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Combustion modelling of a dual fuel diesel – producer gas compression ignition engine

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Abstract

A thermodynamic tool for a dual fuel engine combustion simulation is presented in this work. The modelling represents an initial approach which aims at defining the performance of a single-cylinder diesel engine modified to work under dual fuel conditions and fuelled with producer gas as a gaseous fuel. The tool is based on the application of the first law of thermodynamics to closed systems. A novel approach using a triple Wiebe function has been implemented to describe and predict the heat release rate of dual fuel combustion; ignition delay has also been modelled with a methodology derived from the application of Prakash delay time model. Validation of the proposed combustion model has been realized according to experimental data already present in the recent scientific literature production. The calibrated model has been applied to the UNIBZ energy system set-up in order to predict its main performance characteristics; a future experimental campaign will be carried out to validate the predicted trends.

Keywords: Internal combustion engine; producer gas; combustion model; delay time; dual-fuel engine

1. Introduction

In the actual scenario, fossil fuels consumption still prevails in the whole energy market [1]. However, the uncertainty in future supply, the high costs of expanding proven reserves of fossil fuels and the strongly unstable trend

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of crude oil prices during last 50 years have lead many energy analysts and scientists to seek alternative energy sources, such as renewables. Biomass generated producer gas represents a viable and consolidated alternative to traditional fossil fuels where there is a large availability of this primary source: its use in traditional engines is already solidly established through spark ignition internal combustion engines. On the other hand, the coupling of such a gas together with a pilot fuel to realize a dual fuel compression ignition engine is still facing scientific investigations on both modelling and experimental sides [2]. The use of a dual fuel engine grants a much higher level of flexibility, as the engine is able to operate at different rates of diesel fuel and producer gas [3]. Moreover, the performance loss experienced with simple Otto engines is limited with a dual-fuel operation mode.

At present, there are limited experimental test activities on dual-fuel producer gas engines and there are little data on the possible modelling approach to predict the combustion characteristics of this kind of engines.

1.1. Dual-Fuel Combustion in Internal Combustion Engines

The use of producer gas as a fuel for Internal Combustion Engines (ICEs) is one of the main path in which energy production from biomass generated gas is developing, together with micro-turbines and Stirling engines technologies. Thanks to their diffusion and their availability on the market, studies on both Spark Ignition (SI) and Compression Ignition (CI) engines are worldwide growing up in interests.

Since the average quality of producer gas as a fuel is significantly poorer than gasoline and natural gas, ICEs require design modifications in order to be able to run on producer gas [4]. Spark Ignition (SI) engines require very little modifications to run on producer gas: it is their main advantage, and it justifies their diffusion as producer gas fuelled engines. However, they reveal a relevant output power derating: since producer gas (and syngas as well) is a low-energy-density fuel and SI engines have low compression ratio (in the range of 8 to 12), power degradation is extensive. It is possible to quantify the power derating in a SI engine in as much as $40 - 50\%$; about 30% of the power loss is due to producer gas low energy density while the rest is accounted by the pressure drop in the intake valves and piping. Engine modifications are mainly dealing with the ignition timing: generally depending upon the compression ratio and the engine rotational speed, the ignition timing has to be advanced by about $30 - 40$ degrees. This is done because of producer gas low flame speed [4].

Compression Ignition (CI) engines operate more efficiently with producer gas: because of their high compression ratio and low rotational speeds, the derating of diesel engines running on producer gas is less than 30%. However, CI engines cannot be used without taking into account a proper strategy to start the combustion, since the temperature at the end of the compression stroke might be lower than producer gas auto-ignition one, depending on the composition of the gaseous fuel. Therefore, one of the strategies is to operate the engine in dual-fuel mode [4].

Thanks to their flexibility dual-fuel engines have been employed in a wide range of applications. Their use for alternative fuels driven power generation is attracting an increasing attention from scientists and renewable energies experts. Benefits with the dual-fuel conversion include smoother and quieter operation, fuel savings and the possibility to achieve very low emission levels, particularly in terms of smoke and particulate [5].

Modelling of the combustion process of a dual fuel engine operating with producer gas presents some peculiarities that require a deeper insight of the heat release rate as well as abnormal combustion. In dual fuel combustion, the gaseous primary fuel (biomass generated producer gas in our case), that is introduced along with air, provides most of the energy input. A minor amount of diesel is injected in order for the combustion to start: its self-ignition allows the propagation of the flame front in the air-producer gas premixed mixture. After the fuel injection inside the chamber, combustion mechanism in a dual-fuel diesel engine can be described using four different stages: an ignition delay time, that takes place between the fuel injection and the start of combustion; premixed combustion of the pilot fuel; premixed combustion of the gaseous fuel; diffusion combustion of the remaining pilot fuel together with the gaseous one [4]. Literature experimental data [6] dealing with these dual fuel combustion characteristics have been chosen for their pertinence in terms of the type of engines used and will be later used as case studies. The aim of the work is to define proper modelling approaches and modelling parameters to assess the performance of the engine as a function of the diesel fuel substitution rate.

2. Dual-fuel engine modelling strategy

A thermodynamics model – time dependence based, no spatial resolution is considered - has been realized in order to predict combustion features and simulate the heat release rate and the work cycle of a small-scale ICE operating in dual-fuel mode. This modeling represents an initial approach which aims at defining the performance of a water-cooled, four strokes, single-cylinder diesel engine modified to work under dual fuel conditions. The engine is an auxiliary power generator and it is connected to an electric generator. The present work has been carried out according to the following approach: i) Development and implementation of the mathematical model; ii) model calibration based on literature data; iii) application of the model to the UNIBZ experimental set-up case.

2.1. Development of the model

The mathematical models have been implemented in a Matlab environment. The tool is based on the application of the first law of thermodynamics for variable-volume closed system to a homogeneous mixture of ideal gases. For a closed system in a crank angle domain, such as the combustion chamber, we can write:

$$\frac{dP}{d\theta} = \frac{k-1}{V}\left(\frac{dQ_{in}}{d\theta} - \frac{dQ_{out}}{d\theta}\right) - k\frac{P}{V}\frac{dV}{d\theta} + \frac{P}{k-1}\frac{dk}{d\theta} \tag{1}$$

Where $dQ_{in}/\,d\theta$ represents the heat release and $dQ_{out}/\,d\theta$ the heat losses.

Heat losses through the external walls of the combustion chamber have been taken into consideration using the instantaneous cylinder average heat transfer coefficient modelled by Woschni. This correlation introduces a characteristic gas velocity that, during combustion and expansion sums up the contributes of both piston speed and pressure rise terms [7,8]. The heat transfer rate at any crank angle can be so calculated:

$$\frac{dQ}{d\theta} = am_p\frac{Q_p}{\theta_p}\left(\frac{\theta}{\theta_p}\right)^{m_p-1} e^{\left[-a\left(\frac{\theta}{\theta_p}\right)^{m_p}\right]} + am_d\frac{Q_d}{\theta_d}\left(\frac{\theta}{\theta_d}\right)^{m_d-1} e^{\left[-a\left(\frac{\theta}{\theta_d}\right)^{m_d}\right]} \tag{2}$$

Where:

$$h_g = 3.26P^{0.8}U^{0.8}b^{-0.2}T^{-0.55} \tag{3}$$

$$U = 2.28\overline{U}_p + 0.00324T_r\frac{V_d}{V_r}\frac{P-P_m}{P_r} \tag{4}$$

$\overline{U}_p$ is the mean piston speed (m/s); T_r is the temperature at the intake valve closing (k); V_r is the cylinder volume at intake valve closing (m^3); P_m is the motored pressure (kPa). Mean wall temperature $\overline{T_w}$ has been considered equal to 420K as suggested by C.D. Rakopoulos and E.G. Giakoumis [9].

Heat release has been computed with a Wiebe modelling approach, taking into account the contributes of both diesel and producer gas as a superimposition of two different combustion phenomena after the initial pilot fuel premixed combustion peak. Diesel combustion heat release has been modelled using a combined double Wiebe function which is typically adopted to model the heat release rate in a diesel engine. Double Wiebe function can be expressed as follows [8]:

$$\frac{dQ}{d\theta} = am_p\frac{Q_p}{\theta_p}\left(\frac{\theta}{\theta_p}\right)^{m_p-1} e^{\left[-a\left(\frac{\theta}{\theta_p}\right)^{m_p}\right]} + am_d\frac{Q_d}{\theta_d}\left(\frac{\theta}{\theta_d}\right)^{m_d-1} e^{\left[-a\left(\frac{\theta}{\theta_d}\right)^{m_d}\right]} \tag{5}$$

The subscripts p and d refer to premixed and diffused combustion; a is a non-dimensional constant; θ_p and θ_d are the burning duration for each phase; m_p and m_d are non-dimensional shape factor for each phase.

Producer gas combustion has been modelled taking again into consideration the physical nature of the phenomenon: the charge is a mixture of air and syngas, both coming from the intake manifold. The heat release due to producer gas combustion can be calculated considering a third Wiebe function describing a premixed combustion.

Ignition delay of the pilot fuel in a dual fuel engine is a crucial characteristic and has a relevant effect on combustion, performance and exhaust emission. Long ignition delays may result in poor thermal efficiency and high levels of exhaust emissions of hydrocarbons and carbon monoxide [10]. The estimation of ignition delay has followed Prakash and Ramesh [10] approach. Prakash proposes a modification of the Hardenberg & Hase correlation for ignition delay in diesel engines to better model the dual fuel case by bringing in to effect the changes in the temperature at the end of compression and the oxygen concentration in the charge [10]. Prakash model, tested and validated on single-cylinder, direct injection, air-cooled diesel engine fuelled with biogas as gaseous fuel, has been chosen for its relevance and pertinence in terms of dual fuel combustion. Ignition delay τid can be computed as:

$$\tau_{id} = A \cdot C_f \cdot O_c^k \cdot e^{(E \cdot P + Q^{0.63})} \tag{6}$$

Where: $A = \left(0.36 + 0.22 M_{ps}\right);\ E = \frac{618840}{CN+25};\ P = \frac{1}{R_u \cdot T_{BDC} \cdot C_r^{n_{df}-1}} - \frac{1}{17190};\ Q = \frac{21.2}{P_m \cdot C_r^{n_{df}} - 12.4}$.

P_m is the manifold pressure (bar); M_{ps} is the mean piston speed (m/s); $C_r$is the compression ratio and E is the activation energy given as a function of the cetane number CN [10]. The polytropic index for the dual fuel case, n_{df}, is calculated starting from the diesel polytropic index n_d and the gas concentration f_p, as it follows [9]:

$$n_{df} = n_d - 023 f_p \tag{7}$$

The core of Prakash modification is the introduction of the oxygen concentration ratio O_c: when the low LHV gaseous fuel is mixed in the inlet charge, the concentration of oxygen in the mixture falls, leading to an increase in the ignition delay period of the pilot fuel [10]. O_c is defined as the ratio of concentration of oxygen in the charge present in the cylinder to the concentration of oxygen present in atmospheric air; oxygen concentration in the charge can be expressed as follows:

$$[O] = \frac{N_a}{4.76 N_{mix}} \tag{8}$$

where N_{mix} is the sum of the moles of air, gas and retained exhaust in the charge [10]. Finally, the value of the constant k has been obtained by the method of least squares from the experimental data, as suggested by Prakash [10].

Ignition delay has been studied as a function of the diesel substitution rate d_s, a relevant parameter in dual fuel combustion engines that express the amount of diesel saved thanks to the introduction of the gaseous fuel:

$$d_s = \frac{d_d - d_{df}}{d_d} * 100 \tag{9}$$

where d_d is the amount of diesel used in pure diesel condition, while d_{df} is the quantity of diesel injected as a pilot fuel in dual-fuel mode.

2.2. Model calibration

The calibration of the model has been carried out using data available in literature (fig. 1), obtained by Sambatwong et al. [6]. A single cylinder diesel engine (Mitsubishi D-800) has been tested under different dual fuel conditions using biomass derived producer gas (CO 32.3%, H2 4.2%, N2 62%).

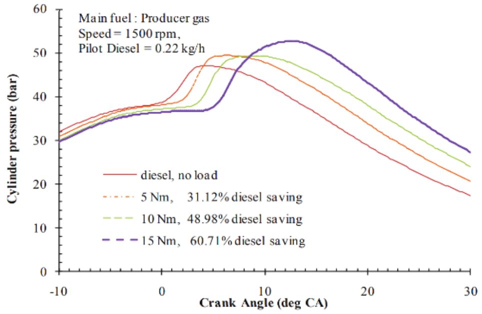

Fig. 1 – Cylinder pressure trend for different substitution rates [6]

In-cylinder pressure modeling has been calibrated based on the the 60.71% diesel saving results (fig. 1). Wiebe functions coefficients have been chosen in order to minimize the error between the experimental and the calculated values; the following coefficients have been used: ap=6, mp=1.9, ad=3.7, md=1.1, asyn=2, msyn=1.4 (where the indices p, d and syn respectively indicate diesel premixed, diesel diffused and syngas premixed combustion related Wiebe coefficients). Fig. 2a shows in-cylinder pressure prediction of the experimental values, which are overestimated by the model by only 3.7%. Prakash ignition delay model has been calibrated on the relative delay time data available

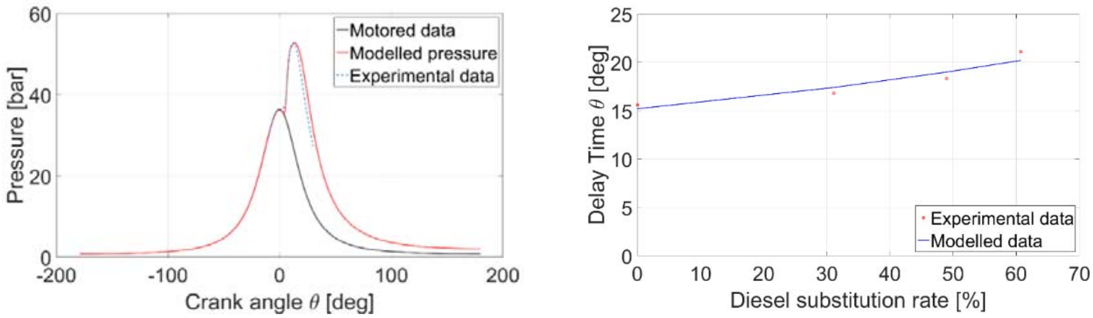

Fig. 2 – Cylinder pressure (a) and ignition delay time (b) model calibration

from the experimental tests. As shown in fig. 2b the model well estimates the ignition delay trend with a coefficient of determination R^2 equal to 0.89.

3. Results

The calibrated model has been applied to the energy system available at the Free University of Bozen/Bolzano, made up of a *Farymann 15W430* diesel engine, whose main characteristics are shown in Table 1, and an open top gasifier, whose producer gas composition is made up of: 16.78% CO, 1.15% H_2, 1.29% CH_4, 24.33% CO_2, 56.45% N_2.

Table 1. Farymann 15W430 main technical data

Displacement [cm3]	242	Specific fuel consumption @3000 rpm [g/kWh]	305
Compression ratio	20	Bore x Stroke [mm]	75 x 55
Max electric power @3000 rpm [kW]	3.7	Diesel fuel injection pressure [bar]	200

Pressure trend is predicted and shown in fig. 3a for a diesel substitution rate of 70%. Other substitution rates have been simulated in terms of P-V diagrams, as shown in fig. 3b.

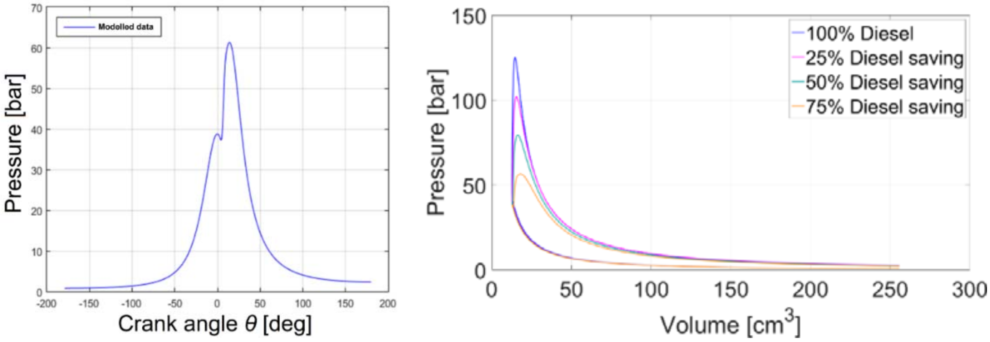

Fig. 3 – Cylinder pressure trend prediction and engine p-V work cycle

Fuel substitution is responsible of a power derating, in terms of gross mechanical power,that reaches peaks of 23% for substitution rates (Ds) around 60%, as shown in table 2.

Table 2. Output gross mechanical power derating due to diesel substitution.

Ds [%]	Pw [kW]	Power derating [%]
0.00	6.71	0.00
31.12	6.31	5.96
48.98	5.68	15.35
60.71	5.17	22.95

Results (Fig.3; table 2) of the application of the proposed model to the compression ignition engine hosted in the laboratory of the Free University of Bozen/Bolzano, permit to predict engine performances when the dual-fuel combustion mode is set.

4. Conclusions

Based on a thermodynamics approach, a 0D model has been developed to study and predict main combustion features of dual-fuel combustion in diesel engines. The tool, realized in a MATLAB environment, is based on the application of the first law of thermodynamics to closed system. Preliminary results, based on experimental data available in literature, show that the proposed approach can be used to predict in-cylinder pressure trends and output mechanical power for different diesel substitution rates. Further investigations and studies will be carried out based on an experimental campaign in the laboratories of the Free University of Bozen/Bolzano.

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