

Mixed-Source EGR for Enabling High Efficiency Clean Combustion Modes in a Light-Duty Diesel Engine

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ABSTRACT

The source of exhaust gas recirculation (EGR), and consequently composition and temperature, has a significant effect on advanced combustion modes including stability, efficiency, and emissions. The effects of high-pressure loop EGR (HPL EGR) and low-pressure loop EGR (LPL EGR) on achieving high efficiency clean combustion (HECC) modes in a light-duty diesel engine were characterized in this study. High dilution operation is complicated in real-world situations due to inadequate control of mixture temperature and the slow response of LPL EGR systems. Mixed-source EGR (combination of HPL EGR and LPL EGR) was investigated as a reasonable approach for controlling mixture temperature. The potential of mixed-source EGR has been explored using LPL EGR as a 'base' for dilution rather than as a sole source. HPL EGR provides the 'trim' for controlling mixture temperature and has the potential for enabling precise control of dilution targets. This approach also has a benefit where LPL EGR does not provide sufficient dilution for achieving conditions appropriate for HECC operation. The balance of the required dilution could be achieved with HPL EGR mitigating the need for throttling or a LPL EGR pump. The results of this investigation revealed significant differences in engine-out emissions and performance for various EGR sources.

INTRODUCTION

Advanced combustion concepts, such as homogeneous charge compression ignition (HCCI) and premixed charge compression ignition (PCCI), which are collectively often referred to as high efficiency clean combustion (HECC) modes, have the potential for achieving simultaneous reduction of nitrogen oxides (NO_x) and particulate matter (PM) emission levels while maintaining high diesel-like efficiency [1-11]. The HECC concept adopted in this study makes use of high levels of dilution and modified injection parameters to increase the ignition delay and in-cylinder mixing, respectively. The combined effect is an increase in the fraction of premixed fuel-air charge before the initiation of combustion. This approach is commonly referred to as PCCI.

EGR is an effective emission control technology for reducing NO_x emissions in diesel engines [12]. Specifically, increased EGR acts to reduce peak combustion temperatures, reducing the production of NO_x emissions. The reduction in combustion temperature is the result of heat absorption by EGR stream, primarily due to the heat absorption capacity of carbon dioxides (CO_2), leading to a reduction in combustion temperatures. Another contributing effect to lower combustion temperatures is dilution of the intake air with inert gases, leading to a decrease of oxygen (O_2) concentration in the combustion process.

This study uses two EGR configurations. The first configuration is a high pressure loop (HPL) commonly adopted in diesel engines. The HPL EGR method has a transfer line from upstream of the turbine to the downstream side of the compressor. The second configuration is a low pressure loop (LPL) [13]. The LPL EGR method has a transfer line from downstream of the turbine to the upstream of the compressor. This system is often employed in conjunction with the aftertreatment system. Rather than extracting EGR from upstream of the turbocharger, LPL EGR system use the exhaust gas that has been filtered through a diesel particulate filter (DPF) [14].

The HPL EGR system applied to modern diesel engines has enabled NO_x reduction. However, the penalty associated with the HPL EGR system was an increase in fuel consumption and in PM emissions. Turbocharger matching is usually revised to compensate for losing the effectiveness of turbocharger because a portion of the exhaust gas is intercepted and its energy rerouted away from the turbine in the HPL EGR system. The LPL EGR system has some advantages over the HPL EGR system. For example, the LPL EGR system has the potential for lower fuel consumption than that of HPL EGR as a result of improved turbocharger performance. Dilution from the LPL EGR system would also have higher heat rejection potentials as compared to the HPL EGR system since the exhaust gas down stream of the DPF is cooler than that provided from upstream of turbocharger. Due to the higher heat rejection potentials of the LPL EGR system, EGR cooling requirements could be reduced further. The cooler EGR stream

of the LPL system allows for further reductions in the intake mixture temperature resulting in improved volumetric efficiency and further reductions in NO_x emissions. Introducing the EGR upstream of the compressor also results in more uniform mixing of the dilution gases before induction into the engine.

The HPL EGR has typically been favored over the LPL EGR system in practice for several reasons. The presence of carbonaceous material in the recirculated exhaust stream may result in erosion of the compressor wheel. Although the recirculated exhaust stream is sourced downstream of a DPF, a fraction of carbonaceous material may still be present in the recirculated gases. Acidic condensation also becomes a problem as the temperature of the recirculated exhaust stream decreases. While acidic condensation is also a problem with the HPL EGR system, the problem is more significant with the LPL EGR due to increased cooling. The LPL EGR transfer system is also much longer than the HPL EGR transfer system resulting in a longer transient response time for the LPL EGR system.

The mixed-source EGR mode is used to control intake mixture temperature for studying the effects on HECC operation. The potential of mixed-source EGR is explored using LPL EGR as a ‘base’ for dilution and HPL EGR as the ‘trim’ for controlling mixture temperature and precise control of dilution levels.

In this study, three EGR dilution methods were investigated for achieving HECC operation. The first method used only the HPL for EGR and was representative of a production configuration. The second method used an LPL approach to EGR. The final approach used a combination of HPL and LPL EGR (or what is referred to as mixed-source EGR) for more precise control of intake mixture temperature. The ratio of HPL to LPL and the effect on emissions and efficiency was also investigated for HECC operation.

EXPERIMENTAL SETUP

The experiments were performed with a modified Mercedes 1.7-liter direct-injection, common rail diesel engine. A complete geometrical description of the engine is summarized in Table 1. The engine was operated with a dSpace-based flexible engine control system capable of actuating EGR valve, intake throttle, start of injection (SOI) timing, injection duration, fuel rail pressure, and number of injection events. The control system was set up to emulate the factory engine calibration and to allow departures from the factory calibration as needed to achieve HECC operation.

The HPL EGR system was modified by the addition of an EGR cooler that uses engine coolant as the working fluid. The production version of this engine does not have an EGR cooler. The EGR system does not have auxiliary temperature control such that the intake charge temperature (mixture of fresh air and EGR) increases with increases in the EGR mass flow rate. A catalyzed

diesel particulate filter (cDPF) is installed approximately 1 m downstream of the turbine exit. An LPL EGR system was added with recirculation of the exhaust gases from the exit of the cDPF through a heat exchanger to the inlet of the compressor. The LPL EGR cooler uses ambient temperature water as the cooling medium and was sufficiently effective that the EGR gas flow through the low pressure loop did not impact the EGR/ fresh air mixture temperature at the compressor inlet. The supplied air to the engine was conditioned to provide constant temperature and humidity levels at the engine air intake.

Table 1. Engine specifications

Bore [mm]	80
Stroke [mm]	84
Displacement [liter]	1.7
Number of Cylinder	4
Compression Ratio	19.5
Valve per Cylinder	4
Piston Bowl	Re-entrant
Fuel Injection System	Common Rail
Injector	Φ0.169 [mm] x 6

The exhaust backpressure was not controlled in this experimental set-up, but the cDPF was installed in the exhaust stream to provide realistic exhaust flow characteristics. The cDPF was a cylindrical cordierite monolith of 144 mm diameter and 152 mm length.

In-cylinder pressure measurements were recorded from each cylinder using Kistler piezoelectric pressure transducers. Typically 1,000 engine cycles were acquired for each condition. Net heat release values were calculated from cylinder pressure data by applying the first law of thermodynamics to the cylinder contents during combustion.

The fuel used in this study was a 2007 certification Ultra Low Sulfur (ULS) diesel fuel with cetane number of 43. Table 2 describes the properties of the fuel in detail.

Table 2. Fuel properties

C/H Ratio	1.84
Cetane Number	43
Specific Gravity	0.8424
Viscosity at 313 [K]	2.3 [cst]
Flash Point	346 [K]
Calorific Value	42.88 [MJ/kg]
Distillation: IBP	464 [K]
50 %	524 [K]
90 %	571 [K]
EP	611 [K]
Sulfur	9.0 [ppm]
Aromatics	30.3 [% vol.]

A standard heated chemiluminescence detector (CLD) and a flame ionization detector (FID) were used for measuring NO_x and unburned hydrocarbon (UHC) emissions, respectively. Carbon monoxide (CO) and CO₂ emissions were measured using non-dispersive

infrared (NDIR) instruments. Paramagnetic detector (PMD) was used to measure O₂ concentrations. The sample was chilled prior to measurement by PMD and NDIR instruments to achieve a non-condensing sample condition. The instruments sampled from the raw engine exhaust using a heated filter in order to remove particulates. The sample stream was conveyed from the heated filter to the instruments through a heated line maintained at 463 [K].

A Nicolet Nexus 670 Fourier Transform Infrared (FTIR) spectrometer was utilized to measure a number of gas phase species. The spectrometer was equipped with a heated gas cell with an optical path length of 10 m. the species of principal interest were acetylene (C₂H₂), ethene (C₂H₄), and propene (C₃H₆) because of the propensity for these species to participate in the formation of PM. The FTIR also allowed measurements of the other species of concern, such as methane (CH₄), formaldehyde (H₂CO), acetaldehyde (C₂H₄O), and nitrous oxide (N₂O). The FTIR sampled from the raw engine exhaust, using a heated filter for removal of particulates.

A smoke meter (AVL 415S) was used to measure soot emissions from the raw engine exhaust. The smoke meter utilizes an optical reflectometer to determine the amount of blackening of filter paper by the PM contained in a fixed volume of exhaust gas. It is not a direct measurement of particulate mass, but it is useful because it provides a rapid and generally accepted estimate of the PM emissions at a given steady-state engine condition. Since the measurement is an optical one, it may not accurately account for condensable organic in the PM, and hence may underestimate gravimetric PM measurements. Soot concentration (ρ) [mg/m³] is determined from the following equation:

$$\rho = \left(\frac{1}{0.405} \right) \times \alpha \times FSN \times \exp(\beta \times FSN)$$

The correlation is valid for filter smoke number (FSN) values up to and including 8. FSN is determined by a linear measurement of filter paper blackening using a gas column of 405 [mm] effective length. The values of the constants are $\alpha = 5.32$ and $\beta = 0.31$ [15].

RESULTS AND DISCUSSION

This paper focuses on operation of a light-duty diesel engine at two part-load conditions in conventional diesel combustion modes and HECC modes. The engine speed/load conditions of 1500 rpm, 2.7 bar BMEP and 2300 rpm, 4.3 bar BMEP were chosen as being representative of light-duty diesel operation. Conventional diesel operation engine parameters were based on a production ECU and HECC operation engine parameters deviated to include high levels of EGR and adjusted injection parameters. Adjusted injection parameters for HECC operation included higher fuel rail pressures for enhancing in-cylinder mixing and a single

injection event with proper phasing near TDC to avoid excessive pressure rise rates, which is constrained to $(dp/d\theta) \leq 10$ [bar/°CA] in this study. Fueling rate was maintained as constant for conventional and HECC operation.

CONVENTIONAL DIESEL COMBUSTION vs. HECC

Figure 1 shows a comparison of NO_x and PM emissions for conventional and HECC operation at 2300 rpm, 4.3 bar BMEP using HPL and LPL approaches for EGR, resulting in significant reduction of PM and NO_x emissions for both EGR systems. Although not shown, a similar trend was observed for the 1500 rpm, 2.7 bar BMEP condition. More detailed engine conditions and data for these comparisons are listed in Table 3.

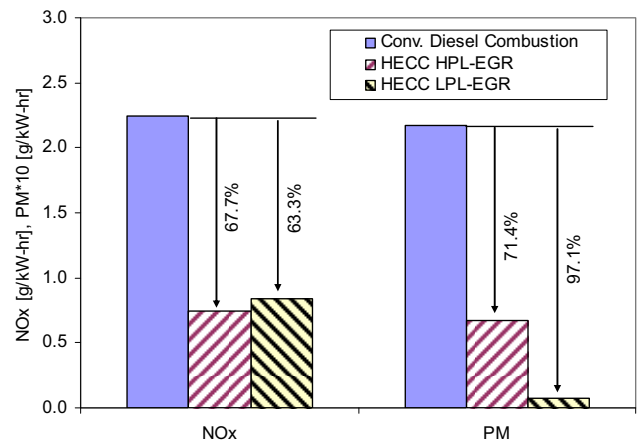


Figure 1. Comparison of NO_x and PM emissions at 2300 rpm, 4.3 bar BMEP condition

A review of Table 3 emphasizes the high dilution approach used to achieve HECC operation. An enabler of HECC operation is the longer ignition delay which results in a larger fraction of premixed combustion as compared to conventional combustion. To better understand the relationship of ignition delay to emissions and efficiency, a metric was defined as the time difference between the start of injection signal (SOI) and the start of combustion (SOC), which is defined for this study as the 5% mass fraction burned time (MFB5). As expected the metric indicated longer ignition delays for HECC operation as compared to conventional operation. This longer ignition delay corresponds to more time available for premixing of the fuel and air before combustion initiation. This increased fraction of premixed fuel/air charge resulted in more rapid burning during the early stages of the combustion process (MBF 5-50) as compared to conventional operation. The increased dilution for HECC operation also resulted in the later or non-premixed portion of the combustion process (MFB50-90) occurring slower than was observed for conventional combustion operation.