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Study on biogas premixed charge diesel dual fuelled engine

Phan Minh Duc *, Kanit Wattanavichien

Mechanical Engineering Department, Faculty of Engineering, Chulalongkorn University, Phaya-Thai Road, Patumwan, Bangkok 10330, Thailand

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Abstract

This paper presents an experimental investigation of a small IDI biogas premixed charge diesel dual fuelled CI engine used in agricultural applications. Engine performance, diesel fuel substitution, energy consumption and long term use have been concerned. The attained results show that biogas–diesel dual fuelling of this engine revealed almost no deterioration in engine performance but lower energy conversion efficiency which was offset by the reduced fuel cost of biogas over diesel. The long term use of this engine with biogas–diesel dual fuelling is feasible with some considerations.

Keywords: Dual fuel; Biogas; Premixed charge; IDI

1. Introduction

Biogas, produced by the anaerobic fermentation of cellulose biomass materials, is a clean fuel for internal combustion engines. In oil crisis situations, it may act as a promising alternative fuel, especially for diesel engines, by substituting for a considerable amount of fossil fuels. Diesel engines can be easily converted to fumigated dual fuel engines. This is the most practical and efficient method to utilize high spontaneous ignition temperature alternative fuels, such as biogas. In the fumigated dual fuel method, biogas mixes with air before this mixture enters the combustion chamber, and at the end of the compression stroke, an amount of diesel fuel, called the pilot injection, is injected to ignite it. This method has the advantage of the ability to switch back to diesel operation in case of a shortfall in biogas supply during an important operation. Because of these benefits, dual fuelling of diesel and biogas [1–4], as well as producer gas [5–10], LPG [11–16], NG [17–26] or hydrogen [11,27,28], have been investigated widely worldwide for some past decades. Karim G.A. et al. [11,17,18,29–31] have investigated dual fuel operation with different gaseous fuels (hydrogen, methane, propane, CNG, LPG) with respect to engine performance, combustion characteristics, exhaust gas emissions and factors influencing them. These factors include the engine loads, diesel substitution, injection timing, intake air temperature and EGR. They concluded that the prolonged ignition delay caused by the presence of gaseous fuel in the compression process, the reduction of oxygen concentration in the charge and the increase in the polytropic index of the charge leads to significant changes in combustion characteristics, exhaust gas emission, engine performance and fuel consumption. This was confirmed by other researchers in this field [20,21,24]. A considerable number of past investigations concentrated on engine performance and fuel consumption. While those revealed decreases in engine output [32], others reported unchanged [19] or even increased [12,14,33] output. A loss in thermal efficiency had been reported by some authors [25,34], whereas others stated comparable or higher efficiencies [15,35–38] or loss at low to medium loads but gains at high to full loads [13,33,39,40]. Solutions to improve dual fuel part load have been investigated and proposed, such as throttling the intake air charge [41], increased intake air pressure [42], temperature [11,25,43,44], controlled amount and time of

* Corresponding author. Tel.: +66 2 2186607; fax: +66 2 2522889.
E-mail address: duc.aut@ud.edu.vn (P.M. Duc).

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Nomenclature

CI	compression ignition	SDC	specific diesel consumption, g/kWh
DI	direct injection	STEC	specific total energy consumption, MJ/kWh
IDI	indirect injection	D_S	diesel substitution
NG	natural gas	A/V	surface to volume ratio of combustion chamber, m^{-1}
CNG	compressed natural gas	m_d	mass of diesel fuel delivered per engine cycle, kg/cycle, mg/cycle
LPG	liquefied petroleum gas	m_{air}	mass of air sucked per engine cycle, kg/cycle
EGR	exhaust gas recirculation	$\overset{*}{m}_{d\_DDF}$	diesel mass flow rate in DDF operation, kg/s
DDF	diesel dual fuel	$\overset{*}{m}_{d\_D}$	diesel mass flow rate in straight diesel operation, kg/s
TDC	top dead center	(A/F)s	stoichiometric fuel air ratio
BDC	bottom dead center	$(A/F)_{s,d}$	stoichiometric air fuel ratio of diesel fuel
bTDC	before top dead center	$(A/F)_{s,biog}$	stoichiometric air fuel ratio of biogas
CA	crank angle	Φ	fuel air equivalent ratio
UHC	unburned hydrocarbon	bmep	brake mean effective pressure, kPa
LHV	lower heating value		
FTIR	fourier transform infrared spectroscopy		
TBN	total base number		

pilot injection [11,12,15,35,45] or controlled EGR flow and temperature [25,40,46–48]. While more information of the type and composition of gaseous fuels used were provided, less detailed information about engine geometry was mentioned. This makes it more difficult to assess/analyze the reported results since the engine performance, thermal efficiency, diesel substitution and exhaust gas emissions depend not only on the physical/chemical properties of the gaseous fuels but also on the engines used. In addition, almost all past investigations were conducted with engines on test benches at which engine cooling water and lube oil temperatures had been controlled to ensure not exceeding a predetermined value. This is contrary to real operational conditions at which the temperatures may increase to high levels. The reported information about long term use with dual fuelling has also not been reported clearly/fully.

It seems that dual fuel operation for IDI engines is less effective than for DI engines because of too high surface to volume ratio of the combustion chamber. In addition, the difference in combustion chamber geometry of this type would have an effect on the dual fuel characteristic. In this work, a comparative investigation between straight diesel and biogas premixed charge diesel dual fuel CI (henceforth called DDF) was conducted to obtain clear information. The following aspects were concerned: engine performance, diesel substitution, energy consumption and the effect of long term use.

2. Description

2.1. Test system

The test system installation is shown schematically in Fig. 1. A small single cylinder IDI CI engine Kubota RT120 with specifications shown in Table 1 was used. There was no engine modification except a gas mixer designed particularly for it and added to the intake manifold as a means to introduce biogas. The engine was coupled with an alternator to form a system loaded by variable resistances. Engine load is the product of alternator current and voltage divided by the mechanic-electricity conversion efficiency of this system. This efficiency had been determined prior to this study to ensure correct engine load setting.

The biogas fuel used has been produced by a biogas producing system at a pig farm in Ratchaburi province, Thailand. Biogas, with pressure higher than atmospheric, from very big cellars of the producing system is led by the pipe system and introduced to the engine via the gas mixer to form a homogeneous charge prior to combustion. Its flow rate was controlled manually by a regulator and a valve located upstream of the mixer.

The consumed intake air and biogas flow rates were measured by means of an orifice plate and inclined manometers. The engine speed signal was sensed by a photodiode sensor. A data acquisition system and a computer program was designed and installed to collect engine speed and load, time to consume a fixed diesel fuel volume ($43 \pm 0.01\ cm^3$) and the temperatures of the intake air, cooling water, lubricant oil and exhaust gas at a frequency of 1 Hz. These data were stored in the computer hard disk for off line calculation and analysis. The instruments used are listed in Table 2.

2.2. Fuels properties

Thai commercial diesel fuel, with the main properties given in Table 3, was used throughout this investigation. The properties of the biogas obtained from very big cellars of the producing system remained nearly unchanged during the test period. Its main properties were determined and shown in Table 4. As observed, methane is the main