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Parametric Optimization of Producer Gas fuelled Spark Ignition Engine through Thermodynamic Modelling

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Development of alternative fuel has become a necessity to quench the current/rising future energy demands and alarming environmental pollution. Producer gas/syngas has potential to be most favorite alternative fuel substitution whose performance characteristics are lower whereas emission characteristics of carbon monoxide (CO), carbon dioxide (CO_2) and hydrocarbons are better than petrol/diesel. Looking at these facts, the present work focuses on thermodynamic modelling and parametric studies on a real-life engine using producer gas as fuel to study its performance and emission characteristics. Simulated program is validated with published literature and results are found within ± 9% in terms of power and efficiency. The model is then used to study the effect of Spark advance (10°–30°), valve timing and combustion duration (40°–80°), fuel-air ratio, stroke to bore ratio and connecting rod length to crank radius to study their impact on emission and performance of real-life engine. The optimum engine performance parameters while also considering emission of nitric oxide (NO) and CO operated at stoichiometric fuel-air ratio turns out to be L/D as 1.0, Spark advance as 20° BTDC (before top dead center), combustion duration as 50°, inlet valve closing as 30° ATDC and exhaust valve opening as 10° BBDC (before bottom dead center). The efficiency, specific fuel consumption, power, CO and NO with these optimal parameters are of the order of 19.58%, 1.23 kg/kWh, 67.0 kW, 0.65 ppm and 0.017 ppb, respectively. It is believed that present work offers optimal design and operating parameters through actual thermodynamic cycle analysis which may be used as a reference for design and development of producer gas-fuelled Spark Ignition (SI) engines.

Keywords: Combustion, Dissociation, Emissions, Heat transfer, Stoichiometric

Introduction

The rate at which the energy is consumed, it will not be possible to fulfill the current and future energy demands within the emission norms by the conventional fossil fuels as they will be depleted in near future due to the limited sources.[1] These facts call for a cleaner and more sustainable green energy solution within the emission norms. Gasification is one of the possible sustainable green energy solutions after conquering the problems associated with conventional gasification technologies.[2–4] "Zero Effluent Discharge Gasification Technology" has been developed which is more efficient than conventional gasification technology and is environmentfriendly.[5,6]

However, the lower energy density of producer gas demands a proper selection of engine parameters/modifications for its efficient utilization in Internal Combustion (IC) engines. The mathematical simulation through thermodynamic modelling is the efficient way for selecting engine parameters to increase the overall performance of the engine before doing the experiments. The present research work addresses this aspect.

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Literature Review

Producer gas can be utilized as a fuel in IC engines by either of the following ways: (1) dual fuel mode in Compression Ignition (CI) engines, (2) converting diesel engines in SI mode and run on producer gas (3) producer gas alone mode in SI engine. Performance analysis of diesel engine was carried out by Raman *et al.*[7] operating the engine with producer gas in dual fuel mode. Power generation efficiency was found to be 21% with a CR of 12:1 and diesel replacement was achieved in the order of 60–65%. Hassan *et al.*[8] analysed the performance and emission characteristics of a supercharged diesel engine operated on PG-diesel dual-fuel mode. Brake thermal efficiency was observed 15% more with supercharged producer gas than premixed producer gas-diesel engine. Carbon monoxide emissions were reduced due to the presence of more air in supercharged mode but NO was increased by a small amount due to an increase in temperature. Gobbato *et al.* carried out the experimental analysis on a heavy-duty NG SI engine with PG as a fuel. During the study, 55% de-rating in power with PG as a fuel compared to NG as fuel was observed.[9] Szwaja *et al.*[10] carried out an investigation on syngas–methane combustion in the SI engine. Due to higher density, the syngas–methane mixture's LHVmix increases significantly which resulted in higher engine performance. Mixtures consisting of 40% methane–60% syngas were found to be optimal. Homdoung *et al.*[11] utilized a gaseous fuel through a downdraft gasifier from agricultural biomass in a 0.6 L SI engine. The cavity chamber was used in place of the conventional swirl combustion chamber in their study. Brake thermal efficiency was achieved in the order of 23.9% whereas the smoke density was lesser with respect to a diesel engine. Shashikanta *et al.*[12] carried out experimental performance optimization for SI produce gas engine and dedicated NG engine technology. From the experimental study, it was found that spark advance required for PG operation is at 35° BTDC as compared to CNG operation where it was 22° BTDC, for the same compression ratio in both cases i.e. 11.5:1.

A good amount of literature is available for producer gas-fired engines but very limited information is available in the open literature for parametric study through thermodynamic modelling on producer gas-fuelled SI engines. Prior to experimental analysis, the parametric optimization through thermodynamic modelling is helpful to arrive at some reasonable values of power, thermal efficiency and emissions.

Looking to this fact, in the present work, optimization of performance and emissions parameters of producer gas-fired engine is carried out using an in-house code developed using C language.

Materials and Methods

The simplest mathematical simulation of a thermal system is carried out through thermodynamic modelling which uses the basic equations such as conservation of mass, energy and equilibrium of a set of chemical reactions.[13]

Major Processes and Assumptions

The entire cycle consists of five processes i.e. suction, compression, combustion, expansion and exhaust. The complete cycle is modelled with certain realistic and logical assumptions explained in subsequent paragraphs.

Producer gas – air is a homogenous mixture that behaves as an ideal gas. The charge burns completely during the combustion process and combustion products are assumed to be CO, H_2O, CO_2, O_2, N_2, OH, H_2 and NO only. All charge gets burnt within the assumed combustion duration. Dissociation of combustion products considered are CO_2 into CO and O_2, H_2O into H_2 and O_2, H_2O into O_2 and OH, and association of N_2 and O_2 into NO.

The combustion process follows Wiebe function[14–16] by considering appropriate values of 'a' and 'm' for producer gas and combustion duration.[13]

Heat transfer area comprises of the cylinder head, cylinder wall and piston head surface only.

In suction and exhaust processes, losses due to heat transfer follow Woschni's model.[14] The heat transfer from cylinder gases to walls follows Anand's Model[17] in compression and expansion.

Dissociation and association of combustion products occur in combustion and expansion processes. Further, dissociation and association of combustion products stop at the end of the expansion process and their composition remains constant till exhaust occurs.[18]

Heat Transfer Model

The Heat transfer loss *(Q_{loss})* from the cylinder walls is calculated using Eq. 1,

$$Q_{loss} = hA(T_{wall} - T) \quad \ldots (1)$$

where, h = convective heat transfer coefficient, A = Heat transfer surface area, T = Cylinder gas temperature and T_{wall} is instantaneous wall temperature and can be given as:[19]

$$T_{wall} = (423 - 0.388\theta) \quad \ldots (2)$$

where, θ = Instantaneous crank angle

Heat Transfer Theory proposed by Anand[17] is used in which expression for the Nusselt number 'Nu' leads to a conventional derivation of the convective heat transfer coefficient.

Reaction Stoichiometry of Combustion Process

Reaction stoichiometry by considering Dissociation can be given as:

$$n^f_{CO}.CO + n^f_{CO_2}.CO_2 + n^f_{CH_4}.CH_4 + n^f_{H_2}.H_2 + n^f_{H_2O}.H_2O + n^f_{N_2}.N_2 + a(O_2 + 3.762\,N_2) \rightarrow$$
$$n_{CO}.CO + n_{CO_2}.CO_2 + n_{H_2}.H_2 + n_{H_2O}.H_2O + n_{O_2}.O_2 + n_{N_2}.N_2 + n_{OH}.OH + n_{NO}.NO \quad \ldots (3)$$

where, $a = a_{st}/\Phi$

Φ	Equivalence ratio
n_{CO}^{f}	Mole of CO in fuel
$n_{CO_2}^{f}$	Mole of CO_2 in fuel
$n_{CH_4}^{f}$	Mole of CH_4 in fuel
$n_{H_2}^{f}$	Mole of H_2 in fuel
$n_{H_2O}^{f}$	Mole of H_2O in fuel
$n_{N_2}^{f}$	Mole of N_2 in fuel
n_{CO}	Mole of CO in flue gas
n_{CO_2}	Mole fraction of CO_2 in flue gas
n_{H_2}	Mole fraction of H_2 in flue gas
n_{H_2O}	Mole fraction of H_2O in flue gas
n_{O_2}	Mole fraction of O_2 in flue gas
n_{N_2}	Mole fraction of N_2 in flue gas
n_{OH}	Mole fraction of OH in flue gas
n_{NO}	Mole fraction of NO in flue gas

Conservation of mass, energy and equilibrium equation of dissociation/association of the combustion products are being used to find the unknown constituents of the Eq. 3.

A detailed analysis of the combustion process during compression stroke along with the overall solution procedure is given below.

Detailed Analysis of Combustion Process till the Completion of Compression Stroke

Completion of a pure compression phase and on the occurrence of ignition of spark, the chemical reaction begins and heat is released within the combustion chamber and hence the temperature and pressure within the cylinder will rise by the combined effect of compression and combustion. At the point of occurrence of spark, some ignition energy (E_{ig}) is added to the system instantaneously and hence it can be written as,

$$E_{\theta_{ig}} = E_{\theta} + E_{ig} = m_{ch} C_{v_{\theta_{ig}}} T_{\theta_{ig}} \qquad \ldots (4)$$

where,

$E_{\theta_{ig}} = Total\ Energy\ at\ the\ inception\ of\ spark,$

$C_{v_{\theta_{ig}}} = Specifc\ at\ Constant\ Volume\ of\ Charge$

$T_{\theta_{ig}} = Cylider\ Gas\ Temperature\ at\ the\ inception\ of\ Spark$

$m_{ch} = Mass\ of\ Charge$

$$m_{ch} = \frac{P_{\theta_{ig}} V_{\theta_{ig}}}{R_{ch} T_{\theta_{ig}}} \left[\because V_{\theta_{ig}} = V_{\theta}\right] \qquad \ldots (5)$$

where, $R_{ch} = Gas\ Constant\ of\ Charge$

$P_{\theta_{ig}} = Cylinder\ Pressure\ at\ the\ inception\ of\ spark$

$V_{\theta_{ig}} = Cylinder\ Volume\ at\ the\ inception\ of\ spark$

Now let us consider the advancement of the piston by 'dθ' after the spark has occurred as shown in Fig. 1. During this period energy is released within the combustion chamber by burning a fraction of charge (Δmb) and at the same time work of compression is being done on the system.

Therefore, energy balance over the control volume as shown in Fig. 1 can be written as,

$$E_{\theta_{ig}+d\theta} = E_{\theta_{ig}} + \Delta E_{b,\theta} + \Delta W_c + Q_{heat} \qquad \ldots (6)$$

where, $\Delta E_{b,\theta}$ = Energy added due to combustion of charge and $\Delta E_{b,\theta}$ = mass of charge burnt in Duration 'dθ'

Assuming that the energy imparted by ignition and released by combustion of charge is uniformly distributed across the entire mass of charge and the charge achieves thermal equilibrium at each time step. The energy balance equation now can be written as

$$m_{ch} C_{v_{\theta+d\theta}} T_{\theta+d\theta} = E_{\theta,ig} + (\Delta m_b CV_{ch}) + \left[\frac{p_{\theta}+p_{\theta+d\theta}}{2}\right] [V_{\theta} - V_{\theta+d\theta}] + Q_{heat} \qquad \ldots (7)$$

Now from ideal gas equation,

$$m_{ch} = \frac{P_{\theta+d\theta} V_{\theta+d\theta}}{T_{\theta+d\theta} R_{\theta+d\theta}} \qquad \ldots (8)$$

Substituting this in energy balance equation and simplifying, we will get

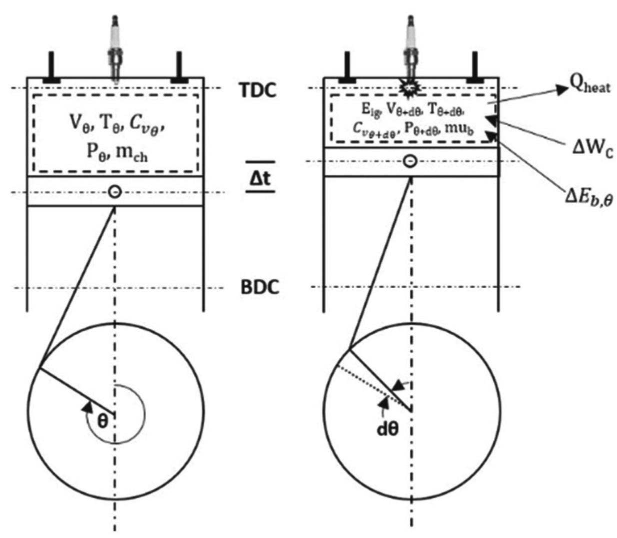

Fig. 1 — Control volume of combustion process during compression stroke before and after 'dθ' interval of crank angle

$$\frac{P_{\theta+d\theta}V_{\theta+d\theta}}{R_{\theta+d\theta}}C_{v_{\theta+d\theta}} = E_{\theta,ig} + (\Delta m_b CV_{ch}) + Q_{heat} + \left[\frac{p_\theta+p_{\theta+d\theta}}{2}\right][V_\theta - V_{\theta+d\theta}] \quad \ldots (9)$$

where,

$C_{v_{\theta+d\theta}}$ = Specific Heat of mixture of partially burnt and unburnt charges at constant volume and

$R_{\theta+d\theta}$ = Gas Constant of mixture of partially burnt and unburnt charges

Now on simplification of Eq. 9,

$$P_{\theta+d\theta} = \left[E_{\theta,ig} + (\Delta m_b CV_{ch}) + \frac{1}{2}P_\theta(V_\theta - V_{\theta+d\theta}) + Q_{heat}\right] x \left[\frac{V_{\theta+d\theta}C_{v_{\theta+d\theta}}}{R_{\theta+d\theta}} - \frac{1}{2}(V_\theta - V_{\theta+d\theta})\right]^{-1} \quad \ldots (10)$$

Thus, the parameters of interest i.e., $P_{\theta+d\theta}$ can be obtained from Eq. (10) while $T_{\theta+d\theta}$ from Eq. (8) with property variations obtained through iterative technique. Similarly, the analysis is carried out for other processes also. The solution procedure of the proposed analysis as discussed above is given in Fig. 2 whereas the overall solution procedure is given in Fig. 3.

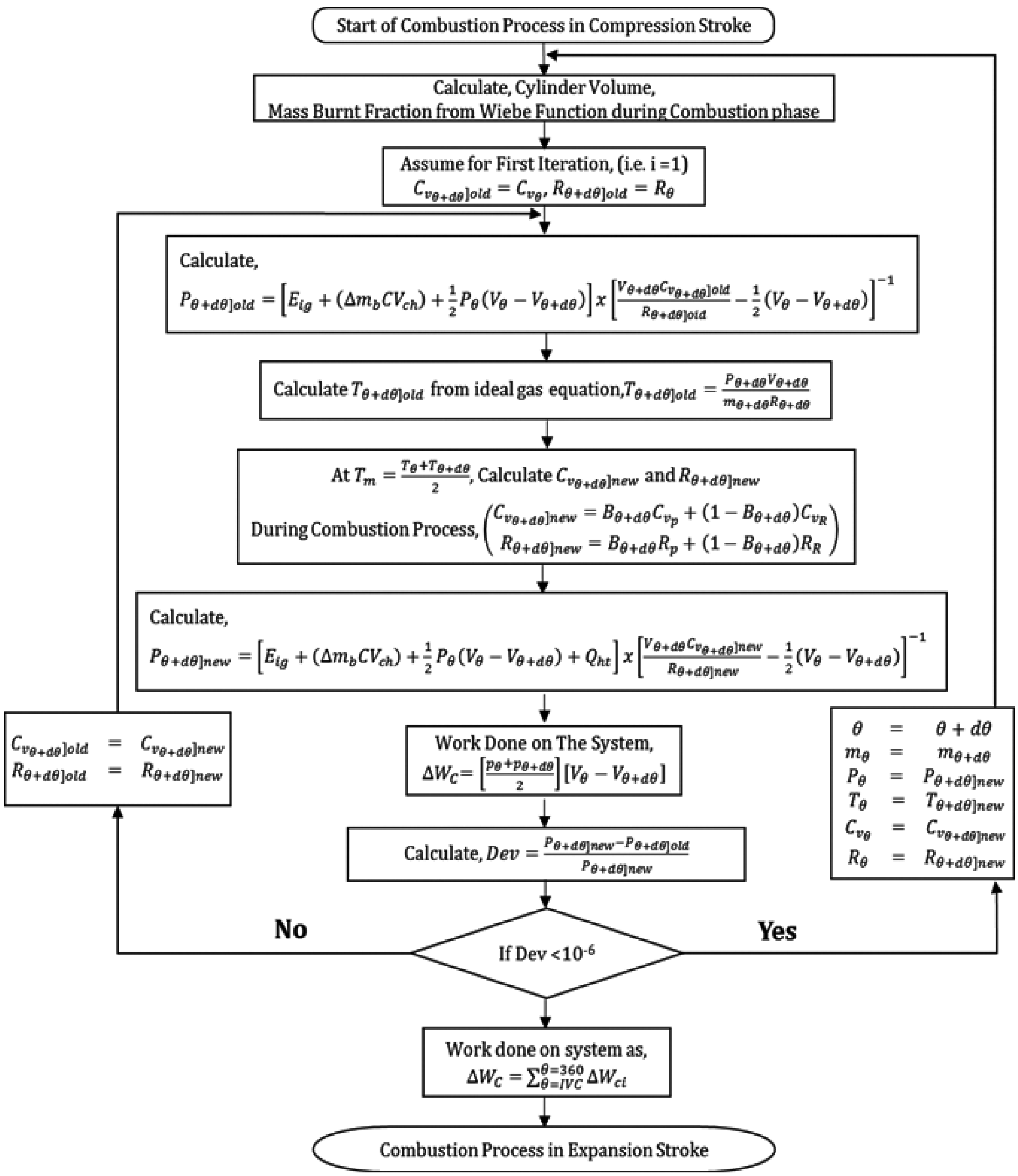

Fig. 2 — Solution procedure of combustion process till the completion of compression stroke

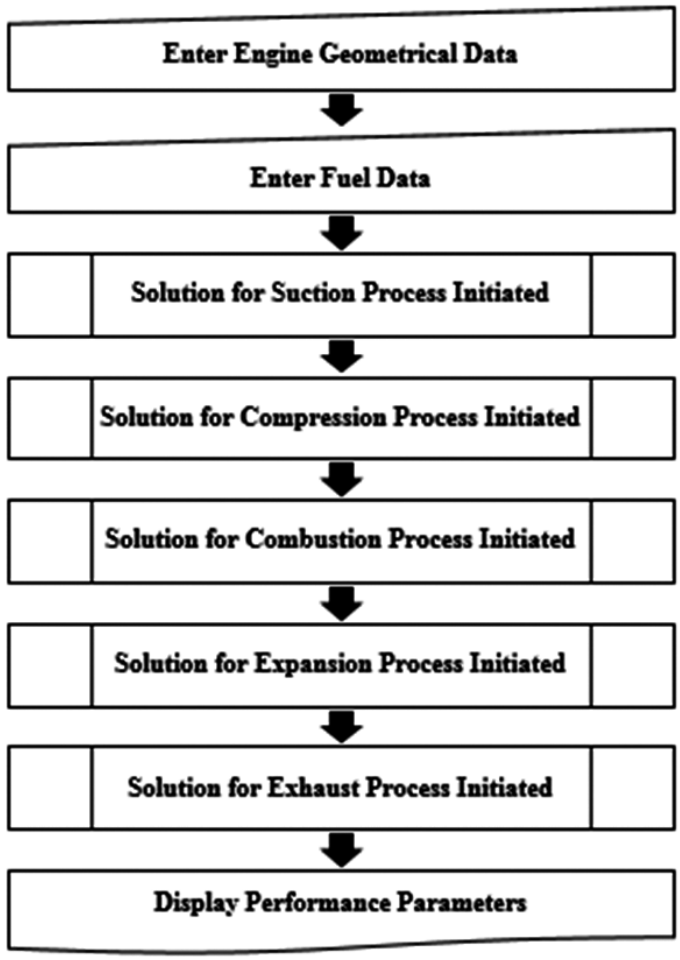

Fig. 3 — Overall solution procedure

Ina mathematical simulation, a six-cylinder SI engine has been selected. Specifications of the engine are shown in Table 1 and producer gas composition is shown in Table 2.[(5)]

Table 1 — Engine specifications

Parameters	Values
Type	4-stroke/NA
Bore × Stroke	140 × 152 mm
Number of Cylinders	Six cylinders
Rated speed	1500 rpm
Compression ratio	12:1
Ignition system	Spark ignited

Table 2 — Producer gas composition[5]

Constituent	CO	CO_2	CH_4	H_2O	H_2	N_2
Composition (% Vol. basis)	25.6	9.5	0.3	6.4	22.5	35.7

Simulation Results and Discussion

The variation of pressure and temperature against crank angle is shown Fig. 4. It can be seen from Fig. 4 that there is a gradual rise in the pressure due to combustion and the pressure rises to its peak value of 49.761 bar at 365° CA because of the exponential reaction rate as per Wiebe Function. Similarly, the temperature also rises to its peak value of 2007.96 K at 372° CA. The difference in crank angles of peak pressure and peak temperature is due to the fact that heat release continues as the charge keeps on burning after 365° CA when the peak pressure is reached.[14,16]

Further, pressure and temperature inside the cylinder keep on decreasing as the high-temperature combustion products start expanding with an increase in cylinder volume, which gives higher energy extraction at the end of the expansion process.

After the expansion process, there is a rapid fall in pressure and temperature because the burnt mass of combustion products is forced to leave the cylinder space on the opening of the exhaust valve due to a very high-pressure differential. A minor pressure rise is observed just before TDC because of the throttling effect.[14]

The pressure v/s cylinder volume diagram is shown in Fig. 5. From the results, power output, thermal efficiency and specific fuel consumption obtained are 66 kW, 20% and 1.23 kg/kWh, respectively.

The variation of NO and CO emissions from the onset of dissociation till the completion of the expansion process is shown in Fig. 6. It is observed that the peak value of these emissions is directly proportional to temperature. This is quite obvious due to the increased reaction rates with an increase in temperature. The maximum values of NO and CO are found to be 416 ppm and 1513 ppm at 372° CA just after TDC, the location of peak combustion temperature.

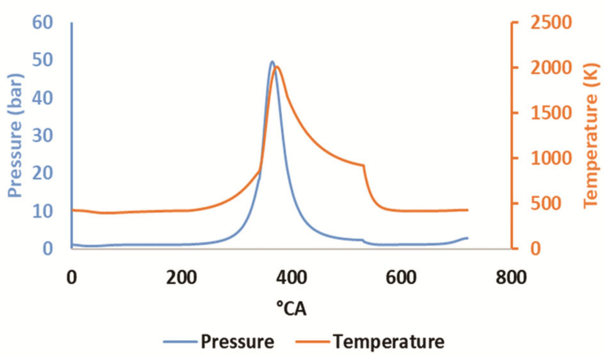

Fig. 4 — Pressure and Temperature vs CA

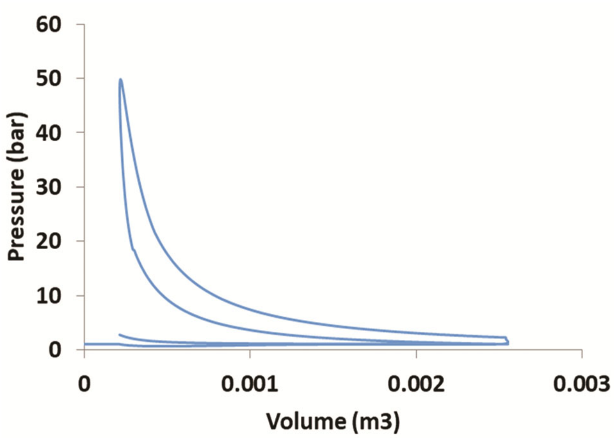

Fig. 5 — P-V diagram for producer gas engine

Validation of Results

The present model has been validated with two published literatures[11,20] as shown in Table 3.

Deviations of present work compared with published literature are within −5% to 9% in terms of indicated thermal efficiency and indicated brake power which is reasonable and the model can be considered as adequately validated.

Parametric Optimization

After validation of the model, parametric optimization is carried out by varying different parameters as presented herein.

Influence of Fuel-Air Ratio (∅)

Producer gas-fuelled engine is mostly operated near stoichiometry air-fuel ratio.[21] In order to achieve better performance with minimum emissions, the effect of the fuel-air ratio is studied. The fuel-air ratio is varied from 0.8 to 1.2 in the interval of 0.1. Variations in power, specific fuel consumption, efficiency, NO and CO with respect to fuel-air ratio are shown graphically in Fig. 7(a) & (b), respectively.

From the results, it is seen that power does not change significantly as the fuel-air ratio increases which are expected. Efficiency reduces with an increase in the fuel-air ratio, which is similar to the gasoline fuelled SI engines. Efficiency increases as the equivalence ratio are decreased below unity. This occurs because the temperature of burnt gases decreases after combustion, thereby decreasing its

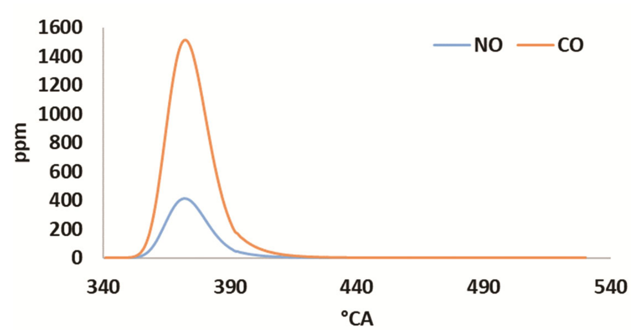

Fig. 6 — NO and CO variation over expansion process

Table 3 — Validation of simulation results

	Experimental Results	Theoretical Analysis	Present Model Predictions	% Dev.	Ref.
Power	3 kW @1700 rpm	—	3.2 kW @1700 rpm	6.25	11
Efficiency	21%	—	20%	−5.00	
Power	3.1 kW @1700 rpm	3.0 kW @1700 rpm	3.124 kW @1700 rpm	0.77	20
Efficiency	18%	17%	19.78%	9.00	

specific heat which causes the burnt gases to expand through a larger temperature ratio prior to exhaust.

As the equivalence ratio increases above unity, a lack of sufficient air for complete combustion/oxidation of the fuel leads to a decrease in temperature that eventually decreases the engine efficiency.[14]

Influence of Stroke to Bore Ratio (L/D)

The stroke to bore ratio plays an important role in the performance of the IC engine. Thus, the L/D variation study helps to obtain an optimal engine configuration. The Effects of the L/D ratio on performance parameters and emissions are shown in Fig. 8 (a & b). L/D ratio is varied from 0.8 to 1.2 and for each variation, swept volume is kept constant.

It is seen from Fig. 8(a) that power and efficiency decrease with an increase in L/D ratio while specific fuel consumption (SFC) increases on increasing L/D ratio, although the change is minimal up to 1.1. The reason for decreasing efficiency with increasing L/D ratio is that as stroke length increases the surface area available for heat transfer increases leading to higher losses, thus affecting the engine efficiency.

The trend of CO and NO emission decreases with an increase in the L/D ratio which clearly indicates that an optimum value must be selected. The trends of performance and emissions are in agreement with Yamin *et al.*[22] which states that though a larger bore-to-stroke ratio engine (low L/D) looks tempting, the selection of L/D ratio should also take care of the emission rate. Thus, a square engine with L/D=1 can be selected as optimal with respect to efficiency 19.947% and emissions of CO and NO are 0.659 ppm and 0.0205 ppb, respectively.

Influence of Connecting rod length to Crank Radius Ratio (l/r)

For an IC engine, the l/r ratio also plays a considerable role in deciding optimum performance and emissions. Connecting rod length is varied from 3.6 to 4.4 in the interval of 0.2 by keeping the crank radius constant. As it can be seen from Fig. 9(a) that the variation in power and efficiency w.r.t. l/r ratio is not significant, however, for l/r above 3.8 the value of SFC reduces. These results are in agreement with Suzuki *et al.*[23] where the improvement ratio of fuel economy starts decreasing as SFC increases above l/r = 3.3. The results are also quite in tune with Adachi *et al.*[24] The reason for increasing efficiency with increasing l/r ratio between 3.8 and 4.2 is that as the l/r ratio increases stroke length increases which results in the increment of efficiency. The optimum value seems to be n = 4 with 19.95% efficiency and 1.2309 kg/kWh SFC along with reasonable emissions (Fig. 9b).

Influence of Spark Advance

As producer gas has a lower energy density, for better utilization, it is suitable to ignite the producer gas relatively earlier than conventional fuels. An optimal spark advance is desired since combustion ceases just after the TDC leading to built-up of maximum pressure at the beginning of expansion stroke, thus giving maximum work output. The

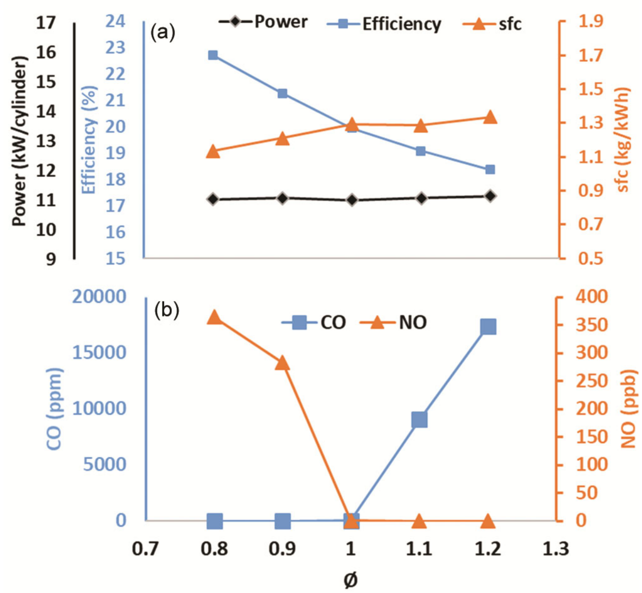

Fig. 7 — Influence of fuel-air ratio (Ø) on (a) Efficiency, Power and SFC; (b) NO and CO emission

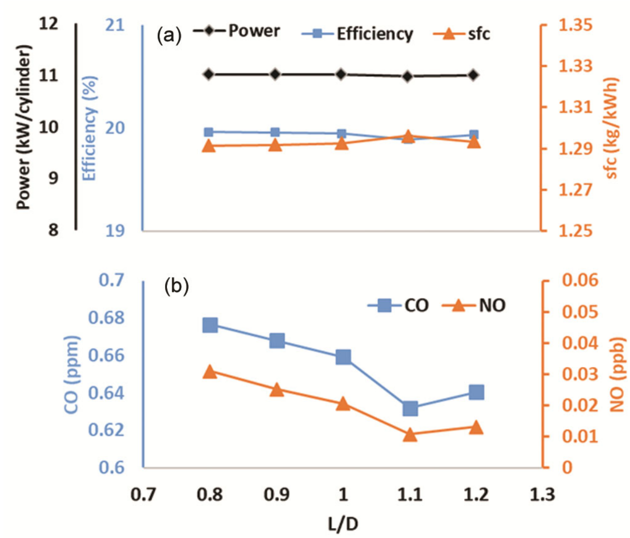

Fig. 8 — Influence of stroke to bore (L/D)ratio on (a) Efficiency, power and SFC; (b) NO and CO

selection of a larger spark advance leads to an increase in compression work which directly reduces the power output and thermal efficiency. Looking at these facts, a spark advance study is carried out and it is varied from 10° to 30° in the interval of 5°.

Variation in power, efficiency, SFC and emissions of CO and NO are shown in Fig. 10 (a & b). The combustion duration was kept constant at 50°. It is seen from Fig. 10 that efficiency, power, NO and CO increase with an increase in spark advance. With an increase in spark advance, pressure and temperature during the combustion process also increase. This is because it allows the charge to burn early before TDC, thereby creating more turbulence in the cylinder and a subsequent increase in efficiency and power. Ecological consideration of fuel shows that a 20° SA seems to be an optimum value with 20.65 % efficiency and reduced CO and NO.

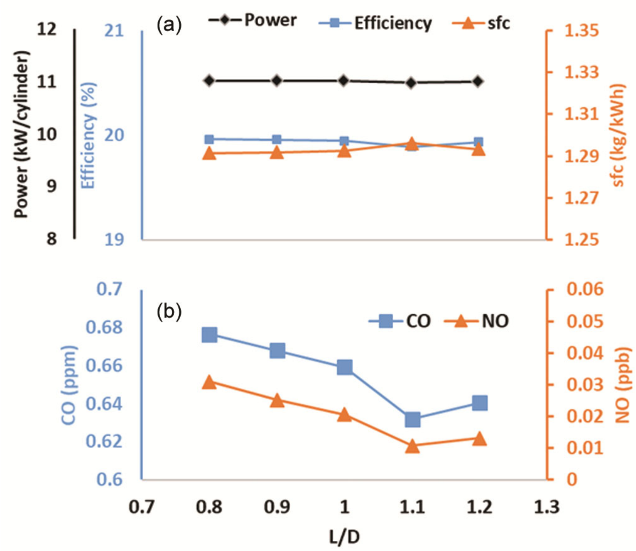

Fig. 9 — Influence of connecting rod length to crank radius ratio (l/r) on (a) Efficiency, Power and SFC; (b) NO and CO

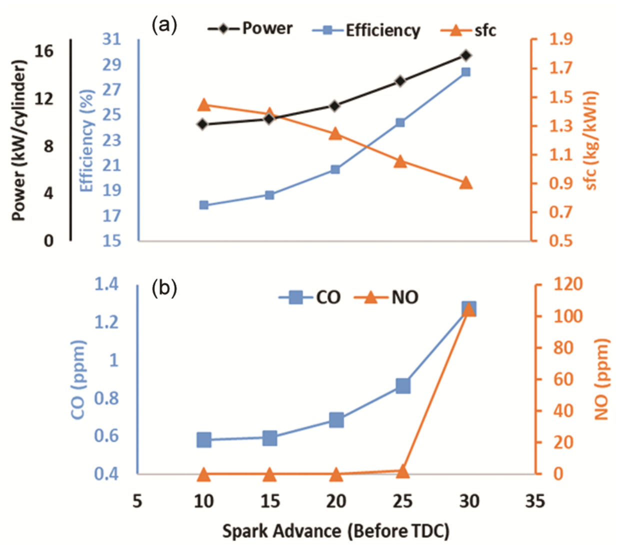

Fig. 10 — Influence of spark advance on (a) Efficiency, Power and SFC; (b) NO and CO

Influence of Combustion Duration

For complete combustion, the combustion duration is an important parameter that can affect the efficiency and power output as it depends on the fuel properties and composition. The combustion duration needs to be optimal in order to achieve maximum pressure and temperature inside the cylinder of the engine for better power and efficiency. Generally, combustion duration in the I.C. Engine is in the range of 50° to 65°.[13–15] In the present analysis, the combustion duration is varied from 40° to 80° in an interval of 10°. The effect on efficiency, power and SFC is shown in Fig. 11(a). The variation in emissions of NO and CO is plotted in Fig. 11(b).

From Fig. 11(a), it is clear that both efficiency and power output show a 'U' trend against the variation of combustion duration. SFC on the other hand shows an inverted 'U' trend. From Fig. 11(b), it is clear that for combustion duration less than 50° and for more than 70°, the emissions increase rapidly. Taking 20° SA into consideration, the optimal combustion duration lies between 50° to 55°. The combustion duration is strongly depending on the amount of H_2 and CO available in fuel (H_2 is fast-burning species, while CO is the slower one). Thus, by consideration of performance parameters and emission characteristics, optimum combustion duration is taken as 50°.

Influence of Inlet Valve Closing (IVC) and Exhaust Valve Opening (EVO)

The overall measure of the effectiveness of a four-stroke engine is affected by many parameters. Valve timings are also one of them which affect the performance of an SI engine.[14] This basically depends on the calorific value of fuel and the selection of valve-based parameters which are optimized based on power ratings of the engine. Early opening of the exhaust valve reduces the blowdown losses. But, too early opening causes loss of a part of the expansion stroke.

In this study, IVC is varied from 10° to 30° in an interval of 5°, i.e., the inlet valve is closed 10° to 30° after BDC. The effect on efficiency, power output and SFC are shown in Fig. 12(a), while the emission characteristics are shown in Fig. 12(b).

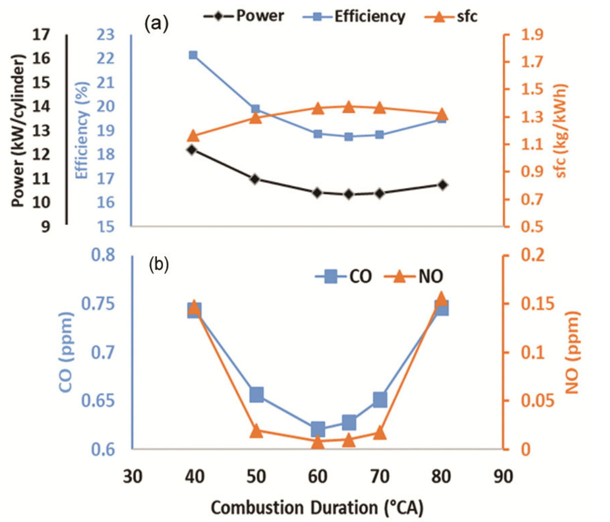

Fig. 11 — Influence of combustion duration on (a) Efficiency, Power and SFC (b) NO and CO

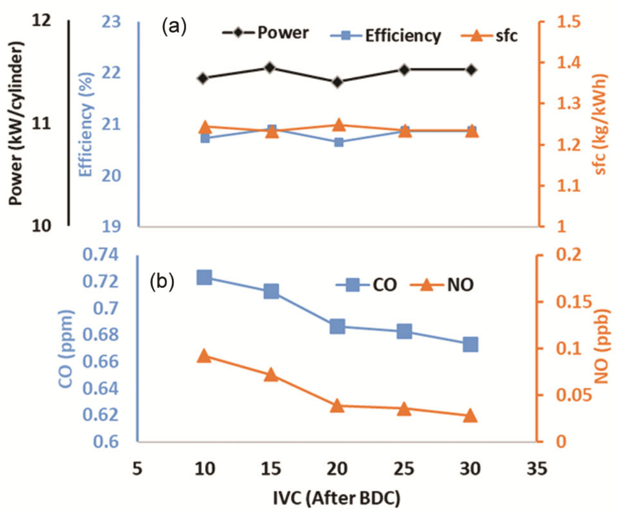

Fig. 12 — Influence of inlet valve closing (IVC) on (a) Efficiency, Power and SFC (b)NO and CO Fig. 12(b)

The effect of inlet valve closing after the BDC on the efficiency and power output is not significant. After 20° IVC, the efficiency and power show slight increment whereas CO and NO reduced by 3.50% and 42.61%, respectively with the increase in IVC timing. Hence, the optimum value of IVC is found to be 30° ABDC after considering all aspects.

EVO angle is varied from 10° to 30° in an interval of 5°. The effect on performance and emissions are shown in Fig. 13 (a & b). It is seen that the effect on efficiency and power is negligible up to 25° before BDC. At EVO of 30° or more, a part of the expansion stroke is lost, hence there is a reduction in power and efficiency.

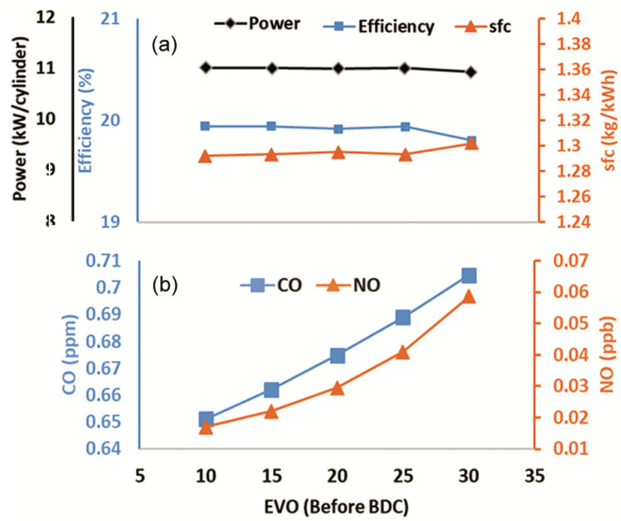

Fig. 13 — Influence of exhaust valve opening (EVO) on (a) Efficiency, Power and SFC;(b) NO and CO Fig. 11(b)

Table 4 — Parameters and their optimum values

Parameter	Optimum value
Ø	1.0
L/D ratio	1.0
Spark advance	20° BTDC
IVC	30° ABDC
l/r ratio	4.0
Combustion duration	50°
EVO	10° BBDC
Efficiency	19.58 %
Power output	67 kW
SFC	1.23 kg/kWh
NO	0.017 ppb
CO	0.65 ppm

Using the above parameters, the performance and emission characteristics were found to be as follows:

Also, as the EVO is increased, i.e., earlier the exhaust valve is opened, higher is the emission of CO and NO. Therefore, the optimum value of EVO is taken as 10° BTDC as CO and NO reduced by 7.59% and 71.05% as compared to EVO at 30°.

From above analysis, the optimum parameters can be summarised in Table 4.

Conclusions

Thermodynamic modelling of producer gas-fueled SI engine is carried out using an in-house code developed in C language. Actual cycle parameters were considered during the modelling of all processes while taking into account heat transfer, dissociation and association of combustion products. The developed thermodynamic model is validated with the published experimental data and overall deviations are obtained within ± 9% in terms of power and efficiency.

A parametric study is carried out by varying parameters like fuel-air ratio, L/D ratio, l/r ratio, spark advance, combustion duration and valve timings. This attempts to give an optimum set of engine performance parameters while targeting low emissions of NO and CO.

The optimum engine performance parameters so obtained are: fuel-air ratio to be maintained as stoichiometric, L/D ratio as 1.0, spark advance as 20° BTDC, combustion duration as 50°, inlet valve closing as 30° ATDC and exhaust valve opening as 10° BBDC. The Efficiency, specific fuel consumption, Power, CO and NO with these optimal parameters are of the order of 19.58 %, 1.23 kg/kWh, 67 kW, 0.65 ppm and 0.017 ppb, respectively.

It is believed that the present work offers optimal design and operating parameters for producer gas fired SI engine through actual thermodynamic cycle analysis which may be used as a reference for design and development of dedicated producer gas-fired SI Engine.

Nomenclatures

CA	Crank angle
CNG	Compressed natural gas
CR	Compression ratio
EVO	Exhaust valve opening
IVC	Inlet valve closing
LHV	Lower heating value
Mix	Mixture
NG	Natural gas
PG	Producer gas
SFC	Specific fuel consumption
SI	Spark ignition
TDC	Top dead centre

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