommendations would allow for elaboration and enhancement of this research:

- Perform an extension of the EGR sweep using an expanded range with excessive amounts of EGR. Increases in the backpressure of the exhaust via throttling can allow larger rates of EGR than can be realized by the engine system alone. A sweep of this nature will allow the researcher to test the maximal limits of EGR until combustion becomes unstable for both LTC and conventional combustion. EGR rates greater than 56% will be in alignment with other literature and will give insight into advanced low temperature combustion [54].
- Examine the characteristics of each operating regime using a sweep of the engine VGT. In this study, the VGT was set using the baseline stock controller condition. However, adjustments in the VGT would allow changes in backpressure as well as changes in boosting and would give valuable insight into the effects of a VGT on LTC.
- Investigate optimal injection timings for both LTC and conventional combustion cases. Where the sweeps of engine parameters such as EGR would enhance brake

fuel conversion efficiency in conventional combustion in this study, a test with optimal injection timings would be indicative of the true impact of the parameter.

- Investigate higher engine loads and the effect of such on the different operating regimes. This would allow the researcher possible insight into the possible reasons for instabilities with higher torque production consistent with LTC.

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Select Publications

Bittle, J., Knight, B., Jacobs, T. (2010, May). Investigation into the use of ignition delay as an indicator of low temperature diesel combustion attainment. Accepted for publication in Combustion Science and Technology. GCST 2010-0021.

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