Fuel 208 (2017) 722–733

Contents lists available at ScienceDirect

Fuel

journal homepage: www.elsevier.com/locate/fuel

Full Length Article

Emissions and performance of diesel–natural gas dual-fuel engine operated with stoichiometric mixture

Jiří Vávra *, Ivan Bortel, Michal Takáts, Marcel Diviš

Czech Technical University in Prague, Faculty of Mechanical Engineering, Prague, Czech Republic

ARTICLE INFO

Article history:
Received 28 February 2017
Received in revised form 15 June 2017
Accepted 13 July 2017
Available online 24 July 2017

Keywords:
Dual-fuel combustion
Diesel engine
Stoichiometric operation
Compression ratio

ABSTRACT

Carbon dioxide emissions, particulate matter and nitrogen oxides are the most important exhaust gas emission components from the compression ignition internal combustion engines. The diesel cycle shows a superior thermal efficiency compared to the spark ignition (SI) combustion.

Unlike SI engines a complete (100%) replacement of liquid fuel with gaseous fuel, containing less carbon, such as natural gas, is not possible for compression ignition (CI) engines. Since resistance of methane to auto-ignition is high, it is necessary to use another fuel with higher reactivity for ignition. Therefore a partial substitution of diesel fuel with low-carbon fuel seems to be a meaningful approach to a step reduction of CO_2 emissions.

One option is a dual-fuel operation of the diesel engine. As of today, the majority of dual-fuel (DF) engines research and development activities are focused on large cylinder bore and heavy duty engines operated with lean mixtures, several previous works describe the evaluation of DF principle in the light duty engines operating in lean and stoichiometric region.

The goal of this work is to identify the potentially interesting regions for practical use of the DF combustion in a broad area of loads, NG substitution ratios, boost level and speed, on a diesel engine of the cylinder size and structure compatible with engines for passenger cars and vans.

The work was performed on a single cylinder engine and the tests were focused on engine performance, efficiency, gaseous emissions and particulate matter. The fouling of injector was identified and quantified on an injector test bed.

1. Introduction

Compression ignition (CI) engines are used in a wide range of applications, including large marine and stationary power-generating engines, and from heavy-duty trucks to small passenger car engines [1]. An advantage of CI engines, in comparison with a spark ignition (SI), is its higher overall fuel conversion efficiency [1,2] and lower CO_2 emissions. The main reason is higher compression ratio compared to the SI engine, unthrottled and lean operation, especially at part load [1].

Emissions of carbon dioxide (CO_2), particulate matter (*PM*) and nitrogen oxides (NO_x) are the most important exhaust gas emission components from the compression ignition internal combustion engines [3].

Since carbon dioxide is considered a greenhouse gas, there are multiple reasons why to reduce its emission [4]. Carbon dioxide emissions can be reduced via higher fuel conversion efficiency or by using a fuel with lower carbon content [4–7].

Spark ignition (SI) engines allow the use of the low-carbon fuel, as natural gas, for a significant reduction of carbon dioxide emissions. However, the complete substitution of the diesel fuel with natural gas (NG) is impossible for compression ignition (CI) engines. Typical solution is a conversion of the CI engine to the SI one. A partial substitution of diesel fuel with the fuel containing less carbon seems to be a meaningful approach to a step reduction of CO_2 emissions, and an attempt to keep fuel conversion efficiency close to CI engines [8–10]. This approach (initiation of combustion of air–NG mixture by a pilot diesel fuel injection) is called dual-fuel (DF) CI combustion.

A dominant source of PM emissions in diesel engines is a high temperature pyrolysis of heavy hydrocarbons in diesel fuel, which is burnt primarily by a diffusion flame [4]. Therefore, minimizing diesel fuel injection quantity in DF CI engine leads to particulate emission reduction. This was observed via PM, opacity or smoke measurement in [2,9,11–14]. This effect is more evident at high load and suppressed at low load [2,8]. Couple works describe

* Corresponding author.
E-mail address: Jiri.Vavra@fs.cvut.cz (J. Vávra).
URL: http://bozek.cvut.cz (J. Vávra).

http://dx.doi.org/10.1016/j.fuel.2017.07.057
0016-2361/

Nomenclature

η_i	indicated efficiency, [1]
λ	air excess ratio, [1]
σ	mass fraction, [1]
*AI*05	position of 5% mass fraction burned, [°*CA*]
aTDC	after top dead center, [°*CA*]
bTDC	before top dead center, [°*CA*]
C_2H_6	ethane
C_3H_8	propane
CA	cranck angle, [°]
*CA*50	position of 50% mass fraction burned, [°*CA*]
CAD	crank angle in degree, [°*CA*]
ccm	cubic centimeter
CH_4	methane
CNG	compressed natural gas
CO	carbon monoxide
CO_2	carbon dioxide
COV	coefficient of variability, [%]
CR	compression ratio or common rail
DF	dual fuel; combustion of air-NG mixture initiated by a pilot diesel fuel injection
diesel	diesel fuel or diesel fuel operating mode of engine
*dQ*1	rate of heat release, [kJ/°]
ECU	engine control unit
EGR	exhaust gas recirculation
EI CH_4	emission index of methane, $[g/kg_{fuel}]$
EVO/*EVC*	exhaust valve opening/closing, [°*aTDC*]
exh	exhaust
FID	flame ionization detector
FSN	filter smoke number
GWI	global warning index
GWP	global warning potential
HC	hydrocarbon
HCCI	homogeneous charge compression ignition
IMEP	indicated mean effective pressure, [bar]
*INJ*1	electric current on injector, $[A]$
int	intake
IS X	indicated specific emmission of X, [g/kWh]
IVO/*IVC*	intake valve opening/closing, [°*aTDC*]
LNT	lean NO_x trap
m	mass, [kg]
main	main injection
N_2	nitrogen
NA	naturally aspirated
NDIR	non-dispersive infrared
NG	natural gas
NO_x	nitrogen oxides
p	pressure, $[Pa, bar]$
p_{cyl}	in cylinder pressure, [bar]
P_{max}	maximal pressure during the engine cycle, [bar]
*PCYL*1	in cylinder pressure, [bar]
pilot	pilot injection
PM	particulate matter, [g/kWh]
PN	particle number, $[\#/kWh, \#/ccm]$
PPCI	partially pre-mixed charge compression ignition
Q_{Diesel}	Diesel injection quantity, $[mg\ per\ cycle]$
R_{max}	maximal rate of pressure rise during the engine cycle, [bar/°]
RCCI	reactivity controlled compression ignition
rpm	revolution per minutes
SCR	selective catalytic reduction
SI	spark ignition
SOI	start of injection
T	temperature, [K, °]
t	time, [s]
TDC	top dead center
THC	total hydrocarbon emission, [g/kWh]
THC_{C1}	total hydrocarbon emission calibrated by methane, [g/kWh, ppm]
TWC	three way catalytic converter

measurement of particle number (PN) [2,10,15]. According to these articles, the PN in raw exhaust gases was in the range from 5×10^5 to 8×10^7 [#/ccm], and trends of PN are approximately corresponding with the gravimetric PM measurements. Results of particle size distribution measurement are mentioned in [2,15], and dual-fuel PM physical structures and size distribution were described in [12].

The DF nitrogen oxides (NO_x) production is associated primarily with the pilot diffusion combustion zone, as a result of high local temperatures and possible longer reaction times [4,16]. Consequently, the injection strategy and pilot injection quantity significantly determine the final exhaust NO_x emissions during DF modes [16]. Königsson in [3] affirmed, that NO_x from the diesel fuel pilot dominates at lean conditions above λ value threshold of approx. 1.6. Below this threshold, NO_x from the methane combustion dominated. In the literature [2,8,9] is mentioned reduction in NO_x for DF engines in comparison with CI engine with the exception of indirect diesel fuel injection CI engine [17]. In this case, DF NO_x emission decreased at higher load and were comparable with CI diesel engine operated at idle. Potential of combining electric hybridization with a DF natural gas-diesel engine has been presented in [18], where researchers attempted to comply with the passenger car EURO 6 limits for NO_x without LNT (lean NO_x trap) or SCR (Selective Catalytic Reduction) catalyst. The last mentioned article did not provide any information on hydrocarbon (*HC*) emission.

However, the emissions of HC are a weak spot of the DF CI engine since methane is insufficiently reactive and it is particularly difficult to suppress its emissions by the conventional aftertreatment system [3]. Significant deterioration of HC emissions with respect to pure diesel CI engines is presented in most articles [2,8,9,11,12,14], mainly at low load [8,11,19]. An original diesel engine combustion chamber with crevices (and consequently low temperature regions) can be partially responsible for above mentioned phenomena [3,16].

Nonuniformity of λ over combustion chamber resulted in rich regions, where lack of oxygen leads to a partial oxidation, and lean regions on the other hand, where the low chemical reactivity leads to partial oxidation or quenching [16]. Emission of HC can be partially reduced by preheating of intake air. Global Warming Potential (GWP) of methane is 25 times higher in comparison with the same mass of CO_2 [20], which makes the control of methane emission more important.

Carbon monoxide (CO) is formed under conditions close to those of HC formation. The fuel-rich region, as the origin of CO, is formed by the introduction of the pilot fuel. However, most of it is oxidized during the ongoing combustion process. Dominant source of *CO* emissions at low engine speed is the squish volume close to the cylinder walls. A lean mixture and the limited residence time for burnup at high engine speed are the likely source of CO [16].