

Investigation of the Load Limits and Emissions of a Naturally-Aspirated Direct-Injection Diesel Engine

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ABSTRACT

Cost and robustness are key factors in the design of diesel engines for low power density applications. Although compression ignition engines can produce very high power density output with turbocharging, naturally-aspirated (NA) engines have advantages in terms of reduced cost and avoidance of system complexity. This work explores the use of direct injection (DI) and exhaust gas recirculation (EGR) in NA engines using experimental data from a single-cylinder research diesel engine. The engine was operated with a fixed atmospheric intake manifold pressure over a map of speed, air to fuel ratio, EGR, fuel injection pressure and injection timing. Conventional gaseous engine-out emissions were measured along with high speed cylinder pressure data to show the load limits and resulting emissions of the NA-DI engine studied. Well known reductions in NO_x with increasing levels of EGR were confirmed with a corresponding loss in peak power output. However, advancing fuel injection timing is shown to partially offset this disadvantage at the cost of higher engine noise. Partially premixed low temperature combustion (LTC) can be used over a portion of the operating range, further lowering the engine-out emissions of soot and NO_x . Since aftertreatment catalysts to reduce tailpipe NO_x and soot add cost and complexity to diesel engine-powered vehicles, the extensive use of EGR may have advantages for low power density NA diesel engines designed for low cost vehicle markets or for electrical power generation in range-extended electric vehicles.

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INTRODUCTION

Thirty years before this writing, naturally aspirated (NA) indirect injection (IDI) diesel engines occupied the majority of the light duty diesel market due to the inability of direct fuel injection systems to provide adequate fuel and air mixing at high engine speeds. These engines are well characterized by Alkidas et al. [1] and others. Direct injection (DI) engines were later introduced that were able to reach 10-15% higher thermal efficiencies due to reduced pumping losses and more advanced placement of the combustion event [2,3]. These NA-DI engines had penalties in higher NO_x emissions over their IDI counterparts due to more advanced combustion and increased hydrocarbon (HC) emissions from the presence of over-lean areas in the combustion chamber.

In the intervening years, turbocharging significantly reduced smoke emissions and combustion noise from diesel engines such that peak operating loads were more limited by

mechanical strength than by soot formation [4]. The introduction of common rail injection systems further reduced smoke emissions by allowing fuel injection pressure to be increased beyond 1200 bar. To reduce NO_x emissions from turbocharged DI engines, exhaust gas recirculation (EGR) was used at part engine load at the expense of higher soot formation. Multiple injection strategies were implemented to reduce increased combustion noise associated with DI engines though this also had higher smoke emissions as a penalty. As a result, state-of-the art DI diesel engines use turbocharging to enable high torque and power density. Advanced, fast-acting high-pressure injection systems and combustion chamber design balance the soot versus NO_x and combustion noise tradeoffs. Aftertreatment systems are then implemented to bring engine-out NO_x and soot, along with CO and HC emissions into legislative compliance.

For some diesel engine markets, the current trend of turbocharged DI using aftertreatment may not be cost



effective. A steady global production of NA-IDI and DI engines is expected in the future due to the desire for robust and inexpensive powertrain options for low power density applications like utility vehicles and goods carriers in developing countries. Although turbocharging allows diesel engines to reach higher power density output, NA engines meet the needs of low power applications at a lower cost.

Another area of application for lower cost NA engines may be as auxiliary power units (APU) for electrical power generation onboard a vehicle. The use of turbocharged DI diesel engines in hybrid powertrain applications is generally avoided due to their higher cost than spark-ignited gasoline engines. However, for range-extended electric vehicles, the APU need only operate within a small window of operation to charge the primary battery [5, 6]. It is conceivable that NA-DI diesel engines could match the cost of gasoline engines at comparable power output but achieve higher fuel conversion efficiency due to their natural advantages from higher compression ratio, lower pumping work and lean operation.

Although low cost NA engines may fit well in niche markets, some doubt exists as to whether they can meet increasingly stringent emissions standards. To address this need, higher fuel injection pressure, made possible by modern DI injection systems, can reduce soot emissions from NA engines such that NO_x can be managed by retarded injection timing at higher engine loads [7]. NO_x aftertreatment from diesel engines using lean NO_x traps (LNT) or selective catalytic reduction (SCR) catalysts are more expensive than soot cleanup using diesel particulate filters (DPF). Therefore, engine management of NO_x at the expense of soot is generally thought to be the optimum strategy for low cost engines to meet Euro IV-comparable emissions standards in developing countries [8].

To help reduce engine-out NO_x , EGR is increasingly being considered for light-duty NA diesel engines [9,10,11]. A difficulty exists in implementing EGR in NA engines due to the lack of backpressure to drive flow from the exhaust manifold back into the intake. However, flow can be induced by creating higher backpressure in the exhaust with a restriction [12] or by reducing the intake manifold pressure with a throttle [13, 14], though both of these reduce brake thermal efficiency to some degree. Published data are scarce to evaluate whether NA engines using a modern DI injection system can achieve desired load capacity with suitably low emissions for use in low power density applications.

With the introduction of high pressure injection systems and EGR, it is also possible that partially premixed low temperature combustion (LTC) can be used in NA engines to further reduce soot and NO_x emissions. In this regime, EGR is used to both extend ignition delay by reducing the mixture oxygen concentration and lower combustion temperatures by raising the specific heat of the trapped mass. High fuel injection pressure is also used to increase mixing and extend ignition delay. The resulting combustion occurs in a primarily

premixed fashion and has been shown to simultaneously reduce soot and NO_x emissions to very low levels. Premixed LTC has been extensively researched and is reviewed by Dec [15] and others. Most work in LTC has used low intake manifold pressures since the avoidance of regions of NO_x formation are largely driven by avoiding high peak cylinder temperature. Therefore, LTC could be used over a portion of the operable range in a NA-DI diesel engine, possibly reducing the requirements for expensive soot or NO_x aftertreatment systems.

This work explores the load limits and emissions of NA-DI engines using experimental data from a single cylinder DI research diesel engine run with a fixed 100 kPa intake pressure. The study was conducted over ranges of engine speed, load, EGR, fuel injection pressure and injection timing. Using the experimental data, opportunities for implementation of LTC operation for NA-DI engines is discussed along with the feasibility of NA-DI engines to meet market needs in low power density applications.

EXPERIMENTAL

To examine the load limits and emissions of NA-DI engines, an experimental investigation was performed using a single cylinder research diesel engine with specifications shown in Table 1 [16]. The engine employed a common rail fuel injection system with a modern piezo fuel injector. A symmetrical re-entrant bowl (ω type) was used as seen in the diagram of the combustion system shown in Figure 1 [17]. The engine had controllable intake and exhaust pressure as well as all injection parameters like timing and rail pressure. EGR was induced using a higher exhaust backpressure than intake and precisely controlled using a valve. Engine speed and load were controlled by a DC motoring dynamometer.

Table 1. Specifications of engine used in the experimental study

Displaced volume	0.825 l
Stroke	99 mm
Bore	103 mm
Compression ratio	16.0
Number of Valves	4
Injection Type	Common Rail – Piezo
Nozzle holes Number	7
Nozzle hole diameter	0.1539 mm

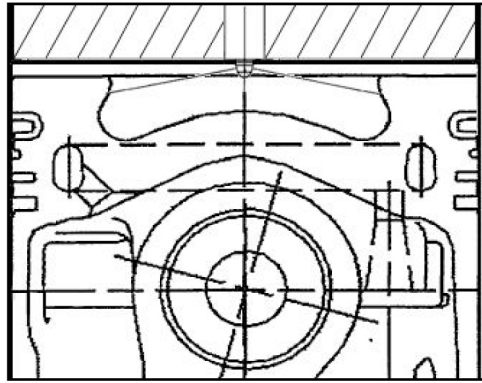


Figure 1. Cutaway diagram of the combustion system

Gaseous emissions were measured using a Horiba MEXA-2 emissions bench which used a chemiluminescence analyzer for NO_x, non-dispersive infrared analyzers for CO and CO₂, a paramagnetic analyzer for O₂ and a flame ionization detector (FID) for total HC. Soot emissions were measured using an AVL 415S filter smoke meter from which three samples were averaged for every data point recorded. High speed cylinder pressure data were recorded with 0.2CA° resolution using an A&D - CAS combustion analysis systems and Kistler 6125C piezo-electric pressure transducers. The fuel injector electrical signal was also measured on a high speed basis.

Emissions are reported here on a mass of species per mass of fuel consumed basis, known as emissions index (EI), and were calculated using the methods described in Ref. [18]. The EI basis is useful for comparison of emissions excluding the effects of engine brake specific fuel consumption which is not an accurate measure of engine performance when using single-cylinder research engines. Using wet concentrations, the emissions index for a given species *j*, is given by eq. 1.

$$EI_j = X_j \left[\frac{MW_j}{MW_f [X_{CO} + X_{CO_2} + 3X_{HC}]} \right] \quad (1)$$

In eq. 1, EI_{*j*} is the emissions index of species *j* in g/kg-fuel, X_{*j*} is the wet mole fraction of species *j*, MW_{*j*} is the molecular weight of species *j* and MW_{*f*} is the molecular weight of fuel per carbon atom. EGR was calculated based on the difference between intake and exhaust CO₂ measurements using the methods described in [18].

Heat release analysis was performed with the high speed crank angle and cylinder pressure data using a code internally developed by General Motors Research and Development. The rate of pressure rise was calculated individually for 100 cycles per data point and filtered using a Butterworth low pass filter [19] with a 1500 Hz cutoff frequency. The heat release profile was then calculated on an individual cycle basis using the procedure outlined in Heywood [4] and the result was averaged. Net apparent cumulative and rate of heat

release (RoHR) are reported here where heat transfer losses from the cylinder were not estimated. This procedure is deemed sufficient for qualitatively comparing the combustion processes between operating conditions. Combustion phasing was estimated using the location of 50% net cumulative heat release. Engine noise was measured using the cylinder pressure signal and an AVL 4050-A combustion noise meter.

Table 2 shows the range of engine operating conditions run in the experimental study. All variations in load for a given engine speed were done by changing fuel injection duration while adjusting injection timing to maintain constant combustion phasing of 5 CA°ATDC. For variation in injection timing and injection pressure, fueling rate was held constant and timing was adjusted to maintain constant combustion phasing. Unless otherwise noted in the paper, the injection rail pressure was 800 bar. Other parameters like intake and exhaust pressure, as shown in Table 2, were also held constant throughout the study.

Table 2. Engine experimental test conditions

Engine Speed	1000 – 2500 rpm
Air to Fuel Ratio	24-78
Injection Rail Pressure	600 – 1000 bar
Exhaust Gas Recirculation	0 – 60%
Injection Timing	-30 – -1 CA°ATDC
Combustion phasing @ 50% burned	5° ATDC
Intake Temperature	35°C
Absolute Intake Pressure	100 kPa
Intake Relative Humidity	0%
Absolute Exhaust Pressure	130 kPa
Coolant Temperature	80°C
Oil Temperature	80°C

RESULTS

ENGINE SPEED

Data were taken at different engine speeds over a range of engine load. Other operating parameters shown in Table 2 were held constant. The engine performance results of these experiments are shown in Figure 2. Net indicated parameters, calculated over 720 CA° including the pumping portion of the cycle, are used here and throughout the paper. Brake values are not realistic measures of single cylinder research engine performance since friction work is inherently higher than in production engines. In Figure 2, net indicated power (P_{i,n}), net specific fuel consumption (NSFC), and exhaust temperature are plotted versus engine load shown as net indicated mean effective pressure (NMEP).

Net indicated power (P_{i,n}) is shown to highlight the power output a NA engine could achieve for the engine displacement used in the experiments. The maximum P_{i,n} of the engine was 12.6 kW at an engine speed of 2500 rpm. P_{i,n} increased linearly with NMEP and monotonically with engine

speed. The net specific fuel consumption (NSFC) decreased with increasing load in part due to work output increasing faster than heat loss from the cylinder. The curves show that minima in NSFC exist toward the upper loads for each engine speed with the highest speed minimum occurring at a lower load (~600 kPa) than for the lower speed cases. As speed increases, the load of minimum NSFC decreases since less time is allowed for heat transfer to occur during the high temperature combustion period. The lowest engine speed shows higher NSFC over the load range due to longer residence time in the cylinder during combustion allowing greater heat loss to occur.

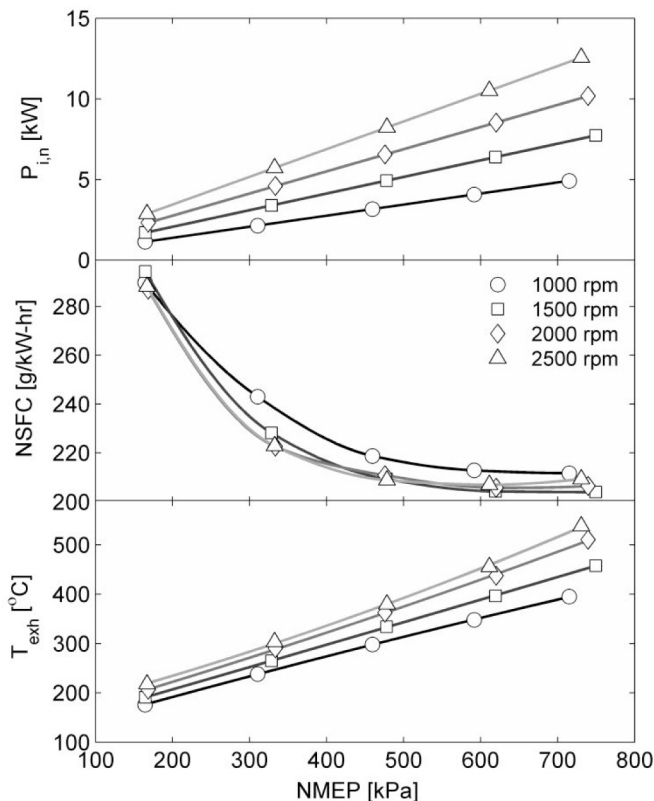


Figure 2. $P_{i,w}$, NSFC and T_{exh} versus NMEP over range of engine speed

Exhaust temperatures (T_{exh}) increased linearly with NMEP and monotonically with engine speed. T_{exh} data also is presented in Figure 2 to draw attention to the applicability of aftertreatment devices for use with NA engines. Since turbo-machinery would not be used in a low cost NA engine, the enthalpy of the exhaust stream is higher, leading to the possibility of lighter loading of precious metal catalysts.

Lower NSFC was also attained at high engine load due to more optimal placement of the high temperature heat release event to maintain a CA50 of 5°ATDC. This can be seen in Figure 3 where the net apparent RoHR and cumulative heat release are plotted versus crank angle. As NMEP increased at a constant speed of 1500 rpm, the premixed heat release event became longer in duration and more advanced while

the CA50 point was held constant. Higher thermodynamic cycle efficiency is obtained as the high temperature heat release event is nearer to TDC since the bulk cylinder gas reaches a higher availability state near TDC. However, as load is further increased, the mixing-controlled portion of the heat release was extended into the expansion stroke reducing the available work that can be done and reducing thermal efficiency due to later burning.

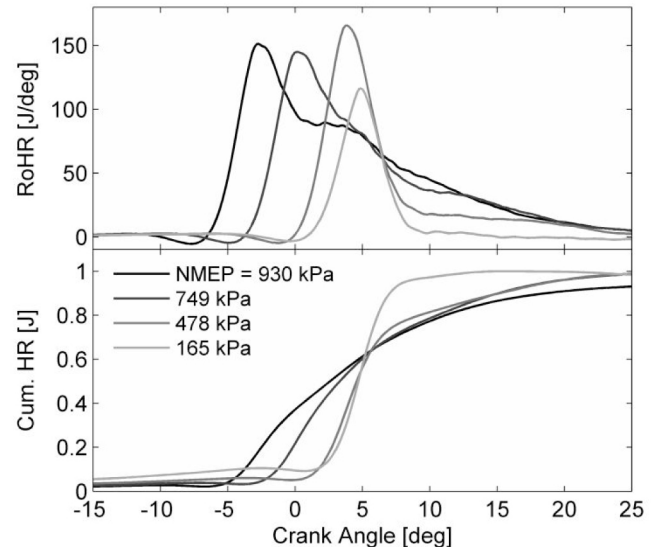


Figure 3. Net Apparent RoHR and normalized, cumulative heat release for 1500 rpm over range of engine load at constant CA50 = 5°ATDC

The emissions of NO_x , smoke, hydrocarbons (HC) and CO as a function of NMEP are shown in Figure 4 over the tested range of engine speed. The emissions index (EI) of NO_x is higher for decreasing engine speed and is essentially constant over the range of load for each speed. The filter smoke number (FSN) increases with load, having a more dramatic rise at high engine speeds. Both NO_x and smoke formation are mainly controlled by in-cylinder temperature, residence time during the cycle and local oxygen availability. At lower engine speeds, longer time is available for NO_x chemistry to proceed. At high engine speeds and high NMEP, soot formed during combustion due to locally rich mixtures has insufficient time to find oxygen and burn out completely.

HC and CO emissions both increase for leaner A/F (lower loads) due to more premixed combustion creating areas of the cylinder where complete oxidation cannot be achieved. Since HC and CO emissions at low loads are mainly controlled by mixing preparation and not the duration of combustion, it is possible to see low sensitivity to engine speed except for the lowest speed, where longer residence time may allow time for additional oxidation of HC. It is necessary to highlight that this sensitivity is not only present at low load but also in the entire load sweep. At high load and speed, the time allowed for combustion decreases and CO emissions increase, arising from insufficient mixing between rich regions where CO is