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Prediction of small spark ignited engine performance using producer gas as fuel

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ABSTRACT

Producer gas from biomass gasification is expected to contribute to greater energy mix in the future. Therefore, effect of producer gas on engine performance is of great interest. Evaluation of engine performances can be hard and costly. Ideally, they may be predicted mathematically. This work was to apply mathematical models in evaluating performance of a small producer gas engine. The engine was a spark ignition, single cylinder unit with a CR of 14:1. Simulation was carried out on full load and varying engine speeds. From simulated results, it was found that the simple mathematical model can predict the performance of the gas engine and gave good agreement with experimental results. The differences were within $\pm 7\%$.

1. Introduction

Producer gas was derived from biomass via gasification with average calorific value of about 5 MJ/Nm^3 [1]. Presently, the use of 100% producer gas in spark ignition (SI) engine was not successful, because producer gas has low energy density, hence, low power output and efficiency [2]. Recently, increasing performance of producer gas engine can be done by increasing compression ratio (CR), changing combustion chamber, mounting gas carburetor and modifying the ignition system [3,4]. Experimental evaluation of a producer gas engine can be costly, complicated and time consuming. Ideally, the engine performance may be predicted using mathematical equations [5]. Establishing mathematical models is of interest. In this work, a single zone cylinder model was used. It can provide quick calculation of optimum conditions. Examination of various engine performance parameters may be achieved [6,7]. The basic assumption of the single zone cylinder model was based on mass balance analysis, regardless of chemical reaction, homogeneous charges, and mixing of gases inside the cylinder [8]. Therefore, the objective of this work was to study the use of mathematical model in small producer gas engine comparing with experimental in term torque, brake power, thermal efficiency and specific fuel consumption.

2. Mathematical modeling

The model was combined with physical based equations for describing phenomena and performance of the small producer gas engine. The details of the mathematical models are as follows:

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2.1. Cylinder pressure

The pressure in cylinder of SI engine can be derived from the first law analysis. The cylinder pressure versus crank angle is shown in Eq. (1) [9].

$$\frac{dP}{d\theta} = \frac{k-1}{V}\frac{dQ}{d\theta} - k\frac{P}{V}\frac{dV}{d\theta} \tag{1}$$

where, P is the pressure inside cylinder, θ is crank angle, k is specific heat ratio, Q is heat releases, V is the cylinder volume and as a function of crank angle, given as

$$V(\theta) = \frac{V_d}{r_c - 1} + \frac{V_d}{2}\left[\frac{l}{a} + 1 - \cos\theta - \left(\left(\frac{l}{a}\right)^2 - \sin^2\theta\right)^{0.5}\right] \tag{2}$$

where, V_d is displacement volume, r_c is compression ratio, l is connecting rod length, a is crank radius.

2.2. Heat input

The total amount of heat input to cylinder versus changes in the crank angle is shown in Eq. 3 [10].

$$\frac{\partial Q}{d\theta} = HV\int_{IVO}^{IVC} m^{\bullet}\, d\theta\frac{df}{d\theta} \tag{3}$$

where, HV is heating value, $m^{\bullet}$ is producer gas flow rate, IVO and IVC are inlet valve open and close positions before and after TDC, $f(\theta)$ is the Wiebe function. Producer gas flow rate through an intake valve was derived empirically from the engine test run between 1100–1900 rpm of engine speed. It is given as

$$m^{\bullet} = 0.00378V_d(0.105N^2 - 0.7922N - 0.0015N^3) \tag{4}$$

where, N is engine speeds and the Wiebe function is used to determine the combustion rate of the fuel, expressed as [11]:

$$f(\theta) = 1 - \exp\left[-5\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^3\right] \tag{5}$$

where, θ is crank angle, θ_0 is start of heat release angle, $\Delta\theta$ is duration of heat release and can be determined from this equation.

$$\Delta\theta = -1.618\left(\frac{N}{1000}\right)^2 + 19.866\left(\frac{N}{1000}\right) + 39.395 \tag{6}$$

2.3. Heat transfer

The heat transfer is necessary for the internal combustion engine to maintain cylinder walls, pistons and piston rings. Normally, the heat transfer in the combustion engine includes conduction, convection and radiation [12]. However, for an SI engine, the primary heat transfer mechanism from the cylinder gases to the wall is convection, with only 5% from radiation [13]. The heat loss to the wall can be determined from the Newtonian convection equation [14] which is given as

$$Q_{loss} = hA(T_g - T_W) \tag{7}$$

where, h is heat transfer coefficient, A is surface area of combustion chamber T_g is gas temperature in cylinder, T_W is cylinder wall temperature. The heat transfer coefficient is instantaneous area average heat transfer coefficient derived from Woschni [15], shown in Eq. (8).

$$h = 0.82b^{-0.2}(P10^{-3}c)0.8T_g^{-0.53} \tag{8}$$

where, b is bore cylinder, c is equal to 6.18. The gas temperature is calculated using following equation from Sitthiracha [16] while, engine speed is in a range of 1000–6000 rpm.

$$T_g = 3.395\left(\frac{N}{1000}\right)^3 - 51.9\left(\frac{N}{1000}\right)^2 + 279.49\left(\frac{N}{1000}\right) + 676.21 \tag{9}$$

Calculation of surface area in cylinder is from the following equation [9] which includes cylinder head, cylinder bore and piston crown. Surface area at any crank angle is given as: