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Improvement in high load ethanol-diesel dual-fuel combustion by Miller cycle and charge air cooling

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HIGHLIGHTS

- The maximum ethanol energy fraction was increased from 0.26 to 0.79.
- Miller cycle effectively delayed the ethanol autoignition process without EGR.
- A lower intake air temperature also retarded the early ignition of the ethanol fuel.
- Dual-fuel combustion yielded a better trade-off between efficiency and NOx emissions.
- NOx emissions were reduced by 44% while increasing net indicated efficiency by 4.1%.

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ABSTRACT

At high load, dual-fuel compression ignition engines often rely on exhaust gas recirculation (EGR) to avoid excessive in-cylinder pressure rise rates caused by the autoignition of the premixed fuel. This can adversely affect the net indicated efficiency depending on the resulting fuel/air equivalence ratio and pressure differential across the cylinder used to drive the requested amounts of EGR. In this work, advanced combustion control strategies have been experimentally investigated to improve ethanol-diesel dual-fuel operation at a high engine load of 1.8 MPa net indicated mean effective pressure. Miller cycle and charge air cooling have been explored to reduce the in-cylinder gas temperature and help control the ethanol autoignition process, potentially minimising the EGR requirements and increasing net indicated efficiency. Experiments were carried out on a single-cylinder heavy-duty engine equipped with a high pressure common rail diesel injection, an ethanol port fuel injection, and a variable valve actuation system on the intake camshaft. Exhaust emissions and net indicated efficiency were measured and discussed for different ethanol energy fractions. Early autoignition of the premixed ethanol fuel at the baseline intake valve lift profile resulted in high levels of pressure rise rate, which limited the ethanol energy fraction to 0.26. The application of a Miller cycle strategy via late intake valve closing events effectively delayed the ethanol autoignition process. The reduction of the intake manifold air temperature via an air-to-water charge air cooler also suppressed the early ignition of ethanol. Both approaches allowed for a substantial improvement in terms of the maximum ethanol energy fraction, which was increased to 0.79 without EGR. Moreover, high load dual-fuel operations with Miller cycle and charge air cooling attained higher net indicated efficiency and lower nitrogen oxides emissions than conventional diesel combustion. These improvements can help generate a viable business case of dual-fuel combustion as a technology for future compression ignition engines.

1. Introduction

The vast majority of the transportation sector's energy needs is met by oil [1]. Increased global energy demand and greenhouse gas (GHG) emissions regulations are driving the use of renewable energy sources and advances in powertrain technology. The introduction of biofuels to highly efficient internal combustion engines can help reduce the transport sector's GHG emissions and petroleum dependence [2].

Biofuels are gaseous or liquid fuels produced from biomass, which is the biodegradable fraction of municipal and industrial waste as well as products, waste, and residues from agriculture, forestry, and related industries [3]. The analysis of the GHG emissions associated with the production, transport, and distribution of different biofuels [4,5] revealed that ethanol produced from sugar cane results in lower overall GHG emissions than the life cycle GHG intensity of fossil fuels [6].

Dual-fuel combustion has been proven as an effective means of

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Nomenclature

ATDC	After Firing Top Dead Centre
BMEP	Break Mean Effective Pressure
CA10	Crank Angle of 10% Cumulative Heat Release
CA10-CA50	10–50% Cumulative Heat Release
CA10-CA90	Combustion Duration or 10–90% Cumulative Heat Release
CA50	Crank Angle of 50% Cumulative Heat Release
CA90	Crank Angle of 90% Cumulative Heat Release
CAD	Crank Angle Degrees
CDC	Conventional Diesel Combustion
CO	Carbon Monoxide
CO_2	Carbon Dioxide
COV_IMEP	Coefficient of Variation of IMEP
DAQ	Data Acquisition
DF	Dual-Fuel
E85	Gasoline with 85% Ethanol in a Volume Basis
ECR	Effective Compression Ratio
ECU	Engine Control Unit
EF	Ethanol Energy Fraction
EGR	Exhaust Gas Recirculation
EIVC	Early Intake Valve Closing
FID	Flame Ionisation Detector
GCR	Geometric Compression Ratio
GHG	Greenhouse Gas
HC	Hydrocarbons
HRR	Apparent Net Heat Release Rate
IAT	Intake Manifold Air Temperature
IMEP	Net Indicated Mean Effective Pressure
ISCO	Net Indicated Specific Emissions of CO
ISHC	Net Indicated Specific Emissions of Actual Unburnt HC
ISNOx	Net Indicated Specific Emissions of NOx
ISsoot	Net Indicated Specific Emissions of Soot
IVC	Intake Valve Closing
IVO	Intake Valve Opening
LHV_{CO}	Lower Heating Value of Carbon Monoxide
LHV_{DF}	Actual Lower Heating Value in Dual-Fuel Mode
LHV_{diesel}	Lower Heating Value of Diesel
$LHV_{ethanol}$	Lower Heating Value of Ethanol
LIVC	Late Intake Valve Closing
$\dot{m}_{diesel}$	Mass Flow Rate of Diesel
$\dot{m}_{ethanol}$	Mass Flow Rate of Ethanol
MFB	Mass Fraction Burnt
NG	Natural Gas
NOx	Nitrogen Oxides
O_2	Oxygen
P_{ind}	Net Indicated Power
PFI	Port Fuel Injector
Pmax	Maximum In-cylinder Gas Pressure
PMEP	Pumping Mean Effective Pressure
PRR	Pressure Rise Rate
SOC	Start of Combustion
SOI	Actual Start of Diesel Injection
SOI_main	Actual Start of Main Diesel Injection
TDC	Firing Top Dead Centre
VVA	Variable Valve Actuation
γ	Ratio of Specific Heats
Φ	Global Fuel/Air Equivalence Ratio

utilising alternative fuels in conventional diesel engines [7]. This strategy can be achieved via the installation of a low cost port fuel injection system for the formation of a low reactivity mixture of air and fuel, such as natural gas (NG), gasoline, or ethanol [8]. The stock diesel combustion and fuel injection systems can be retained in the dual-fuel engine. Direct injected diesel fuel serves as the ignition source for the premixed charge [9]. The use of different fuels as well as variations in the diesel injection timing and energy fraction of each fuel can change the dual-fuel combustion characteristics, emissions, and efficiencies.

Optimised dual-fuel combustion can attain lower nitrogen oxides (NOx) and soot emissions than conventional diesel combustion (CDC) [10,11]. Improvements in efficiency are also achievable [12–14]. However, relatively high levels of carbon monoxide (CO) and unburnt hydrocarbon (HC) emissions are usually reported at low engine loads [15]. Furthermore, dual-fuel operation at high load conditions have been proved extremely challenging due to peak in-cylinder pressure [16] and/or pressure rise rate limitations [17,18]. This is associated with the autoignition of the low reactivity fuel.

A number of studies have investigated combustion control strategies to allow for safe high load dual-fuel operation, such as the direct dual fuel stratification [19–21] and the use of high EGR rates [22–25]. However, these approaches often require relatively complex and expensive engine hardware modifications and/or high levels of boost pressure in order to supply enough fresh air for lean operation. Therefore, experimental research has been mostly focused on the use of a lower compression ratio to decrease the in-cylinder gas pressure and temperature during the compression stroke. This delays the ignition of the fuel and allows for longer fuel–air mixing process [26].

A reduction in compression ratio is typically attained via a modified piston [27]. High load gasoline-diesel dual-fuel combustion has been achieved on a medium-duty engine using a piston with a lower geometric compression ratio (GCR) of 12.75 [28]. Despite the improvement, it is likely challenging to simultaneously attain high levels of boost pressure and EGR rate at an intake charge temperature of 293 K in a production engine. Additionally, experiments and computational optimisations performed on a heavy-duty engine with a GCR of 12 showed that controlling the dual-fuel combustion process at high loads can be quite demanding due to the sensitivity to fluctuations in the EGR rate [29]. Furthermore, the use of a low GCR piston can lead to less efficient dual-fuel combustion at light engine loads [28].

Alternatively, Miller cycle can be used to lower the effective compression ratio (ECR) via an early or late intake valve closing event [30–32]. The strategy delays the actual initiation of the compression process, reducing the charge density and decreasing both the in-cylinder gas temperature and pressure prior to the start of combustion [33]. The approach also allows for a more flexible combustion control when the stock piston and original compression ratio are retained and the intake valve timings are varied according to engine operating condition. However, the reduction in charge density observed with Miller cycle can increase the mean in-cylinder gas temperature during combustion, as the heat is released into a lower in-cylinder mass trapped. Ickes et al. [34] have shown that this can potentially increase heat transfer losses, as supported by lower gross net indicated efficiency for the cases with lower ECRs.

Previous Miller cycle research with an early intake valve closing (EIVC) showed gasoline-diesel dual-fuel combustion can be used over the entire engine speed-load map while maintaining NOx emissions below 0.4 g/kWh [35]. The maximum engine load was increased from 1.2 MPa to 2.2 MPa break mean effective pressure (BMEP) when the ECR was reduced from 14.4 to 11. However, the study also relied on the use of high levels of EGR, which can decrease the actual engine efficiency depending on the resulting fuel/air equivalence ratio and pressure differential across the cylinder (e.g. pumping losses) used to drive the requested amounts of EGR [23].

The introduction of premixed fuels with high knock resistance such as ethanol and NG potentially allows for the use of relatively higher