Energy 94 (2016) 728–741

Contents lists available at ScienceDirect

Energy

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

Experimental and theoretical analysis of the combustion process at low loads of a diesel natural gas dual-fuel engine

Weifeng Li, Zhongchang Liu, Zhongshu Wang*

State Key Laboratory of Automotive Simulation and Control, Jilin University, Changchun 130025, China

ARTICLE INFO

Article history:
Received 8 August 2015
Received in revised form
9 November 2015
Accepted 23 November 2015
Available online 18 December 2015

Keywords:
Dual-fuel engine
Natural gas
Diesel
Combustion
Low loads

ABSTRACT

To construct an effective method to analyze the combustion process of dual fuel engines at low loads, effects of combustion boundaries on the combustion process of an electronically controlled diesel natural gas dual-fuel engine at low loads were investigated. Three typical combustion modes, including *h*, *m* and *n*, appeared under different combustion boundaries. In addition, the time-sequenced characteristic and the heat release rate-imbalanced characteristic were found in the dual fuel engine combustion process. To quantify these characteristics, two quantitative indicators, including the TSC (time-sequenced coefficient) and the HBC (HRR-balanced coefficient) were defined. The results show that increasing TSC and HBC can decrease HC (hydrocarbon) emissions and improve the BTE (brake thermal efficiency) significantly. The engine with the *n* combustion mode can obtain the highest BTE and the lowest HC emissions, followed by *m*, and then *h*. However, the combustion process of the engine will deteriorate sharply if boundary conditions are not strictly controlled in the *n* combustion mode. Based on the *n* combustion mode, advancing the start of diesel injection significantly, using large EGR (exhaust gas recirculation) rate and appropriately intake throttling can effectively reduce HC emissions and improve the BTE of dual fuel engines at low loads with relatively high natural gas PES (percentage energy substitution).

1. Introduction

In recent years, strict emission regulations, the environmental awareness, and the high price of conventional fuels have led to the creation of incentive to promote alternative fuels [1,2]. Among the alternative fuels, NG (natural gas) is very promising and highly attractive for its abundant resources, clean nature of combustion and low encouraging prices [3,4]. NG as an alternative fuel should be usable in CI (compression ignition) and SI (spark ignition) engines [5,6]. It has been successfully utilized in SI engines due to its similar fuel properties to those of gasoline fuel. CI engines have inherently higher compression ratios than SI engines, and correspondingly higher thermal efficiencies. NG does not autoignite under compression alone with typical CI-engine compression ratios. It is used to power CI engines via the dual-fuel mode, where a high-cetane fuel is injected along with the NG in order to provide a source of ignition for the charge. In recent years, as one of the main forms of NG engines, the diesel NG dual-fuel engine has been concerned widely [6]. Dual-fueling in CI engines reduces NOx emissions significantly compared with normal DI diesel engine operation [5]. At the same time, smoke, soot and particulate emissions in dual-fuel engines are very low and in some cases undetectable [5]. This is extremely difficult to achieve in DI diesel engines. Another attractive alternative fuel is bio-fuel, which is produced from renewable sources. Biofuels made from agricultural products (oxygenated by nature) may not only offer benefits in terms of exhaust emissions (some of those even defeating the NOx-Soot trade off), but also reduce the world's dependence on oil imports, with local agricultural industries supported and farming incomes enhanced. Among those, liquid bio-fuels, such as vegetable oils, their derived bio-diesels (methyl or ethyl esters), ethanol, butanol, and ethers are considered as very promising bio-fuels for engines [7–9].

Generally, diesel is directly injected into the cylinder around the combustion TDC (top dead center), providing an ignition source for the NG-air mixture, while NG is inducted into the intake port by MPI (multi-point injection) in dual fuel engines [5]. If the mixing process of the oxidizer and the fuel is much faster than the combustion reaction rates or the oxidizer has been mixed with the fuel before it reaches the flame front, this combustion process is called

* Corresponding author.
E-mail address: wangzhongshu@jlu.edu.cn (Z. Wang).

http://dx.doi.org/10.1016/j.energy.2015.11.052
0360-5442/

Nomenclature

AFR_{th}	theoretical air fuel ratio of natural gas
ATDC	after top dead center
BTDC	before top dead center
CA	crank angle
CA10	crank angle where 10% total heat released
CA50	crank angle where 50% total heat released
CAPHRR1	crank angle where the first peak of the heat release rate appeared
CI	compression ignition
CNG	compressed natural gas
$dP/d\theta$	pressure rise rate
$(dP/d\theta)_{max}$	maximum pressure rise rate
ED	eddy current
EGR	exhaust gas recirculation
HBC	HRR-balanced coefficient
HC	hydrocarbon
HCCI	homogeneous charge compression ignition
HRR	heat release rate
HRRCA50	heat release rate at CA50
ICE	internal combustion engines
MFC	mass flow controller
MFM	mass flow meter
MPI	multi-point injection
NG	natural gas
NOx	nitrogen oxides
P	in-cylinder pressure
PES	percentage energy substitution
P_{inj}	diesel injection pressure
P_{max}	maximum in-cylinder pressure
PHRR	peak of heat release rate
PHRR1	the first peak of heat release rate
PHRR2	the second peak of heat release rate
PM	particulate matter
ppm	parts per million
SCR	selective catalytic reduction
SI	spark ignition
SOI	start of diesel injection
TDC	top dead center
TSC	time-sequenced coefficient
THC	total hydrocarbon
TWC	three-way catalyst
UHC	unburned hydrocarbons
$\dot{m}$	mass flow rate
Φ	equivalence ratio

the premixed combustion. On the contrary, if the oxidizer combines with the fuel by diffusion, and the combustion rate is limited by the rate of diffusion, it is called the diffusion combustion. Hence, the combustion process of dual fuel engines includes the premixed combustion and the diffusion combustion (only when the amount of diesel is relatively large or the amount of diesel participating in the premixed combustion is relatively little) of diesel and the premixed combustion of NG. Compared with SI NG engines, CI diesel engines and HCCI (homogeneous charge compression ignition) diesel engines, dual fuel engines have more complicated combustion processes and show significantly particular characteristics [5]. In SI NG engines, NG is ignited by a spark plug and the flame front propagates in the homogeneous mixture by the thermal diffusion, which is known as the premixed combustion. The HRR (heat release rate) curve generally only has one peak, which appears very close to CA50 (crank angle where 50% total heat released). This is due to that the amount of NG ignited in the ignition delay is relatively small (It is generally considered to account for 10% of the total energy released in this cycle) [6], resulting from that the ignition energy provided by the spark plug is small and the ignition source is generally single point. In CI diesel engines, the diesel injected earlier into the cylinder participates in the premixed compression ignition combustion (It is generally considered to account for 10% of the total energy released in this cycle), which produces the first significant peak of the HRR resulting from the compression ignitions simultaneously at multiple regions in the cylinder. However, the diesel injected later participates in the diffusion combustion, which primarily depends on the thermal chemical reactions produced by diffusions of fuel vapor and oxygen. Therefore, the first peak of the HRR often appears near CA10 (the crank angle where 10% of the heat released) and far away from CA50 in CI diesel engines [10], and the premixed combustion stage and the diffusion combustion stage can be distinguished clearly by analyzing the curve of HRR. In HCCI diesel engines, the HRR only has one peak, and the peak is usually very close to CA50. This is because almost the entire diesel injected into the cylinder ignites simultaneously at multiple regions. Generally, there is no obvious flame propagation/flame surface and no obvious turbulent mixing effect in the combustion process of HCCI diesel engines. In other words, almost the entire diesel participates in the premixed combustion in HCCI diesel engines. It should be noted that the low temperature chemical kinetics reaction (i.e. cooled flame and blue flame reactions) is the most significant characteristic of HCCI combustion. However, the first peak of the HRR of the HCCI combustion in this paper refers to the first peak in the primary combustion stage, since the peak of the HRR produced by the low temperature chemical kinetics reaction is very low. The combustion process of diesel NG dual-fuel engines has features of a variety of combustion (e.g. CI diesel combustion, SI NG engine combustion and HCCI engine combustion) [5]. Diesel participates in the premixed combustion firstly resulting from that it is ignited by compression simultaneously at multiple regions, and then participates in the diffusion combustion, which primarily depends on the thermal chemical reactions produced by diffusions of fuel vapor and oxygen, if diesel is enough in dual-fuel engines. NG is ignited simultaneously at multiple regions depending on the ignition of diesel, and then participates in the premixed combustion by flame propagations depending on thermal diffusions. Therefore, the combustion process of dual fuel engines is the convergence of ignition in multiple regions simultaneously (for diesel and NG that burn earlier) like HCCI and CI engines, diffusion combustion (for diesel that burns later) like CI diesel engine, and premixed flame propagation (for NG that burns later) like SI NG engines.

In summary, compared to CI diesel engines, SI NG engines and HCCI diesel engines, dual-fuel engines have individual characteristics. First, the profile of the HRR curve is various. Papagiannakis and Hountalas [11] investigated the effect of natural gas percentage on performance and emissions of a DI dual fuel diesel engine. The results showed that it was easy to identify the first peak of the HRR curve, but it was hard for the second one with different natural gas percentage. Yang et al. [12] conducted some experiments to study the effects of natural gas injection timing on the combustion