PERFORMANCE ANALYSIS OF A BIOGAS ENGINE

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Abstract

A single zone S.I. engine model based on the physical geometries (bore, stroke, crank radius, and connecting rod length), variation of specific heats of gas mixture (air-fuel mixture and combustion products) and chemical equilibrium has been developed in an in-house code to predict the pressure and temperature variation along crank angle, work done, mean effective pressure, power output and other engine parameters suitable for biogas. The intake model has also been incorporated in the developed model. Model has been validated using biogas (methane). Effect of spark timing in spark ignition model have been observed and optimized to obtain maximum work as well as brake mean effective pressure with biogas. It has been observed that at 27° BTDC spark timing give the best performance. The spark ignition model at optimized condition, the brake thermal efficiency is calculated as 24% and brake mean specific fuel consumption is 0.29 m³/kWh.

***Key Words**: Biogas; Biogas Engine; Engine Model, Engine Performance.*

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1. INTRODUCTION

In modern days, there is a need of alternative renewable automotive fuels other than conventional fuel because the conventional fossil fuels are depleting in nature and there is a rapidly rise in price in the international market. Moreover, gaseous fuels are getting more importance other than the liquid fuel due to its clean burning. Gaseous fuel has high hydrogen to carbon ratio which also leads to low carbon dioxide emission. LPG, CNG are petroleum based gaseous fuel are not renewable energy but hydrogen, biogas, producer gas derived from biomass are renewable in nature. Energy from biomass will be a reliable energy option where there is lack of conventional sources and located far away from the grid and abundant availability of biomass other than wood biomass [1]. According to 'National Dairy Development Board India' India had 199.1 million cattle in the year of 2007 that leads to a high potential source of renewable biogas [2]. Lime water scrubber is used to reduce the carbon dioxide and hydrogen sulphide (H_2S) level in biogas [3]. Researchers have been spending essential effort to improve the performance of SI engines. They have been practiced on both empirical and theoretical methods. A realistic simulation model could reduce the enormous effort, money and time required to perform an experiment. Furthermore, the mathematical models enhanced the understanding of the flow field, fuel-air mixing, and combustion in the engines. These models can be classified into two groups: thermodynamic and dimensional models. Thermodynamic models can in turn be divided into single and multi-zone models. Researchers developed a theoretical model of Otto cycle, with a working fluid consisting of various gas mixtures having temperature dependent specific heats [5,6]. It is being found that it is more realistic to use the gas mixture model instead of air as the working fluid for the analysis of spark ignition engines. Rakopoulos *et al.*[7] provided the availability analysis to the cylinder of a spark ignition engine with biogas–hydrogen blends. It is revealed that the addition of increasing amounts of hydrogen in biogas promotes the degree of reversibility. Ghazal [8] evaluated the effect of using blend hydrogen-methane fuel with varying fuel induction systems for spark ignition engines on its overall performance and emission. It is being found that there is a general reduction in CO with the addition of hydrogen. In the present work, a single zone thermodynamic model of a S.I. engine has been developed and validated with methane. Single zone model considers that the mixture composition, pressure, and temperature are uniform throughout the combustion chamber, the working fluid has been considered as a gas mixture, with temperature dependent specific heats. The model is incorporated with the intake and exhaust stroke in the developed model without considering the gas mixture model to have a better prediction of the performance parameters. Heat loss from the cycle is also considered as a percentage from the engine [9]. The effect of spark timing on various engine performance parameters such as brake work done/cycle, brake thermal efficiency, brake specific fuel consumption has been observed and optimized.

1.1 Model Descriptions

Model has been developed considering single zone using in-house code and thermodynamic parameters have been calculated considering the properties each species present in the cylinder during the cycle considering following assumptions

- The cylinder charge (air-fuel mixture) and combustion product mixture are assumed as homogeneous and ideal gas.
- The compression and expansion stroke of engine cycle have been considered neglecting pumping work during suction and exhaust stroke.

- Chemical equilibrium exists among the gaseous species inside the combustion chamber.
- The specific heats of the gaseous species vary with the local temperature.
- Variations in specific heats have been considered with the change in crank rotation due to change in temperature and composition.
- Heat added to the cycle based on the mass fraction of the fuel burnt as a function of crank angle using Weibe function.
- There is no effect of combustion chamber design.
- Initial pressure at compression stroke is assumed as atmospheric pressure.
- Heat loss from the cycle is considered as a percentage heat leakage from the engine [9].

The relationship between the cylinder volume and the crank angle is the function of compression ratio, bore, stroke and connecting rod length. Combustion model has been developed based on the conservation principle applied to a combustion chamber as the control volume. The relation from ideal gas law and its differential form,

$$PV = mR_gT \quad \text{and} \quad \frac{dV}{V} + \frac{dP}{P} = \frac{dT}{T} \tag{1}$$

The relation from conservation of energy can be expressed in differential form and applied to a control volume. The equation becomes for net heat release rate is

$$\frac{dQ_n}{d\theta} = mc_v\frac{dT}{d\theta} + P\frac{dV}{d\theta} \tag{2}$$

Rearranging equations (1 and 2)

$$\frac{dP}{d\theta} = \frac{k-1}{V}\frac{dQ_n}{d\theta} - \frac{kP}{V}\frac{dV}{d\theta} \tag{3}$$

The net heat term has both heat added and heat loss in case of combustion

$Q_n = Q_{in}x_b - Q_l$ Where, Q_{in} indicates total heat input and x_b as fraction of heat release, the heat loss can be obtained as,

$$\frac{dQ_l}{d\theta} = \frac{h_g(\theta)A_w(\theta)}{\omega}(T_w - T_g(\theta)) \tag{4}$$

According to Rakapoulos [10] the temperature of wall can be taken as constant. In the present work the ratio of temperature is taken as $T_w / T_1 = 1.2$

The mass fraction of fuel burned in an internal combustion engine can be expressed as a function of crank angle using the Wiebe function,

$$x_b(\theta) = 1 - \exp\{-a(\frac{\theta - \theta_s}{\Delta\theta})^n\} \tag{5}$$

Net heat input (Q_{in}) can be expressed as

$$Q_{in} = V_f \eta_v Q_{LHV} \tag{6}$$

Where, V_f is the volume of fuel inside the combustion chamber, η_v is the volumetric efficiency and Q_{LHV} is the lower heating value of fuel.

The final relation of combustion chamber pressure with the crank angle is obtained as

$$\frac{dP}{d\theta} = \frac{k-1}{V}[Q_{in}\frac{dx_b}{d\theta} - \frac{h_g(\theta)A_w(\theta)}{\omega}(T_w - T_g(\theta))] - \frac{kP}{V}\frac{dV}{d\theta} \tag{7}$$

The above equation is solved by using explicit finite difference technique with second – order accuracy as [5]. The pressure variation during the intake and exhaust stroke can be obtained as [11]:

$$\frac{dP}{dt} = kP(\frac{1}{M}\frac{dM}{dt} - \frac{1}{V}\frac{dV}{dt}) \tag{8}$$

The equation (9) requires mass flow rate dM/dt, which is to be found from the equations of fluid mechanics.For the intake stroke, mass flow rate can be obtained as :

$$\frac{dM_{in}}{dt} = AP\sqrt{\frac{2k}{RT(k-1)}(\frac{P_{in}}{P})^{\frac{k-1}{k}}\left[(\frac{P_{in}}{P})^{\frac{k-1}{k}} - 1\right]}$$

[When (Pin/P)<PRcric] (9)

$$\frac{dM_{in}}{dt} = AP\sqrt{\frac{k}{RT}(\frac{2}{k+1})^{(k+1)/(k-1)}}$$

[When(Pin/P)>PRcric] (10)

The critical pressure ratio that defines the two regimes is called PR_{crit} and depends only on the value of k of the gas.

$$PR_{crit} = (\frac{k+1}{2})^{\frac{k}{k-1}} \tag{11}$$

Where P_{in} the intake manifold pressure and P is is the cylinder pressure.

Similarly for the exhaust stroke there is a little change in the mass flow rate equation which is mentioned below.

$$\frac{dM_{ex}}{dt} = AP\sqrt{\frac{2k}{RT(k-1)}(\frac{P}{P_{ex}})^{\frac{k-1}{k}}\left[(\frac{P}{P_{ex}})^{\frac{k-1}{k}} - 1\right]}$$

[When (P/Pex)<PRcrit] (12)

$$\frac{dM_{ex}}{dt} = AP\sqrt{\frac{k}{RT}(\frac{2}{k+1})^{(k+1)/(k-1)}}$$

[When (P/Pex)>PRcrit] (13)

Where P_{ex} is the exhaust manifold pressure and P is the cylinder pressure.

Where valve area can be obtained as:

$$A = A_{in}(|\sin\theta|)^{1/3} \tag{14}$$

Where A_{in} is the wide-open valve area. In this model the value of A_{in} has been taken as 3.5 cm2 [11].

The mean effective losses of power due to friction in different moving parts are calculated by using the following empirical relations by Gogoi [12].

i) Mean effective pressure (MEP) lost due to friction in the piston and rings

$$FMEP1 = 12.85 * \frac{P_{sl}}{B*S} * \frac{100*Upe}{1000} \tag{15}$$

Where, P_{sl} is the piston skirt length (0.65 to 0.85 times of Bore) [13], B is the cylinder bore (m), S is the stroke length (m) and U_{pe} is the mean piston speed (m/s).

ii) MEP lost in bearing friction

$$FMEP2 = 0.0564 * \frac{B}{S} * \frac{N}{1000} \quad (16)$$

Where, N is the engine speed in rpm.

iii) Mean effective pressure (MEP) absorbed in overcome friction due to the valve gear

$$FMEP3 = 0.226 * \left(30 - \frac{4N}{1000}\right) * \frac{G * Dvi^{1.75}}{B^2 * S} \quad (17)$$

Where G is the number of intake valves per cylinder and D_{vi} is the diameter of the intake valve.

iv) MEP lost in pumping

$$FMEP4 = 0.0275 * \left(\frac{N}{1000}\right)^{1.5} \quad (18)$$

v) MEP lost in friction due to the wall tension of rings

$$FMEP5 = \frac{10 * 0.377 * S * n_{pr}}{B^2} \quad (19)$$

Where, n_{pr} is the number of rings.

In this model, two rings have been considered.

vi) MEP lost in overcoming combustion chamber and wall pumping losses

$$FMEP6 = \sqrt{\frac{p_{imep}}{11.45}} * 0.0915 * \left(\frac{N}{1000}\right)^{1.5} \quad (20)$$

Where p_{imep} is the indicated mean effective pressure.

The total mean effective pressure (MEP) lost in pressure can be calculated as,

$FMEP=(FMEP1+FMEP2+FMEP3+FMEP4+FMEP5+FMEP6)$

2. MODEL VALIDATION AND PERFORMANCE ANALYSIS

The model predicted results has been validated with the experimental and simulated results reported by Pourkhesalian et al. [14]. In this model the variation of specific heat is calculated using the following equation [5].

$$C_p = 2.506\times10^{-11}T_g^2 + 1.454\times10^{-7}T_g^{1.5} - 4.246\times10^{-7}T_g + 3.162\times10^{-5}T_g^{0.5} + 1.3303 - 1.512\times10^{4}T_g^{-1.5} + 3.063\times10^{5}T_g^{-2} - 2.212\times10^{7}T_g^{-3}$$

Where, Tg is the temperature of the gas inside the cylinder. The engine specification and operating parameters used in the validation is presented in the table 1.

2.1 Pressure-Crank Angle Curve Validation

Model Validation has been done with Methane (Fig. 1). Different value of parameters a and n of the Weibe function are chosen to fix pressure variation with crank angle.

Table -1: Engine specifications and operating parameters

Model Engine Specifications	
Fuels	Methane
Type of engine	Four stroke, four cylinder, spark ignition engine
Bore × Stroke	0.086m×0.086m
Connecting rod length	0.153m
Compression ratio	8.6

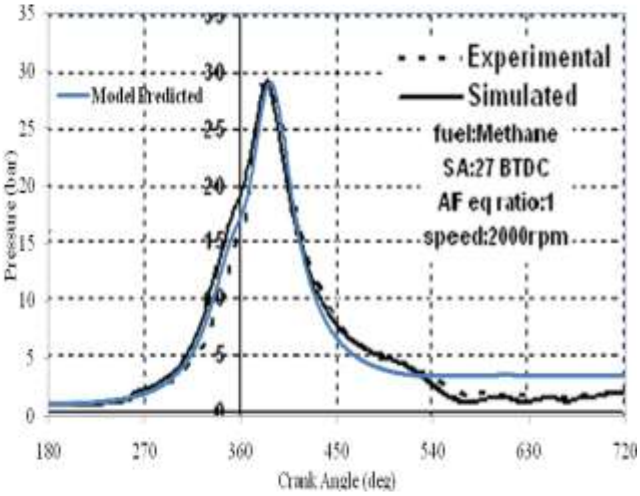

Fig-1: Variation of cylinder pressure with crank angle for methane

2.2 Effect of Variation in Spark Timing

The effect of spark timing in the spark ignition engine model is observed and investigated at validated condition considering the burning duration is same for all case (64° crank angle). Methane has been taken as fuel with equivalence ratio 1.0 and wall temperature is 415K. The value of constant parameters are taken as a=4.0, n=3.0 and C_1=0.18, C_2=0.00324 at power stroke and zero at compression stroke for the analysis for methane. Sparking is done near the end of compression stroke and before the power stroke begins. The spark timing has been varied from 0° BTDC to 35° BTDC to investigate the effect on engine performance. In figure-2, the effect of the variation in spark timing on brake work done per cycle and brake mean effective pressure (bmep) has been observed. It can be observed that at 0° spark advance BTDC work output is very less. When the spark is advanced, the work output from the engine is increased as more heat is released during power stroke. But as the spark advances, the peak pressure also increases which means the pressure at the end of the compression also increases. As a result, more negative work is done by the engine. So, an optimization is required to have maximum work done per cycle and maximum bmep.

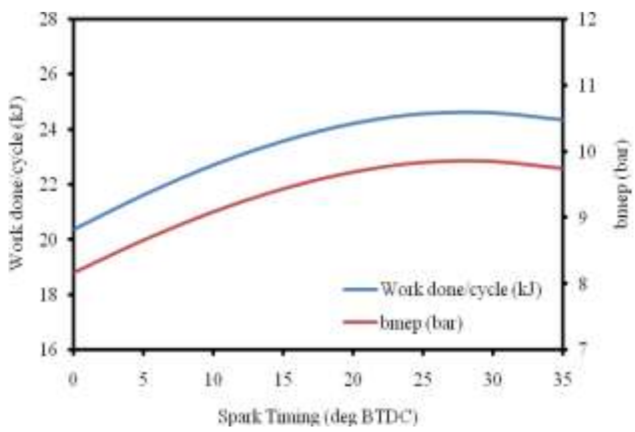

Fig -2: Effect of spark timing on brake work done/cycle and brake mean effective pressure

In figure. 3, the effect of variation of spark timing on brake thermal efficiency (BTE) and brake specific fuel consumption (bsfc) is observed. It shows the same trend as the effect of spark timing on brake work done/cycle and brake mean effective pressure as the fuel quantity does not change in a cycle with the change in spark timing.

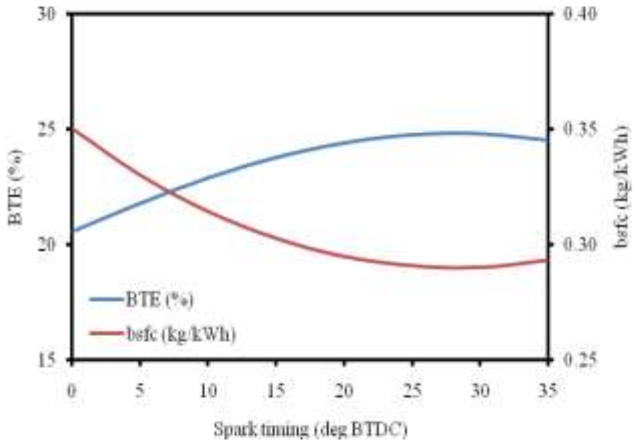

Fig- 3: Effect of spark timing on brake thermal efficiency (BTE)and brake specific fuel consumption (bsfc)

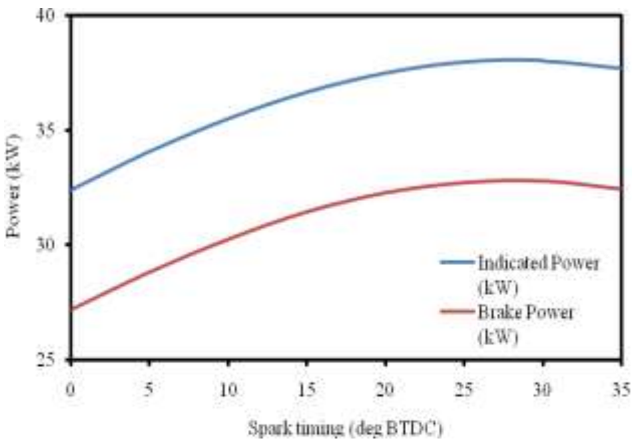

Fig -4: Effect of spark timing on indicated power and brake power

Figure 4, shows the effect of spark timing on indicated power and brake power. The brake power is directly proportional to the brake mean effective pressure, whereas brake mean effective pressure depends on brake work done/cycle. Hence it follows the same trend as brake work done per cycle. However, there is a constant difference between indicated power and brake power in form of frictional power due to frictional losses.

From the above performance analysis it has observed that the optimum spark timing which gives best result *i.e*, maximum brake power, brake thermal efficiency and minimum brake specific fuel consumption has been found at 27° BTDC for methane.

3. CONCLUSIONS

- A single zone S.I engine model based on physical geometries (bore, stroke, crank radius, and connecting rod length) has been developed in an inhouse FORTRAN code for biogas (methane).
- The biogas fired spark ignition model has been validated with experimental results from literature at 2000 rpm with biogas. The results from the model correspond well with the experimental results.
- Effect of spark timing in spark ignition model have been observed and optimized to obtain maximum work and brake mean effective pressure with biogas. It has been observed that at 27° BTDC spark timing give the best performance.
- The spark ignition model at optimized condition, the brake thermal efficiency is calculated as 24% and brake mean specific fuel consumption is 0.29 m^3/kWh.

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