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Diethyl ether as an ignition improver for biogas homogeneous charge compression ignition (HCCI) operation - An experimental investigation

K. Sudheesh, J.M. Mallikarjuna*

Internal Combustion Engines Laboratory, Department of Mechanical Engineering, Indian Institute of Technology Madras, Chennai 600 036, India

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ABSTRACT

This paper deals with experimental investigations of a homogeneous charge compression ignition (HCCI) engine using biogas as a primary fuel and diethyl ether (DEE) as an ignition improver. The biogas is inducted and DEE is injected into a single-cylinder engine. For each load condition, best brake thermal efficiency DEE flow rate is determined. The results obtained in this study are also compared with those of the available biogas-diesel dual-fuel and biogas spark ignition (SI) modes. From the results, it is found that biogas-DEE HCCI mode shows wider operating load range and higher brake thermal efficiency (BTE) at all loads as compared to those of biogas-diesel dual-fuel and biogas SI modes. In HCCI mode, at 4.52 bar BMEP, as compared to dual-fuel and SI modes, BTE shows an improvement of about 3.48 and 9.21% respectively. Also, nitric oxide (NO) and smoke emissions are extremely low, and carbon monoxide (CO) emission is below 0.4% by volume at best brake thermal efficiency points. Also, in general, in HCCI mode, hydrocarbon (HC) emissions are lower than that of biogas SI mode. Therefore, it is beneficial to use biogas-DEE HCCI mode while using biogas in internal combustion engines.

1. Introduction

Generally, spark ignition (SI) and compression ignition (CI) engines are most commonly used for power generation and in transportation vehicles. The SI engines have low smoke emissions and brake thermal efficiency with high HC, CO and NOx emissions. However, the CI engines show comparatively higher brake thermal efficiency, but they emit high amounts of smoke and NOx. In these engines, achieving low exhaust emissions with low fuel consumption is a challenge. Also, today's emission legislations are forcing engine manufacturers to search for engines having low exhaust emissions and higher fuel economy. Today, HCCI is emerging as an effective alternative combustion process for CI mode. With certain fuels, it can provide higher brake thermal efficiency like CI engines with ultra-low NOx and particulate matter (PM) emissions. In these engines, the premixed nature of a charge effectively eliminates the PM, whereas combustion with lean mixture without definite flame propagation reduces cylinder gas temperatures leading to ultra low NOx emissions simultaneously.

Biogas is an attractive source of energy for rural areas especially in countries like India. It is generated by anaerobic digestion of cow dung, other animal wastes and plant matters such as leaves and water hyacinth. All the above mentioned sources are renewable in nature and abundantly available. Biogas contains approximately two-thirds (by volume) of methane (CH_4) and the rest is mostly carbon dioxide (CO_2) with traces of hydrogen sulphide (H_2S). Biogas has a low energy density due to the presence of CO_2. The properties of biogas are given in Table 1. Usage of biogas in SI engines enhances knock resistance due to the presence of CO_2. However, it reduces brake thermal efficiency and increases HC emissions [1,2]. It is not possible to use biogas directly in CI mode due to its higher auto-ignition temperature. Dual-fuel mode of operation is a feasible way of using biogas in CI engines with diesel as a secondary fuel [3,4]. But, dual-fuel mode emits more HC and CO emissions with lower brake thermal efficiency. However, smoke and NOx emissions are lower as compared to the conventional CI mode due to the gaseous nature of biogas and the presence of CO_2.

Diethyl ether is considered as a renewable fuel because it can be produced from ethanol through the dehydration process [5]. Low autoignition and boiling temperature of DEE are reasons for selecting it as an ignition improver along with lethargic fuels like biogas. Also, DEE has a higher energy density than ethanol. The important properties of DEE are given in Table 2. Generally, DEE is used as a cold starting aid in CI engines. DEE is also blended with diesel for improving the brake thermal efficiency and reducing emissions in CI engines [6].

Onishi et al. (1979) introduced controlled autoignition combustion in a two-stroke engine in order to reduce instability at

* Corresponding author. Tel.: +91 44 22574698; fax: +91 44 22574652.
E-mail address: jmmallik@iitm.ac.in (J.M. Mallikarjuna).

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Nomenclature

λ_{BG}	biogas excess air ratio
λ_{DEE}	DEE excess air ratio
λ_T	total excess air ratios
M_{air}	mass flow rate of the air
MBTE	best brake thermal efficiency
NOP	nozzle opening pressure
M_{BG}	mass flow rate of the biogas
M_{DEE}	mass flow rate of the DEE
AF_{BG}	biogas stoichiometric air–fuel ratio
AF_{DEE}	DEE stoichiometric air–fuel ratio
CAD	crank angle degree

part-loads and they also achieved good reduction of emissions and fuel consumption [7]. Najt et al. (1983) extended the HCCI combustion into a four-stroke engine using primary reference fuels [8]. A four-stroke gasoline HCCI engine was tested by Thring et al. (1989) and the important parameters required for successful gasoline HCCI operation at part-loads were investigated [9]. Ryan et al. (1996) used a port fuel injection (PFI) injector to supply diesel into the intake air stream at various inlet air temperatures and compression ratios [10]. This resulted in early heat release during compression stroke itself, and they concluded that low compression ratio is most suitable for port injected diesel fuelled HCCI engine. Garcıa et al. (2009) investigated the effect of inlet charge temperature, cool EGR, injection timings and equivalence ratio on the performance of a diesel fuelled HCCI engine and achieved comparatively higher loads by using cool EGR [11]. Shi et al. (2006) studied the effect of internal and externally cooled EGR on the performance of a diesel fuelled HCCI engine. They concluded that internal EGR benefited the formation of homogeneous mixture and reduced smoke emission, whereas externally cooled EGR could help extend upper load limit of HCCI operation [12]. Due to the early heat release characteristics with low volatility of diesel, researchers tried to use various alternative fuels with high autoignition temperature, viz. natural gas, methanol, ethanol, liquefied petroleum gas (LPG) without many modifications to the original engine. Inlet charge heating, variable compression ratio (VCR) and use of secondary low octane fuels as an ignition improver were methods tried for achieving HCCI combustion with the above fuels. Christensen et al. (1997) investigated the HCCI combustion characteristics using natural gas with inlet air heating [13]. Swami Nathan et al. (2008) adopted acetylene as a fuel for HCCI engine because of its moderate autoignition temperature and high flammability limits. They used inlet charge temperature to control combustion phasing [14,15]. Chen et al. (2000) introduced dual-fuel HCCI combustion of natural gas with dimethyl either (DME). They found that by optimizing a proportion of DME and natural gas, NOx emissions could be lowered to near zero levels. The dual-fuel operation gave higher brake thermal efficiency than that of CI mode [16]. Zheng et al. (2004) used DME as an ignition controller in a methanol fuelled HCCI engine [17]. Mack et al. (2009) experimentally proved the suitability of wet ethanol in HCCI combustion. They used inlet charge heating to control the combustion phasing [18]. Swami Nathan et al. (2008) investigated biogas HCCI combustion with manifold injection of diesel. They compared the results of biogas-diesel HCCI mode with that of biogas-diesel duel-fuel mode of operation. Due to the low volatility and high boiling temperature of diesel, inlet heating was used for proper mixing of diesel with manifold inducted biogas. They concluded that biogas-diesel HCCI operation is superior to dual-fuel mode of operation in a BMEP range of 2.5 to 4 bar [19]. However, they couldn't operate an engine in HCCI mode below 2.5 bar and above 4 bar BMEPs due to system limitations and difficulty in controlling inlet charge heating. The motivation for the present work is to operate a single-cylinder engine in biogas fuelled HCCI mode in a wide load range by using DEE as an ignition improver. These types of studies are limited in literature. Therefore, a detailed study would help not only to evaluate and compare the HCCI mode with conventional biogas SI and biogas diesel-dual-fuel modes, but also to find the feasibility of using biogas in HCCI mode.

In this study, the effect of DEE flow rate on the performance, emissions and operating load range of a biogas fuelled HCCI mode are studied at a constant engine speed of 1500 rev/min., and cooling water outlet temperature of 50 °C. Finally, DEE mass flow rate for best brake thermal efficiency point at each load condition has been found out. In addition, the performance and emission characteristics of biogas-DEE HCCI mode are compared with the available results of biogas SI [1,2] and biogas-diesel dual-fuel [3] modes.

2. Experimental setup

A single-cylinder, water-cooled, direct injection CI engine is used for conducting experiments. The engine is coupled to an eddy current dynamometer for loading and measurement purposes. The engine specifications are shown in Table 3. The DEE is stored in an accumulator and is injected into the intake manifold using an injector at a line pressure of 2 bar. An in-house built electronic circuit is used for controlling DEE flow rate. The biogas is directly inducted through the intake manifold. The biogas is generated in a nearby plant, which uses cow dung and water to produce it. It is collected in a flexible bag at the plant and transported to the place of usage. It is actually sent through a floating drum in order to maintain the

Table 1
Properties and composition of biogas.

Calorific value		17 MJ/kg
Density (1 atm and 15 °C)		1.2 kg/m^3
Flame speed		0.25 m/s
Stoichiometric air fuel ratio (mass basis)		5.7
Autoignition temperature		650 °C
Flammability limits with air (%)		7.5–14
Research octane number		130
Typical biogas composition in % volume		
Methane	57.37	
Carbon dioxide	42.1	
Carbon monoxide	0.08	

Table 2
Properties of DEE.

Calorific value	33.9 MJ/kg
Density	713 kg/m^3
Boiling point	34.4 °C
Stoichiometric air fuel ratio (mass basis)	11.1
Autoignition temperature	160 °C
Cetane number	>125

Table 3
Engine specifications.

Bore × stroke	80 × 110 mm
Connecting rod length	231 mm
Compression ratio	16:1
Rated power output	3.7 kW @ 1500 rpm
Displacement volume	553 cm^3
Injector NOP	220 bar