DIGITAL SIMULATION OF PRODUCER GAS FIRED SI ENGINE

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Abstract—Engine emissions becoming stringent, gaseous fuels are gaining prominence as cleaner fuels like LPG and CNG both for stationary and automotive applications. But scarcity of those fuels arises big question for future. Producer gas obtained from biomass gasification can be good alternative for non-renewable fuels especially in developing country like India, where biomass is available in huge quantity. In the present study simulation based on actual thermodynamic cycle analysis is performed to assess the performance of 118 cc S.I. engine. All the four basic processes taking place in an S.I. engine are analyzed and the values of pressure and temperature at every 2° of crank rotation are found out with the aid of certain assumptions. The model involves good deal of calculations and iterations and hence, it is coded in 'c'. The simulation result is validated with the technical specifications provided in the technical manual of the engine. Digital simulation shows with producer gas as a fuel power and thermal efficiency relatively de-rated by 11.54 % and 5.72 % respectively, compared to gasoline.

Index Terms— Digital Simulation, Producer Gas Fuel.

NOMENCLATURES

ATDC After Top Dead Centre
BTDC Before Top Dead Centre
BMt Billion Metric tones
CA Crank Angle
cc Cubic Centimeters
C_h Convection Heat-Transfer
C_h Convective Heat-Transfer Co-Efficient
C_k Thermal Conductivity
C_m Mean Piston Velocity
CNG Compressed Natural Gas
CPG Compressed Producer Gas
CR Compression Ratio
D Cylinder Bore
GHG Greenhouse Gases
h Steady Turbulent Heat-Transfer
HC Hydro-Carbons
IA Ignition Advance
L Cylinder Stroke Length
LNG Liquefied Natural Gas
Mtoe Million Tons of Oil Equivalent
NO_X Nitruc Oxides
Nu Nusselt Number
Re Reynolds Number
S.I. Spark Ignition
V_s Displacement/Swept Volume
W_{mv} Mean Gas Velocity
μ_{cy} Viscosity of Cylinder Gas

I. INTRODUCTION

In the year 2013, India's net imports are nearly 144.3 million tons of crude oil, 16 Mtoe of LNG and 95 Mtoe coal totaling to 255.3 Mtoe of primary energy which is equal to 42.9% of total primary energy consumption [1]. India imports nearly 75% of its 4.3 million barrels per day crude oil needs but exports nearly 1.25 million barrels per day of refined petroleum products which is nearly 30% of its total production of refined products [2]. The growth of electricity generation in India has been hindered by domestic coal shortages and as a consequence, India's coal imports for electricity generation increased by 18 % in 2010. The electricity sector in India had an installed capacity of 249.488 GW as of end June 2014 [3]. World CO_2 (GHG) emission will grow from 31 BMt in 2010 to 45 BMt in 2040, which is mainly responsible for global warming. This data shows there is desperate need of new renewable energy resource. India has biomass capacity of 66000 MW [2] in form of resources such as rice husk, crop stalks, small wood chips, and other agro-residues. Biomass can be gasified in gasifiers and used as alternative fuel for current depleting fuels.

II. LITERATURE REVIEW

Gasoline has the fastest flame propagation development followed by LPG and CNG, CPG. CPG burns with blue flame compared to violent combustion of gasoline. Presence of H_2 causes fast burning rates initially, which slows down due to low burning speed of CO after H_2 has burnt [4]. CNG having higher efficiency and lower CO, CO_2, HC emission compared to gasoline and LPG but produces more NOx emission [5], [6]. But these fuels are on the verge of depletion. Shashikantha [7] reported that Producer gas with low energy density (5 MJ/kg) but reasonably high mixture energy density (2.12 MJ/kg) can replace these gases with almost same thermal efficiency of 28–32 %. but power derating of up to 30 %. Presence of Hydrogen does not give any pre-ignition problem due to gas being dilute. Producer Gas efficiency increases with increase in CR 30.7 % at CR of 17 and reduces to 27.4 % at CR of 10 [8]. Ignition advance should be retarded for increase in CR. Producer gas can be used up to CR of 17 without formation of knock [8], [9]. It reduces CO emissions considerably but NO_X and CO_2 will be increased. Hydrogen content in gas increases thermal efficiency but it is limited by process of gasification.

Dual-fuel mode operation requires less modifications giving good performance with diesel as pilot fuel used to generate spark. Supercharging in dual fuel mode improves performance of engine. Brake Thermal efficiency with supercharged producer gas-diesel is 15 % more than premixed producer gas-diesel engine [10]. Diesel can be replaced by biofuels like Honge-Oil making the fuel complete renewable. This dual fuel mode operation in CI engine shows maximum efficiency 20 % with reduction in all emissions. Tri-generation can be best option to utilize maximum part of biomass energy in higher generation systems [11].

III. SIMULATION OF ACTUAL CYCLE

The various models and equations used in simulation are briefly presented herein.

A. *Model for Heat Transfer*

Woschni's equation was used which is based on the similarity law of steady turbulent heat transfer [13], [14].

$$h = 0.820 * D^{-0.2} * p^{0.8} * W_{mv}^{0.8} * T^{-0.53} \ (kW/m^2.K) \quad (1)$$

The reference velocity W_{mv} in the above formula represent the mean gas velocity affecting heat transfer and is given for each process. D is the cylinder bore taken as the characteristic length,

$$W_{mv} = \left[C_1 * C_m + C_2 * \left(\frac{V_S * T_1}{p * v_1}\right) * (p - p_0)\right] \quad (2)$$

For the gas exchange processes, C_1 = 6.18 & C_2 = 0,

For the compression processes, C_1 = 2.28 & C_2 = 0,

For the combustion & expansion processes, C_1 = 2.28 & C_2 = 3.24*10-3

Here p_0 is the pressure in the MPa obtained for motoring and V_s is the displacement volume in

m^3 and the coefficient C_m is the mean piston speed. The subscript 1 denotes a specified time when the pressure and the temperature are known.

In the Suction and Exhaust process Woschni's heat-transfer model was used during the simulation.

Anand [15], separates out the convection and radiations terms. Typical approach to the heat transfer theory proposed by Anand is his expression for the Nusselt number 'Nu' leading to a conventional derivation of the convection heat transfer coefficient C_h.

Anand recommends the following expression to connect the Reynolds and Nusselt number:

$$Nu = b*Re^{0.7} \quad (3)$$

Where, b = 0.26, for the two stroke engines and

b = 0.49, for the four stroke engines.

The Reynolds number is calculated as,

$$Re = \frac{\rho_{cylinder}*C_m*D}{\mu_{cylinder}} \quad (4)$$

The viscosity should be that of the cylinder gas,

$$\mu_{cy} = 7.457*10^{-6} + 4.1547*10^{-8}*T - 7.4793*10^{-12}*T^2 \quad (5)$$

The mean piston velocity C_m is found from the dimensions of the cylinder stroke, L and the engine speed, N in rev/min.

$$C_m = \frac{2*L*N}{60} \quad (6)$$

The convectional heat transfer coefficient can be extracted from the Nusselt number, as

$$C_h = \frac{C_k*Nu}{D} \quad (7)$$

The parameter C_k is the thermal conductivity of the cylinder gas and can be assumed to be identical with that of air at the instantaneous cylinder temperature.

$$C_k = 6.1944*10^{-3} + 7.3814*10^{-5}*T - 1.2491*10^{-8}*T^2 \quad (8)$$

In the compression and expansion process this heat-transfer model is adopted.

B. Model for Combustion [13]

Wiebe function was used to find out the mass fraction burnt during the combustion process.

$$X_b = 1 - \exp\left[-a*\left(\frac{\theta-\theta_s}{\Delta\theta_c}\right)^{m+1}\right] \quad (9)$$

Where, X_b is the mass fraction burned, "a" is an efficiency parameter and "m" is a slope parameter.

C. Input Data [16]

For the simulation purpose Honda GX-120 engine had been selected. Its specifications are as follows:

Engine Type	: Air Cooled 4-s petrol engine
Bore * Stroke	: 60 * 42 mm
Swept Volume	: 118 cc
Comp. Ratio	: 8.5:1
Max. Net Torque	: 7.3 Nm@2500 rpm
Net Power	: 2.6 kW@3600 rpm
No. of Cylinders	: 1

Producer gas composition on volume basis: CO=24.6 %, H_2=21.9 %, CO_2=8.17 %, H_2O^g=7.75 %, N_2=37.53 %.

IV. SIMULATION RESULTS

1. Suction Stroke:

Fig. 1 shows during suction stroke pressure falls rapidly (from initial 0.93 bar) after start of piston downward acceleration up to 50^0 CA, because of low valve lift and lower curtain area allowing less air-fuel mixture coming in cylinder. The cylinder pressure falls to 0.58 bar at 50° CA. However, thereafter a gradual pressure building is observed due to increased availability of mass flow with higher range due to valve lift and curtain area is sufficiently high to allow gases to pass inside and increase the inner pressure. Peak mass inflow found at 90^0 CA. Suction happens till final pressure reaches atmospheric pressure (1.013 bar). As shown in fig 2 temperature obviously will reduce with increased availability of mass flow with increasing crank angle. The mass-θ curve in fig. 3 clearly shows gradual rise in mass flow during initial valve lift and thereby explain the trend of P-θ curve too. Thus, the basic results of suction process are as per logical trend observed in actual I.C. engines and this validates the model used in present case.

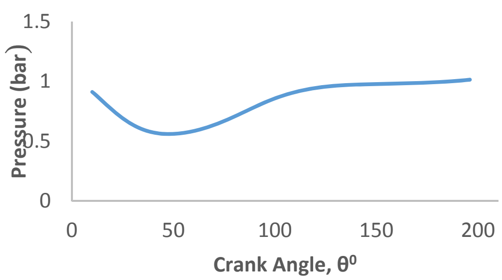

Fig.1 P-θ Curve for Suction Stroke

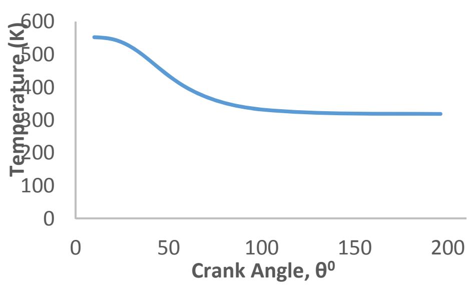

Fig.2 T-θ Curve for Suction Stroke

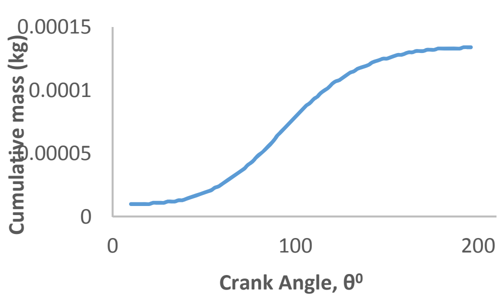

Fig.3 Mass-θ Curve for Suction Stroke

2. Compression stroke:

After suction charge is sealed and compressed with gradual rise of pressure up to 290^0 CA and stiff rise in pressure after 290^0 CA till the final pressure of 14.7 bar as shown in fig.4 and temperature 657 K as shown in fig. 5 at 336^0 CA. Specific heats found to be increasing and compression index gamma reducing at higher temperature. More heat loss occurs as inside temperature increases.

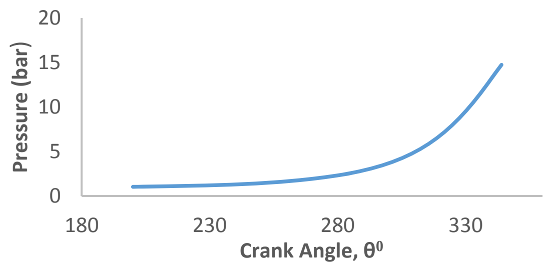

Fig.4 P-θ Curve for Compression Stroke

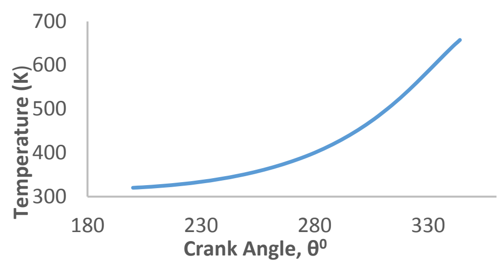

Fig.*5* T-θ Curve for Compression Stroke

3. Combustion Process:

Ignition occurs after end of compression 16^0 BTDC causing release of heat from combustion of charge produces peak pressure 38 bar at 372^0 CA as shown in fig.6. The rate of heat addition under these circumstances is more than the heat losses. As a result the temperature continues to rise and reaches to its peak value 1809 K at 379^0 CA as shown in fig.7. Mass burning curve in fig.8 shows higher mass burning rate found during high pressure and temperature formation. At 370^0 CA, 75% of mass has been burnt releasing huge amount of heat to produce peak pressure and temperature. Heat losses found more in this process.

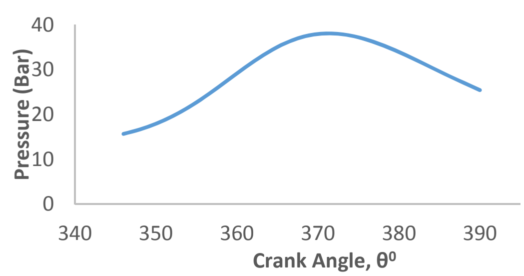

Fig.6 P-θ Curve for Combustion Process

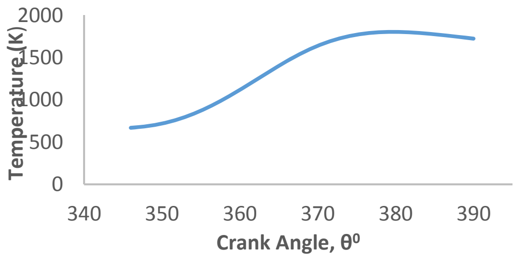

Fig.7 T-θ Curve for Combustion Process

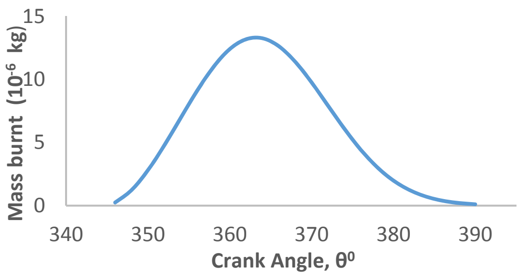

Fig.8 Mass burnt-θ Curve for Combustion Process

4. Expansion Stroke:

Combustion products with very high temperature get expanded due to increase in cylinder volume, which in turn reduce the pressure inside the cylinder drastically. At the end of combustion process the pressure inside the cylinder is 25.42 bar as shown in fig.9. This gives higher energy extraction. At the end of expansion process, pressure reduces to 3.5 bar at 510° CA, whereas temperature reaches to 990 K as shown in fig.10. P-θ Curve for expansion stroke shows initial stiff curve of pressure reduction up to 440^0 CA and then gradual expansion i.e. most of power will be transmitted before 440^0 CA.

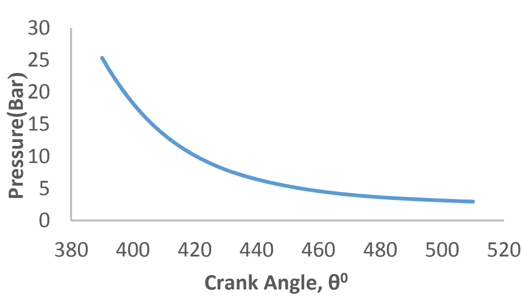

Fig.9 P-θ Curve for Expansion Stroke

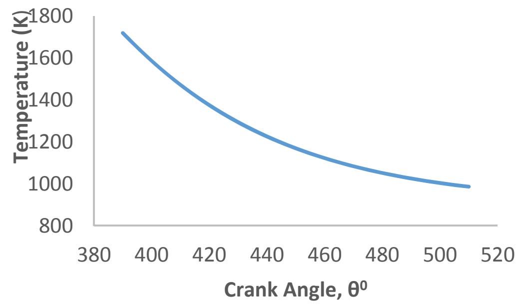

Fig.10 T-θ Curve for Expansion Stroke

5. Exhaust Stroke:

During exhaust stroke, pressure falls to 1.02 bar at 610° CA and then rises up to 1.8 bar at the end of exhaust process shown in fig. 11. After 640° CA slow pressure building inside the cylinder is observed due to throttling effect. The temperature obviously will reduce with high temperature burnt mass leaving to the atmosphere with increasing crank angle as shown in fig. 12. It is observed from the T-θ curve that, like pressure, there is not any rise of temperature during the later stage of exhaust process.

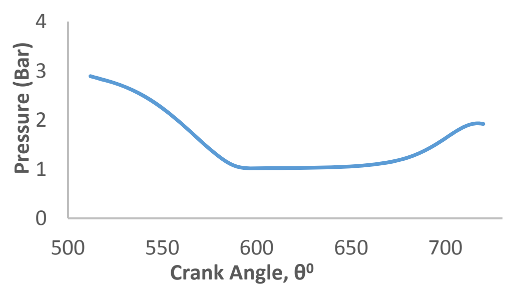

Fig.11 P-θ Curve for Exhaust Stroke

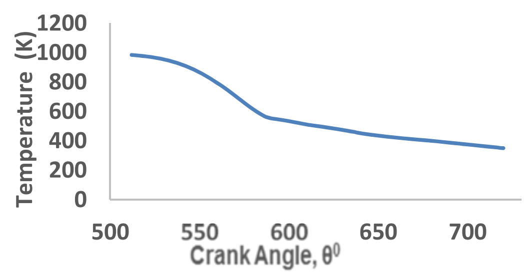

Fig.12 T-θ Curve for Exhaust Stroke

6. Full Cycle

Fig. 13 shows the variation of pressure with volume for the full cycle. Area under the curve gives the work done and thereby power output of engine. The indicated power calculated by data obtained from the simulation is 2.3 kW @3600 rpm for the Producer gas as a fuel. Thermal efficiency found to be 28.19 % with Specific fuel consumption 2.62 kg/kWh and mean effective pressure found to be 6.7 bar. Fig. 14 T-θ in fig. 14 shows high temperature region where heat loss from cylinder will be more.

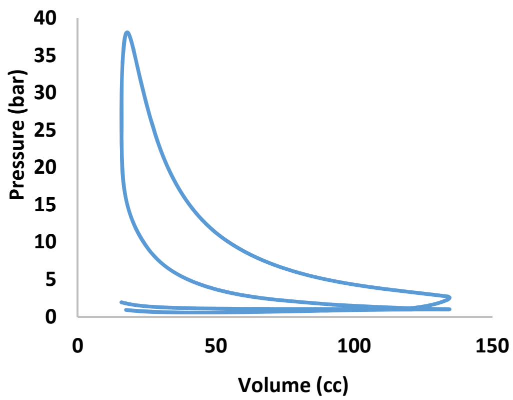

Fig.13 P-θ Curve for full Cycle

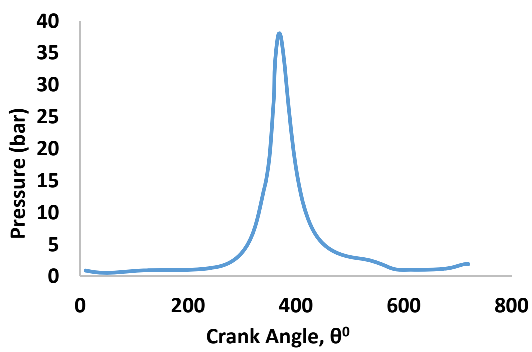

Fig.14 T-θ Curve for full Cycle

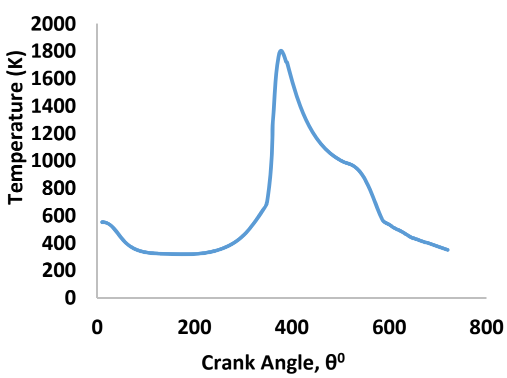

Fig.15 P-V Curve for full Cycle

V. Validation of Simulation Results

As per the specifications given in the manual of Honda GX-120 engine the net power is 2.6 kW @3600 rpm for the gasoline. The indicated power calculated by data obtained from the simulation is 2.3 kW @3600 rpm with thermal efficiency 28.19 % for the Producer gas as a fuel, which shows the relatively reduction in power and thermal efficiency 11.538 % and 5.71 % respectively as compared to gasoline. The literature clearly indicates that the engine offers 15-30 % power de-rating and 3-10 % reduction in thermal efficiency as that of gasoline [7], [9] and thus looking to this fact present simulation may be treated as adequately validated.

VI. Conclusion

- Peak pressure and temperature produced in combustion is 38 bar and 1809 K.
- The indicated power calculated by data obtained from the simulation is 2.3 kW @ 3600 rpm. These simulation results are quite in tune with Honda GX-120 engine power output considering the fact that there exist the relative de-rating of engine by 15-30 % with Producer gas as a fuel.
- Thermal efficiency found to be 28.19 % which is relatively de-rated by 5.7 % compared to gasoline as fuel.

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