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Studies on the effects of methane fraction and injection strategies in a biogas diesel common rail dual fuel engine

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ABSTRACT

Biogas is an environment friendly renewable fuel which is a valuable resource in the current context of increased energy requirement and sustainability. The quality or the proportion of methane in biogas can vary significantly based on the raw material and method of production. This experimental work was aimed at evaluating the influence of such variations in composition on the energy conversion efficiency and emissions of a common rail dual fuel engine under different output conditions. The effects of post and pilot injection of diesel were also studied in this mode.

Biogas with methane proportions in the range 24–68% could be utilized without significant changes in efficiency and emissions till a biogas energy share (BGES) of 60% when the injection timing of diesel was suitably adjusted. Higher than normal methane concentrations (normal: 51–53%) only elevated the NO levels with little impact on efficiency. However, when low proportions of methane were used NO could be controlled effectively particularly at low BGES. Simulation studies indicated that this reduction in NO is due to the lowered in-cylinder temperature rather than the reduced concentration of oxygen as a result of increased CO_2. When the proportion of methane was decreased from 68% to 24% the start of injection of diesel had to be advanced by 3 °CA (at a BGES of 60%) to compensate for the increase in ignition delay and reduction in combustion rate. With pilot injection there was a reduction in smoke emission because of improved charge homogeneity due to the split injection process. However, post injection which is generally effective in diesel engines was not advantageous in the biogas diesel dual fuel (BDDF) mode because of the diffusion combustion the post injected fuel undergoes.

1. Introduction

Biogas is a good renewable gaseous fuel for decentralized power generation [1–3] particularly in rural areas. It can be produced by the anaerobic fermentation of bio-wastes like cow dung and unwanted vegetable matters in a digester. It contains predominantly methane (CH_4: 40–75%) while the remaining is mostly carbon dioxide (CO_2: 20–55%) with traces of hydrogen sulfide (H_2S) and moisture. Based on the feed stock, the concentration of methane in biogas can vary significantly. The proportion of CH_4 in biogas obtained from different sources is as follows; landfill: 40%, algae: 50%, cow dung: 55% and oil seed cake: 70% [2,4,5]. Biogas has a high self-ignition temperature (650 °C) with a low flame speed (0.25 m/s) and limited flammability range as compared to natural gas as shown in Table 1. The presence of CO_2 reduces its calorific value and stoichiometric air requirement [6]. In large industries, CO_2 can be removed from biogas conveniently using different techniques [7,8]. However, in small rural plants this will not be viable and one has to use the raw biogas directly.

Abbreviations: BDDF, biogas diesel dual fuel; BGES, biogas energy share; BMEP, brake mean effective pressure; BTE, brake thermal efficiency; CA, crank angle; CA50, crank angle at which 50% mass burn occurs; CE, combustion efficiency; CH_4, methane; CI, compression ignited; CNG, compressed natural gas; CO, carbon monoxide; CO_2, carbon dioxide; COV, coefficient of variance; DAQ, data acquisition system; DP, double pulse; ECU, engine control unit; EGR, exhaust gas recirculation; EGT, exhaust gas temperature; EPi.I, early pilot injection; FID, flame ionization detector; FSN, filter smoke number; H_2, hydrogen; H_2S, hydrogen sulphide; HC, hydrocarbon; HCCI, homogenous charge compression ignition; HRR, heat release rate; ICT, intake charge temperature; IMEP, indicated mean effective pressure; kW, kilowatt; Max. ROPR, maximum rate of pressure rise; MBD, mass burn duration; MBT, maximum brake torque; MI, main injection; NDIR, nondispersive infrared detector; NO, nitric oxide; NO_x, oxides of nitrogen; °aTDC, degree after compression top dead center; °bTDC, degree before compression top dead center; O_2, oxygen; PI, post injection; PM, particulate matter; ppm, parts per million; RCCI, reactivity controlled compression ignition; SI, spark ignited; SOC, start of combustion; SOIT, start of injection timing; SP, single pulse; THC, total hydrocarbon; Vol %, volume percent; η, efficiency

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Table 1
Comparison of properties of biogas and natural gas [6].

Properties	Biogas	Natural Gas
Composition (by volume)	CH_4 – 57%, CO_2 – 41%, CO – 0.18%, H_2 – 0.18% H_2S + Moisture – Balance	CH_4 – 85%, C_2H_6 – 7% C_3H_8 – 2%, CO_2 – 5% N_2 – 1%
Calorific value (MJ/kg)	17	50
Density at 1 atm and 15 °C (kg/m^3)	1.2	0.79
Research octane number	130	120
Flammability limits (vol% in air)	(7.5–14)	(5–15)
Stoichiometric A/F (kg of air/kg of fuel)	5.7	17.3
Auto-ignition temperature (°C)	650	540
Flame speed (m/s)	0.25	0.34
Octane number	160	130

1.1. Biogas in SI engines

In a spark ignited (SI) engine neat biogas can be directly utilized because of its high resistance to knocking which is due to its high self-ignition temperature [6]. Techniques like enhanced compression ratio, removal of CO_2 and use of small amounts of fast burning fuels along with biogas have shown that enhancements in thermal efficiency and reduction in emissions are possible in biogas based SI engines [9,10]. Effect of EGR (exhaust gas recirculation) (15%) was also studied in a biogas SI engine and it resulted in a significant reduction in NOx emission (from about 220 to 40 ppm) however with a reduction in efficiency [11].

1.2. Biogas in CI engines

Combustion of neat biogas in a compression ignition engine requires intake charge temperatures (ICT) above 250 °C [12]. Biogas is generally utilized in compression ignition (CI) engines in the dual fuel mode because of its high self-ignition temperature and low cetane number. This can reduce particulate and NOx emissions as compared to neat diesel engines [13–15]. Mathur et al. were some of the early people who worked on biogas fuelled engines with diesel as a secondary fuel [16]. An accurate control over the pilot fuel quantity is also essential in dual fuel engines for good performance [17]. Utilization of biogas in the dual fuel mode has the disadvantages of low thermal efficiency, high hydrocarbon and carbon monoxide emissions when compared to the CI mode because of the incomplete combustion of biogas [18]. In a CI engine where biogas was inducted along with air and diesel was directly injected, PM (particulate matter) emission reduced by about 70% and also a decrease in the NOx emission was observed [19] due to the high proportion of CO_2 [20]. Increased ignition delay, longer combustion duration and reduction in exhaust gas temperature (EGT) have been observed in biogas diesel dual fuel engines as compared to operation on neat diesel [21–23]. Increase in compression ratio decreased the hydrocarbon and carbon monoxide emissions in a biogas diesel dual fuel engine [24]. In another work with biogas the efficiency of a dual fuel engine reached values comparable to the neat diesel at full load [25]. With synthetic biogas also in a dual fuel engine thermal efficiency higher than neat diesel values were reached along with a reduction in HC emissions because of improved combustion [26]. When CNG, H_2 and biogas were used along with diesel advancement in the start of injection timing (SOIT) of diesel increased the heat release rate (HRR) and thus improved the performance [27].

1.3. Influence of varying the composition of biogas in SI and CI engines

When the proportion of CO_2 in synthetic biogas was changed from 10 to 40% NOx emission was reduced from 5500 to 1500 ppm in a SI engine [28]. Reducing the amount of CO_2 in biogas and enhancing the level of swirl to improve the combustion rate were effective in a biogas SI engine to reduce HC emissions and improve the thermal efficiency but with increased NO emission [6,8,9,29]. It was possible to substitute 37% of gasoil by natural gas with added CO_2 in a modified diesel engine without affecting the overall efficiency. It is also reported that the CO emission is majorly affected by natural gas substitution rather than the proportion of CO_2 in the mixture [30]. Enhancing the proportion of methane from 55 to 81% in a Jatropha oil – biogas dual fuel engine improved the brake thermal efficiency significantly along with a reduction in HC emissions [31]. However, in another work (biogas-diesel dual fuel mode) when the proportion of CO_2 was varied from 0 to 30 % the thermal efficiency was not affected significantly [32]. The best methane fraction was found to be 70% for dual fuel operation in another work [20]. When high proportions of CO_2 (49%) in biogas were used combustion was unstable, which led to vibrations [33].

1.4. Multiple pulse injection of diesel in compression ignition engines

Use of multiple injection strategies has enabled diesel engines to attain improved performance with reduced emissions. In a small diesel engine with the multiple pulse injection strategy (i.e. with post and pilot injection) along with EGR a 40% reduction in particulate emissions was reported as a result of improved particulate oxidation due to increase in-cylinder temperature with the post injected diesel. It was also recommended that the post injection quantity has to be kept low to reduce the soot emission so that no deterioration in thermal efficiency occurs [34]. Proper selection of the injection timing and quantity of the post injection pulse are important [35]. Splitting up of the pilot pulse into two resulted in the reduction in cylinder wall wetting [36]. Thermal efficiency was reported to drop with higher quantities in the secondary pulses even though there was enough dwell between the pulses in a diesel engine with triple pulse strategy (all SOITs were after TDC) [37]. In a homogeneous charge compression ignition (HCCI) engine when diesel was injected in eight pulses the NO emission was decreased to a very low level of 12 ppm because of increased homogeneity. HC emission was also lowered because of the reduction in wall impingement in the same work [38]. In a reactivity controlled compression ignition (RCCI) engine with natural gas induction and direct injection of diesel, increase in smoke was observed when the first injection of diesel was very early and the second one was after TDC. This was because the second injection occurred after start of combustion and led to diffusion combustion [39]. Biogas diesel HCCI operation results in considerably lower NO emissions as compared to the dual fuel mode, however, with limited load range [40–42]. Earlier work of the present authors indicated that by changing the amount of CO_2 in biogas it was possible to extend the load range, reduce the smoke emissions and also enlarge the range of biogas to diesel proportions in a HCCI engine fueled with biogas and diesel [43].

On the whole, the composition of biogas can have a significant