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Performance analysis of a producer gas-fuelled heavy-duty SI engine at full-load operation

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Abstract

Biomass gasification converts a solid fuel into a gaseous mixture (syngas or producer gas) which can be burnt in reciprocating internal combustion engines (ICEs) to produce electrical power. A wide variety of bio-residues can be processed to obtain syngas, making biomass gasification a very interesting way to exploit the energy content of industrial by-products and agricultural wastes.
This paper focuses on the operation of a spark ignition (SI) ICE burning low-heating value gas produced in a fixed-bed downdraft gasifier. The biomass gasification power plant has collected more than nine months of operation till now without need of any extraordinary maintenance of the engine. Engine performance is calculated using experimental data acquired at different air-to-fuel ratios and spark timings, and then compared with results of test performed by other authors. The work is mainly aimed at analysing the effect of PG fuelling on brake power, efficiency and emissions of heavy-duty engines.

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Keywords: fixed-bed downdraft gasifier; producer gas-fuelled internal combustion engine; biomass power plant

1. Introduction

The recent introduction of European policies aimed at reducing the green-house gases emissions and the dependence on fossil fuels has increased the use of alternative fuels in both transportation and power generation. While in transportation liquid fuels are preferred due to their high energy density per unit volume, gaseous fuels are extensively used for power generation because of the possibility of cleaner combustion [1] compared to liquid and solid fuels. In this context, biomass gasification is receiving an

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increasing attention as interesting solution to utilise various organic wastes coming from industrial and agricultural processes with a considerable CO_2 emission reduction.

In a gasification process a wide variety of bio-residues are converted into a gaseous fuel (usually termed as syngas, if the heating value ranges from 10 to 28 MJ/Nm^3, or producer gas (PG), if the heating value ranges from 4 to 7 MJ/Nm^3 [2]) which can be directly used in internal combustion engines (ICEs), gas turbines and fuel cells [3]. When a gasifier must be coupled to an ICE, downdraft layout is the best choice, because of the low tar content in the PG resulting from the gasification process.

So far, the research on the use of PG into ICEs has been limited by the variability in gas composition [4], and some misconceptions related to the auto-ignition tendency at high engine compression ratio and the large power de-rating due to the gas low heating value [1,3,5]. These issues will be briefly discussed in the next Section. In addition, only part of the experimental studies found in the literature focuses on 100% PG fuelled SI engines, since low-heating value fuels can be also burnt in dual-fuel compression ignition (CI) engines [3]. Finally, tests are usually performed on small engines due to the high fuel consumption during PG operation [4, 6, 7].

In this paper the performance of a heavy-duty 12-cylinder ICE is presented and analysed. The engine, design to be fuelled with natural gas (NG), was modified to burn a PG stream resulting from the gasification of wood chips into a downdraft fixed-bed gasifier. Engine electrical power output and emissions were measured at several engine settings (spark timings and air-to-fuel ratios). Starting from these data, the efficiency of the engine is calculated. Missing parameters needed for the computation are estimated using NG operation data provided by the engine manufacturer. Producer gas composition needed to calculate the fuel heating value and the stoichiometric air-to-fuel ratio is computed by using a model of the gasifier in which the values of some operating parameters were measured on the real plant. The aims of the paper are twofold: (i) to increase the still limited database of PG-fuelled medium-size ICE performance and emissions. (ii) to provide a simplified procedure to calculate the engine efficiency starting from a rather basic set of measurements.

2. SI engines fuelled with PG

Compared to fossil fuels, the power de-rating in PG-fuelled engines is mainly caused by the lower energy density per mixture unit volume (15-30% less than NG [1,5]) and by the drop in volumetric efficiency. Fuel conversion efficiency η_f (see Eq.(1)) contributes to power de-rating as well. On the basis of the few data found in the literature, η_f may reduce from 10% to 20% compared to fossil fuel operation [1,3]. On the whole the power de-rating can range from 30% to 70%, depending on the gas quality [4,5], and can be partly reduced by increasing the engine CR. In fact, CO_2 and N_2, which constitute about 60% of a PG volume, act as knock suppressor and explain the high methane number compared to NG [1,5].

In [7] the authors demonstrated that PG engines can operate stably under lean condition (actual air-to-fuel ratio/stoichiometric air-to-fuel ratio, λ, equal to 2), although η_f decreases when λ exceeds 1.5. The authors attributed this drop to the decrease in flame speed. Stationary engines tested in [8] showed stable operation with leaner mixtures (up to λ close to 3).

Except for CO_2, PG generates lower quantities of pollutant emissions when burnt in ICEs compared to fossil fuels [1]. In fact, flammable components in PG are mostly simple molecules (easy to oxidise), and combustion temperature is relatively low, limiting the NO_x production.

3. Biomass power plant

The biomass power plant is constituted by a gasifier, a gas cleaning unit and a naturally-aspired ICE which can operate also in combined heat-and-power mode (Fig. 1). The gasifier is a typical closed-top fixed-bed downdraft gasifier suitable to be directly coupled to power units up to 150 kW_e. The biomass (wood chips) is introduced through the top of the gasifier. The gasification air is inflated in the combustion zone by a blower, after being preheated by the PG stream exiting the gasifier. Producer gas is cleaned from tar and particles in a train of filters and then further cooled to about 70 °C in a gas-water heat exchanger before entering the ICE container.

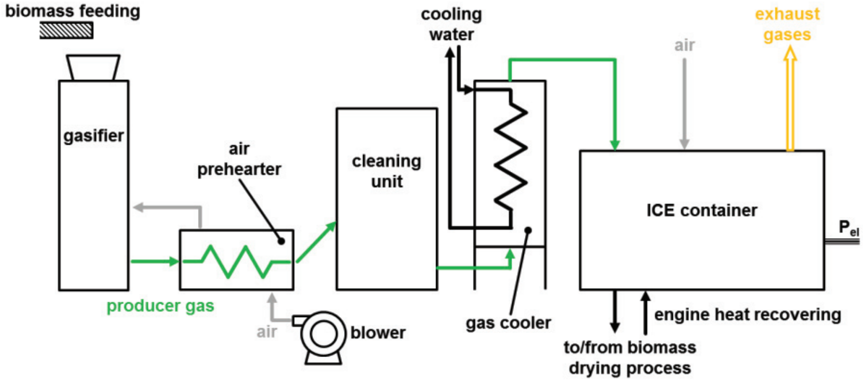

Fig. 1. Biomass power plant scheme.

The engine is the MAN®21.927-litre natural gas engine E 2842 E 312 modified to burn low-heating value fuels. In particular, the original Venturi fuel-air mixer was replaced by a simple "T" mixer (Fig. 2). A globe valve is installed in between the mixer and the air filter to adjust the air mass flow rate, and consequently the air-to-fuel ratio. Heat recovered from the engine coolant and by exhaust is used to dry the biomass which fed the gasifier.

The NG engine rates 250 kW@1500 rpm with a fuel conversion efficiency η_f equal to 0.375 at stoichiometric air-to-fuel ratio and ISO conditions (i.e. 100 kPa, 25°C, 30% relative air humidity). PG engine maintains the same design of both the combustion chambers and the pistons, so that the CR equals the original value (12.5). Instead, the spark timing was changed from 16° BTDC (standard value for the NG operation) to 28° BTDC. Further engine data are listed in [9].

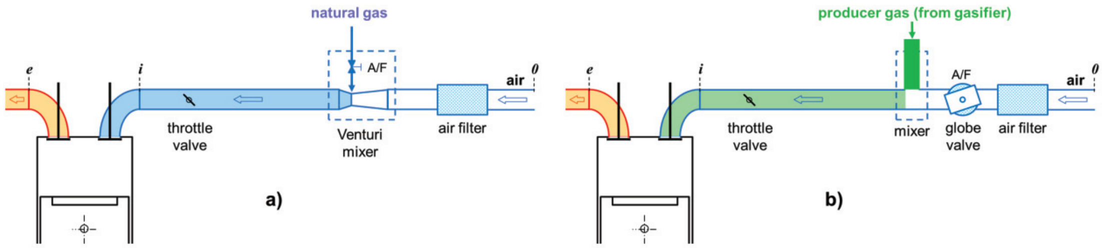

Fig. 2. Schematics of the intake systems. (a) Natural gas (NG); (b) Producer gas (PG).

4. Experimental tests

Tests were performed using the standard control system to measure the electric power (P_{el}), the PG temperature at the engine container inlet and the position of the air globe valve. Spark ignition timing was regulated manually acting on the spark ignition control unit, which however did not allow to specify preset values. Therefore, actual spark timing was measured by stroboscopic timing light. The portable flue gas analyser Messtechnik EHEIM GmbH Visit 01 LR was used for measuring O_2, CO and NO_x

concentrations in the exhaust gases just downstream of cylinders. The analyser includes infrared sensors able to detect volume concentrations of O_2 from 0% to 21%, CO from 0 to 10000 ppm and NO_x from 0 to 4000 ppm.

In a first series of tests, electrical power and engine emissions were measured at different air-to-fuel ratios maintaining the spark timing θ equal to 28° BTDC. In a second series of tests, spark timing was fixed at 23° and 29° whereas the engine burned fuel-air mixtures close to stoichiometric. All tests were performed at full throttle, constant gasification conditions, and constant engine speed (1500 rpm).

5. Engine performance calculation

Fuel conversion efficiency for a 4-stroke engine is given by the following expression [10]:

$$\eta_f = \frac{P}{\left(\rho_{a,0} \cdot V \cdot \frac{n}{120}\right) \cdot \frac{H_u}{A/F} \cdot \eta_v} \tag{1}$$

where P is the engine brake power, $\rho_{a,0}$ is reference air mass density, V is the total engine displacement, n is the engine speed, H_u is the fuel low heating value, A/F is the actual air-to-fuel ratio and η_v is the volumetric efficiency. In Eq. (1), $P=P_{el}/\eta_{el}$ (with η_{el} electrical generator efficiency equal to 0.95 in the present case) and $\rho_{a,0}$ is equal to 1.205 kg/m^3 for dry air (ambient temperature and pressure during tests were 20 °C and 101400 kPa, respectively). Hence, to calculate η_f, the missing parameters H_u, A/F and η_v must be evaluated for the PG operation:producer gas properties can be calculated once the gas composition is known, whereas the volumetric efficiency can be estimated from the full-load NG value as explained in the following.

5.1 Calculation of the PG composition

PG compositions was calculated by modelling the gasification process using an equilibrium model based on the minimization of the Gibbs free energy [11]. The model allows to calculate molar fractions of the PG components, once the composition and the humidity of the biomass, the temperature of the reaction zone, the oxidizer mass flow rate and the chemical species included in the PG are known. Producer gas it is assumed to be composed by hydrogen, carbon monoxide, methane, carbon dioxide, nitrogen and water. The values of the remaining parameters are known from experimental data. The resulting PG composition (18% H_2, 20% CO, 1% CH_4, 12% CO_2, 49% N_2) complies with experimental compositions available in the literature [1,3,5]. The corresponding low heating value and stoichiometric air-to-fuel ratio are equal to 4300 kJ/kg and 1.155, respectively.

5.2 Calculation of the volumetric efficiency

Compared to NG operation, PG-engine volumetric efficiency is primarily affected by the different fuel chemical composition, fuel-to-air ratio and volume displaced by the fuel [12]. Therefore, supposing that the nature and the amplitude of both the unsteady and quasi-steady phenomena do not change, the following expression can be used to correct the value of the NG-engine volumetric efficiency:

$$\frac{\eta_{v,PG}}{\eta_{v,NG}}=\frac{\dfrac{M_{c,PG}\,p_{i,PG}}{T_{i,PG}}\,\dfrac{1}{\left[1+(F/A)_{PG}\right]}}{\dfrac{M_{c,NG}\,p_{i,NG}}{T_{i,NG}}\,\dfrac{1}{\left[1+(F/A)_{NG}\right]}} \tag{2}$$

where M_c is the molecular weight of the cycle working fluid, p_i and T_i are the intake pressure and temperature, respectively, and *F/A* is the actual fuel-to-air ratio. In the expression, subscripts *PG* and *NG* refer to PG and NG operation, respectively. Moreover, it is assumed that the effect of the residual gases on the thermodynamic state of the mass entering the cylinders is the same for NG and PG operation.

At full load and stoichiometric fuelling (λ=1, *A/F*=17.23), $\eta_{v,NG}$ is equal to 0.72 in ISO operating conditions [9] and to 0.714 in actual operating conditions. $M_{c,NG}$ is calculated as mass weighted average of the working fluid components and results 28.24 kg/kmol. $T_{i,NG}$ can be calculated once estimated the flow preheating through the intake system, here assumed to be equal to 45°C.

Considering the parameters related to PG operation at right hand side of Eq. (2), $M_{c,PG}$, can be calculated using the same approach adopted for $M_{c,NG}$, once the actual air-to-fuel ratios at the different PG operating conditions are known. These ratios can be calculated by using simple stoichiometry relations from the measured oxygen concentrations in the exhaust gases. Mixture intake temperature$T_{i,NG}$ is estimated by applying the energy balance equation to the mixer and assuming the same flow preheating through the intake system of NG operation. The ratio $p_{i,PG}/p_{i,NG}$ can be calculated by considering that, in the PG intake system, the PG-air mixer and the air globe valve replace the NG-air Venturi mixer. In the present case, pressure losses through the PG-air mixer-globe valve system is evaluated to be triple compared to losses through the NG-air Venturi mixer. The dependence of pressure losses on the air-to-fuel ratio (that is on the air valve position) is neglected because, in the present case, the other effects are by far more relevant.

6. Results

Engine brake power and volumetric efficiency for PG operation are listed in Tab. 1. The first series of tests (from #2 to #4) were performed at fixed spark timing to investigate the effect of λ on engine performance. In the last two tests (#5 and #6) spark timing was varied keeping the air-to-fuel ratio close to stoichiometric value ($\lambda\approx1$). Test #1 is taken as reference for all the other tests.

Table 1. Results of experimental tests and calculations.

Test	λ	Spark timing, θ [° BTDC]	Brake power, P_{PG} [kW]	$\eta_{v,PG}$
#1 (nominal)	1.047	28°	116.8	0.313
#2	1.218	28°	103.2	0.339
#3	1.415	28°	81.1	0.366
#4	1.661	28°	64.2	0.394
#5	1.023	23°	114.7	0.309
#6	1.064	29°	120	0.316

Compared to the actual nominal NG brake power (254.8 kW), power de-rating in PG operation is close to 55%. This value is in agreement with de-ratings specified in other studies (see Section 2). The strong reduction of engine power is mainly caused by the drop of volumetric efficiency, which is about halved compered to NG fuelling. It can be noted also a significant power decrease as the air-to-fuel ratio increases, and a slight power gain by advancing the spark timing.

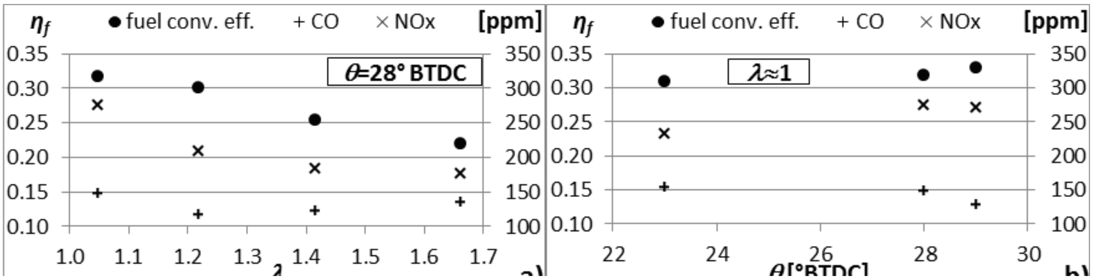

Fig. 3. (a) Fuel conversion efficiency and CO and NO_x emissions as a function of λ; (b) Fuel conversion efficiency and CO and NO_x emissions as a function of spark timing θ. Concentrations in part per million dry @5%O_2.

Fuel conversion efficiency and emissions at variable air-to-fuel ratios and spark timings are plotted in Fig. 3a) and 3b), respectively. As found by other authors, engine shows smooth operation under lean conditions, although the η_f drops when λ exceeds 1.3 due to low flame speed. Emissions of CO and NO_x are by far lower than the maximum emissions declared by the manufacturer for the NG operation (3200 and 3250 ppmd@5%O_2, respectively) in all the operating conditions. NO_x concentrations reduce as λ increase because of the reduction in flame temperature. A similar trend was found also by other authors [1,8]. CO emissions are very low compared to other literature data [8], but show a dependence on the excess air similar to that presented in [7].

A reduction of θ causes a slight decrease of engine efficiency and NO_x emissions. These results are reasonable, although in [8] the author found quite different trend for NO_x concentrations at λ=1.2. Instead, the weak dependence of CO emissions on spark timing agrees with results presented in [8].

7. Conclusions

In the paper performance and emissions of a heavy-duty PG-fuelled engine are analysed and compared to other data found in the literature related to SI engines. Power de-rating during PG operation exceeds 50% because of the significant reduction of the volumetric efficiency compared to NG operation. Instead, the reduction of fuel conversion efficiency is less relevant (about 17% in nominal conditions). Exhaust gas analysis confirms what found by other authors, that is PG fuelling contributes to reduce both CO and NO_x emissions in all the operating conditions.

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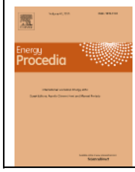

Biography

Paolo Gobbato is research fellow on biofuels-fuelled internal combustion engines since 2014. He earned PhD degree in Energetics in 2010 and he was research fellow from 2010 to 2014 at the University of Padova on design and modelling of combustion systems.