Energy 89 (2015) 148–157

Contents lists available at ScienceDirect

Energy

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

Effect of partial replacement of diesel or biodiesel with gas from biomass gasification in a diesel engine

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ARTICLE INFO

Article history:
Received 15 November 2014
Received in revised form
15 April 2015
Accepted 7 July 2015
Available online 4 August 2015

Keywords:
Biomass gasification
Diesel engine
Dual-fuel
Gaseous emissions
Particulate matter
Biodiesel

ABSTRACT

The injected diesel fuel used in a diesel engine was partially replaced with biomass-derived gas through the intake port, and the effect on performance and pollutant emissions was studied. The experimental work was carried out in a supercharged, common-rail injection, single-cylinder diesel engine by replacing diesel fuel up to 20% (by energy), keeping constant the engine power. Three engine loads (60, 90, 105 Nm), three different EGR (exhaust gas recirculation) ratios (0, 7.5, 15%) and two intake temperatures (45, 60 °C) were tested. Finally, some of the tested conditions were selected to replace diesel injection fuel with biodiesel injection. Although the brake thermal efficiency was decreased and hydrocarbons and carbon monoxide emissions increased with increasing fuel replacement, particulate emissions decreased significantly and NO_x emissions decreased slightly at all loads and EGR ratios. Thermodynamic diagnostic results showed higher premixed ratio and lower combustion duration for increasing diesel fuel replacement. High EGR ratios improved both engine performance and emissions, especially when intake temperature was increased, which suggest removing EGR cooling when diesel fuel is replaced. Finally, when biodiesel was used instead of diesel fuel, the gas replacement improved the efficiency and reduced the hydrocarbon, carbon monoxide and particulate emissions.

1. Introduction

Although stationary SI (Spark Ignition) engines are nowadays the primary option to transform the chemical energy of a gas into mechanical energy, diesel engines are receiving increasing attention for this application in the last decade mainly for their higher thermal efficiency. In this case, the poor autoignition quality of gaseous fuels requires injecting a small volume of liquid diesel fuel as a pilot fuel around TDC to ensure stable and optimized combustion timing. In this combustion mode, generally known as dual mode, the majority of the energy is released by the gaseous fuel (primary fuel). However, the scale of the biomass gasifiers is often too small for supplying the amount of gas required in gas engines or dual diesel engines to ensure stable operation. Although not sufficiently explored, another option is to replace the diesel fuel with gas from biomass gasification in a minor proportion, with the diesel fuel remaining as the primary fuel. This option is especially interesting for stationary engines used in small–medium scale cogeneration plants of industries producing biomass residues.

The composition of the gas obtained from gasification is variable depending on the type of biomass, the type of gasifier, the operating conditions (temperature, pressure, relative fuel/gasifying agent ratio, etc.), and the type of gasifying agent (mainly air, oxygen, steam or a mixture of them). For engine applications, among the different options, gasification with steam has the advantage of reducing the inert gas content and increasing the heating value of the gas [1]. With regard to the biomass, among the typical industries in Mediterranean Europe which could benefit from this engine configuration, those producing dealcoholized marc of grape (distilleries and wineries) are abundant.

Despite the mentioned benefits in efficiency, and consequently in direct CO_2 emissions, the diesel engine option requires controlling tailpipe gaseous emissions such as total HC (hydrocarbons), NO_x (nitric oxides) and CO (carbon monoxide), and PM (particulate matter) in both mass and number/size basis. The addition of a gaseous fuel has an interesting potential to reduce mainly PM, as well as life-cycle CO_2 emissions, as far as the original feedstock comes from renewable and waste sources, as in the present case.

With appropriate and relatively simple modifications, diesel engines can be prepared to burn gaseous fuels efficiently [2]. With the described configuration, a mixture of air and gaseous fuel is compressed. Among the techniques reported in the literature to

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http://dx.doi.org/10.1016/j.energy.2015.07.050
0360-5442/

inject or induct the gaseous fuel in the air manifold, carburetion, TMI (timing manifold/port injection), DI (direct gaseous fuel injection) and CMI (continuous manifold induction) are the mostly used [3]. The compressed mixture of air and gas fuel does not autoignite due to its high autoignition temperature, but is fired by a liquid fuel injection which ignites spontaneously at the end of the compression stroke [1,4] and is then burned with acceptable efficiency. However, the fact that most of dual engines are prepared to operate interchangeably in both dual and single fuel modes usually leads to a substantial loss of efficiency in both cases [5].

Numerous studies have been reported in the literature about dual diesel combustion of different gaseous fuels [1], such as natural gas [6–10], hydrogen [11], biogas [12,13], propane [14], LPG (Liquefied Petroleum Gas) [15], DME (Dimethyl Ether) [16] or even of low-cetane volatile liquid fuels such as alcohols [17] and gasoline [18]. The latter fuels are either fumigated or port-injected, and their combination with high cetane fuels provides a mean to optimize the combustion phasing and duration through technologies such as PCI (Premixed Compression Ignition) or RCCI (Reactivity Controlled Compression Ignition) [19]. Also considerable information can be obtained about dual diesel combustion of syngas (or producer gas) from biomass gasification in dual diesel engines [5,20–23], although in this case, the tar content in the gas should be eliminated or minimized [5]. A more limited number of references was found about the use of biodiesel in dual mode with different gaseous fuels (natural gas [24,25], biogas [26,27], hydrogen or reforming gas [28], propane [29], producer gas from biomass gasification [30], simulated syngas [31] or volatile fuels [17]). Most of the previous studies consider the gaseous fuel as the primary energy source, while the role of the liquid injected fuel is limited to ensure a stable autoignition and a timely initiation of the flame propagation. On the contrary, studies reporting about gaseous fuels being burned in the partial-replacement mode are scarce [32–34], and none of them used gas from biomass gasification as a secondary fuel. In this configuration the combustion is mainly diffusive and the flames are concentrated around the injection sprays. With an appropriate specific optimization for a given partial replacement of the liquid fuel, the loss of efficiency derived from the interchangeable operation (switching from single liquid fuel or dual pilot fuel) could be avoided, or at least minimized.

The objective of this study was to investigate the effect of replacing diesel or biodiesel fuels with gas from biomass gasification, while keeping the same engine power, on the performance and emissions, including particulate matter, under different operating conditions. Another objective is to propose guidelines to optimize the engine operation conditions for this configuration as a function of the amount of energy replaced.

2. Experimental setup, test matrix and materials

The engine testing bench consists of an AVL 501 single-cylinder connected to a Piller 225.26B dynamometric brake. The geometrical and distribution parameters of the engine are shown in Table 1. All the auxiliary systems (such as the intake, exhaust, refrigeration, lubrication and common-rail injection systems) can be controlled individually (see Fig. 1).

The injection pressure of diesel fuel was 1200 bar, while the injection timing was modified from 20 deg bTDC (before Top Dead Center) to TDC, and later fixed at 12 deg bTDC. The engine speed was kept constant at 1500 rpm (usual engine speed for electric generation in Europe), and the torque was set to 60 Nm, 90 Nm and 105 Nm. The two latter torque values, 90 and 105 Nm, are within the usual range of torque for this engine size, whereas 60 Nm is below the usual torque range in stationary engines, but it was selected as an example of low-efficient and high-emission mode.

Table 1
Technical characteristics of the single-cylinder engine.

Geometrical data	Number of cylinders	1
	Cylinder bore	130 mm
	Connecting rod length	275 mm
	Cylinder volume	1999 cm^3
	Comb. chamber volume	135.2 cm^3
	Compression ratio	15.78
	Bowl volume	101 cm^3
	Bowl diameter	74 mm
Distribution data	Number of inlet valves	2
	Number of exhaust valves	2
	Inlet valve opening	15 deg bTDC
	Inlet valve closure	50 deg aBDC
	Exhaust valve opening	72 deg bBDC
	Exhaust valve closure	16 deg aTDC

When the injected fuel was partially replaced, injection was reduced and then the gaseous fuel was introduced in the intake manifold until both engine speed and torque initial values were recovered. EGR ratio was initially eliminated, but then it was increased to 7.5% and to 15% in mass (by opening the EGR valve with control of CO_2 exhaust concentration), without cooling. Two different liquid fuels were used as injected fuels. First, a diesel fuel typically used in Europe, distributed by CEPSA, containing 5.8% biodiesel. Second, a biodiesel fuel derived from 90% w/w soybean oil and 10% w/w palm oil. The main properties of these fuels are shown in Table 2. In the case of injecting biodiesel fuel, an additional test was made with heated EGR, trying to improve the combustion efficiency of the inlet gas.

The in-cylinder pressure was measured with a piezoelectric pressure transducer (Kistler 6041A). The chamber pressure signal was monitored with the data acquisition system YOKOGAWA OR1400. The regulated gaseous pollutant emissions such as NO_x, CO, CH_4 and THC were measured with an ENVIRONNEMENT S.A equipment. PM emissions were measured with a dilution sampling system NOVA MESS TECHNIK. The flow rate of the gaseous fuel is controlled through a mass flow controller manufactured by BRONKHORST. The gaseous fuel is fed, together with the air, at the intake manifold. The intake pressure of the mixture (air and gaseous fuel) was kept constant at 1.2 bar and the temperature was maintained at 35 °C when EGR was not used.

Simulated producer gas was used for all tests. The composition corresponds to a steam gasification process performed in a small-scale drop-tube gasification pilot plant (described in Ref. [35], later modified as described in Ref. [36]) using dealcoholized marc of grape as biomass. This waste has been proven as an interesting biomass fuel, for two reasons: a) it is an abundant solid biomass residue obtained in wineries during the production of wine, and b) the resulting gas is similar to that obtained from other abundant sources for energy from the inland regions of Spain, such pine (Pinus pinaster), olive (Olea europaea) and grapevine (Vitis vinifera) [35]. Table 3 shows the experimental conditions of the gasification tests leading to the simulated producer gas composition, which was selected based on a previous work [1]. In this table, subscripts *f* denote solid fuel, *daf* means fuel in a dry ash-free basis and *s* means steam, *S/B* refers to the steam/fuel mass ratio, *tr* is the space residence time and *LHV* the lower heating value of the producer gas (MJ/kg).

3. Results and discussions

3.1. Engine timing optimization

A preliminary study was carried out varying the diesel fuel injection timing in order to optimize this variable when gasification