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Simulation Models for Spark Ignition Engine: A Comparative Performance Study

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Abstract

A single-zone zero-dimensional model for any hydrocarbon fuel based on Wiebe heat release function has been implemented in Simulink to test the performance of spark ignition engine. Annand's model for convective heat losses is taken for modeling of engine cycle. The Simulink results are validated with experimental results from literature. The peak pressure during combustion drops with decrease in intake pressure which is similar to the experimental trend. With increase in speed from 1000rpm to 4000rpm, the brake thermal efficiency drops from 25% to 17% while the indicated efficiency is seen to be almost constant (32%). Two dimensional Computational Fluid Dynamic model using FLUENT of experimental test engine is prepared for axi-symmetric flow in the combustion chamber. It is a setup for closed cycle simulation that can predict pressure, mass burn fraction. CFD Simulation is carried out on an experimental engine set up at rated rpm of engine (1.8 HP @3600 rpm). The CFD results with the validated Simulink model for this engine configuration shows that at low speeds (1000 rpm), the maximum cylinder pressure prediction is about 8% higher for CFD analysis while this deviation is seen to be about 3% at higher speed (3600rpm). The Simulink model is subsequently used to test the predications of brake power and subsequently compared with the experimental results and CFD studies. Both the predictions are found to be in good agreement with the experimental results.

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*Keywords:*spark ignition engine; single zone combustion model; combustion duration; heat release rate and mass burn fraction.

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1. Introduction

The Internal combustion engines have been present since a century and their performance, fuel economy has been greatly improved from the past and will continue to increase. There has been a lot of research going on worldwide experimentally to bring new fuel injection and combustion technologies for gasoline and diesel engines such as HCCI (Homogeneous Charge Compression Ignition), GDI (Gasoline Direct injection) etc. which are more efficient and produce lesser emissions [1]. All the technologies introduced, cannot be tested experimentally as it is costly affair and time consuming. The mathematical model based simulation using different environments like Simulink, CFD etc., makes the analysis easy, less time consuming and economical, for the different technologies. The results of these simulations can be implemented for the new technology in the engine.

Nomenclature

a	Crank radius (m)	Q_{in}	Total heat released during combustion (kJ)
A_p	Piston area (m^2)	r	Compression ratio
B	Engine bore diameter (m)	R	Gas constant for air (kJ/kg K)
c_p	Specific heat at constant pressure (kJ/kg)	T_i	Intake temperature (K)
c_v	Specific heat at constant volume (kJ/kg)	T_m	Manifold temperature (K)
F_n	Piston Force (N)	V_c	Clearance volume (m^3)
h_c	Heat transfer coefficient (W/m^2-K)	W_d	Indicated Work done by engine (N-m)
K	Thermal conductivity (W/m-K)	X_b	Mass burn fraction
l	Connecting rod length (m)		
m_o	Mass of charge at intake valve closing (kg)	**Greek symbols**	
m	Mass of fuel consumed (kg/hr)	θ	Crank Angle (°)
N	Engine speed (rpm)	η_{ith}	Indicated thermal efficiency
n	Number of cycles	$\Delta\theta$	Combustion duration (°)
P	Cylinder pressure (bar)	θo	Crank angle at spark time (°)
P_o	Intake pressure (bar)	μ	Dynamic viscosity (N-s/m^2)
Q_{ch}	Chemical energy released during combustion (kJ)	φ	Fuel air equivalence ratio
Q_{ht}	Heat loss from wall (kJ)	γ_1	Specific heat ratio of mixture

A zero-dimensional model was used to study computationally the generation of irreversibility during combustion between hydrogen and hydrocarbon fuels like natural gas and landfill gas [3]. The author [4] tested the double-Wiebe function model using least square method for SI engine, he found that the double Wiebe function based model fits better than the single Wiebe function model. Sometimes the engine crank angle position also described in terms of mass fraction burn [5]. Further, it was demonstrated that the decrease in irreversibility with increase in hydrogen content of fuel, thus reflecting the increase in second law efficiency [6]. The model was applied to a multi-cylinder four strokes, turbocharged and after cooled natural gas spark ignition (SI) engine running on synthesis gas. The estimation of irreversibility associated with various sub-processes occurring during combustion in a hydrogen fuelled spark ignition engine was reported by Ismail and Mehta [7]. They developed a quasi-three dimensional phenomenological combustion model and used for predicting the pressure and temperature histories of a single cylinder four stroke spark ignition engine operating at equivalence ratios of 0.3 to 0.75 in speed range 1500 to 3500 rpm. The spark ignition engine torque identification using nonlinear modelling focused on the time lag between the cylinder pressure change with torque change [8] In the present investigation, the development of mathematical model of combustion process for small scale in-house engine with its own simulation (using MATLAB Simulink) is attempted. Here, a simulation model of actual process occurring during thermodynamic cycle of a real spark ignition engine (97 cubic centimetre air-cooled four-stroke SI engine) is developed. The first law of thermodynamics is proposed for single-zone model spark ignition engine where the heat transfer is obtained through Annand's correlation [2, 9]. This model is used to predict the engine heat transfer rate and combustion performance parameters.Shapiro and Van Gerpen developed the model to perform the exergy analysis for both SI and diesel engines by using a two-zone combustion model [10].

2. Model Development

A single-zone thermodynamic model is chosen for computer simulation for a four stroke SI engine. The model is divided into parts (subsystems) with different processes (compression, combustion, expansion, exhaust and intake) during 720° crank angle (CA) of one-thermodynamic cycle in the following sequence: intake valve closing (IVC), spark timing, end of combustion (EOC), exhaust valve opening (EVO), exhaust process (from EVO to EVC/IVO) and intake process (from IVO to IVC).

During the combustion process, the mixture can be assumed to be unburned and in expansion process (end of combustion to EVO), the cylinder consists of burned gases. Spatial homogeneity of pressure and temperature for the whole cylinder are considered for each zone. From IVC to EOC, the specific heats are calculated by considering only fresh gases to be present in the mixture while the properties from EOC to EVC are calculated considering only burned gases. The gaseous mixture follows ideal gas law. It is assumed that exhaust valve closes at the instant the intake valve opens. Heat transfer is considered to happen only during the closed cycle operation i.e. IVC to EVO. The Simulink model has following computation methodologies for determination of various parameters;

<u>Volume:</u> Using the geometry parameters of the engines, the cylinder volume at any crank angle by following relation;

$$V(\theta) = V_c + \left(\frac{\pi B^2}{4}\right)\left[1 + a(1-\cos\theta) - \sqrt{l^2 - (a\sin\theta)^2}\right] \tag{1}$$

<u>Heat release during combustion:</u> The experimental investigation of combustion process in engines is very difficult. So, the combustion investigation in engines has been carried out by analyzing cylinder pressure data in single-zone thermodynamic model using Wiebe function for calculating heat release during combustion [9]. A functional form (Eq. 2) is often used to represent the mass fraction burned versus crank angle curve [3, 4] while the heat release rate from combustion is obtained from Eq. (3).

$$X_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{m+1}\right] \tag{2}$$

If δQ_{ch} is the heat released from combustion, then the following relation;

$$\frac{dQ_{ch}}{d\theta} = \begin{cases} 0, & \theta_{ivc} < \theta \le \theta_0 \ and\ \theta > \theta_{eoc} \\ Q_{\max}\dfrac{dx_b}{d\theta}, & for\ \theta_0 < \theta \le \theta_{eoc} \end{cases} \tag{3}$$

Here, $\frac{dx_b}{d\theta}$ is the fuel mass burn rate which can be calculated from Wiebe function (Eq. 2). So the equation of mass burn fraction takes the following form;

$$\frac{dx_b}{d\theta} = \frac{a(m+1)}{\Delta\theta}\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^m \exp\left[-a\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{m+1}\right] \tag{4}$$

where, 'a' is combustion completeness factor and tells about the efficiency of combustion; It depends upon the intensity of charge motion, engine design [4] and falls in the range of $2 \le a \le 6$, $\Delta\theta$ is the combustion duration and m is the form factor that affects the shape of mass burned profile. These parameters determine the shape of Wiebe function and hence its accuracy to predict the actual heat release during combustion.

Combustion duration: The total mass burnt during combustion process and the heat release are independent of the combustion duration ($\Delta\theta$). However, if same heat is released in small combustion duration, the rate of heat release will be more and pressure peak would be higher as compared to the large combustion duration [11].

$$\Delta\theta(r,N,\varphi,\theta_0) = f_1(r) f_2(N) f_3(\varphi) f_4(\theta_0)\Delta\theta_1$$

$$f_1(r) = 3.2989 - 3.3612\left(\frac{r}{r_1}\right) + 1.0800\left(\frac{r}{r_1}\right)^2;\ f_2(N) = 0.1222 + 0.9717\left(\frac{N}{N_1}\right) + 5.0510\times10^{-2}\left(\frac{N}{N_1}\right)^2 \tag{5}$$

$$f_3(\varphi) = 4.3111 - 5.6393\left(\frac{\varphi}{\varphi_1}\right) + 2.3040\left(\frac{\varphi}{\varphi_1}\right)^2;\ f_4(\theta_0) = 1.0685 - 0.2902\left(\frac{\theta_0}{\theta_{01}}\right) + 0.2545\left(\frac{\theta_0}{\theta_{01}}\right)^2$$

Thus, the combustion duration at any operating conditions for a particular engine can be estimated from Eq. (5).

Cylinder temperature: The first law of thermodynamics applied to a closed volume of combustion chamber can be written in the following manner.

$$\delta Q_{ch} = dU_s + \delta Q_{ht} + \delta W \quad \Rightarrow \frac{\delta Q_{ch}}{d\theta} = mCv\frac{dT}{d\theta} + \frac{dQ_{ht}}{d\theta} + P\frac{dV}{d\theta} \tag{6}$$

Rearranging Eq. (6), the following relation is obtained;

$$\frac{dT}{d\theta} = \frac{1}{mCv(T)}\left[\frac{dQ_{ch}}{d\theta} - \frac{dQ_{ht}}{d\theta} - P\frac{dV}{d\theta}\right] \tag{7}$$

Convective heat transfer rate: Using the Annand's correlation [2, 9] for convective heat transfer coefficient is obtained.

$$\frac{h_c B}{k} = a\left(\frac{\rho \bar{S}_p B}{\mu}\right)^b;\ b = 0.7;\ \bar{S}_p = \frac{2aN}{60}\ (Mean\ Piston\ Speed)$$

$$\mu = \frac{\mu_{air}}{1+0.027\varphi} = \frac{3.3\times10^{-7}\times T^{0.7}}{1+0.027\varphi};\ k = \frac{9\gamma-5}{4}\mu Cv \tag{8}$$

Here value of a changes with intensity of charge motion and engine design and falls in the range $0.35 \le a \le 0.8$.

Pressure rise: Ideal gas law for closed cycle operation is used to obtain the pressure rise in the combustion chamber and this equation can be differentiated with respect to crank angle.

$$P = \frac{m_0 RT}{V} \quad \Rightarrow \frac{dP}{d\theta} = \frac{1}{V}\left[m_0 R\frac{dT}{d\theta} - P\frac{dV}{d\theta}\right] \tag{9}$$

Gas exchange process: In this process, the system is considered to be open system, from which gas can leave or enter the system. It is correlated with the valve opening area and pressure variation with crank angle [12].

$$A_{ex} = A_{exo}\left|\left\{\sin\left(\frac{180(\theta-\theta_{evo})}{\theta_{evc}-\theta_{evo}}\right)\right\}^{1/3}\right|;\quad A_{in} = A_{ino}\left|\left\{\sin\left(\frac{180(\theta-\theta_{ivo})}{720+\theta_{ivc}-\theta_{ivo}}\right)\right\}^{1/3}\right|$$

$$\frac{dp}{d\theta} = \gamma_1 p\left(\frac{1}{m}\frac{dm}{d\theta} - \frac{1}{V}\frac{dV}{d\theta}\right)_{exhaust};\quad \frac{dp}{d\theta} = \gamma_2\left(\frac{RT_m}{V}\frac{dm}{d\theta} - \frac{p}{V}\frac{dV}{d\theta}\right)_{intake} \tag{10}$$

Here, A_{ino} and A_{exo} are maximum opening in intake and exhaust valves. Both the equations require mass flow rate which can be found from the equations of fluid mechanics.

Indicated torque: Indicated torque $T(\theta)$ as a function of crank angle [13] is obtained from the piston force (F_P) by neglecting inertia of various components and friction.

Piston force,
$$F_P = (p - p_{atm}) A_p; \quad T(\theta) = F_p\, a \left(\sin\theta + \frac{\sin 2\theta}{2\sqrt{(l/a)^2 - \sin^2\theta}} \right) \tag{11}$$

Indicated work done,
$$W_d = \int_{IVC}^{IVC+720} \left(p(\theta) \frac{dV}{d\theta} \right) d\theta \tag{12}$$

Indicated Power,
$$I.P. = \frac{W_d}{120} \times N \tag{13}$$

Indicated Efficiency,
$$\eta_{ith} = \frac{I.P.}{m \times CV} \tag{14}$$

Based on the above formulation for the zero-dimensional and single zone thermodynamic model simulation, the Simulink model is being introduced for a four-stroke SI engine. This model has been built in solvers for ordinary differential equation, when the model is formulated in appropriate logical manner. The flow chart of Simulink model is shown in Fig. 2.

Table 1. Specification of test engines

Sl. no	Engine specification	Simulink validation with experimental results [12]	Simulink with CFD validation
1	Bore (B) , mm	79.4	52
2	Stroke ($2a$) , mm	111.2	46
3	Connecting rod (l), mm	233.4	81
4	Compression ratio (r)	7.4	4.8
5	Speed (N), rpm	4000 @5 HP	3600 @1.8HP
6	Spark timing in terms of crank angle	25° bTDC	25°bTDC
7	Intake manifold pressure (P_o), bar	1	1
8	Relative Fuel-air ratio (φ)	1	1
9	Fuel	C_8H_{18} (Gasoline)	C_8H_{18} (Gasoline)

3. Results and Discussion

The Simulink model is developed by considering mathematical formulation given in Section 2 and the flow chart (Fig. 1). First, the model simulation results were validated with the experiments performed on the engine by the researchers [12]. The specifications of this engine are given in Table 1 (Column 3). Subsequently, the same Simulink model is used for another engine with specifications as mentioned in Table 1 (Column 4). Then, the performance of the model is compared with that CFD results. In this way, the suitability of the Simulink model is justified for its application to spark ignition engine.

3.1. Validation of Simulink model

Using the heat release rate obtained from experimental data [12], the combustion duration for those operating conditions of experimental engine, the reference values were found after tuning combustion model. At crank angles

25°before top dead centre (bTDC), the heat release rate becomes zero and can be approximately taken as end of combustion so that the combustion duration remains as 45°.

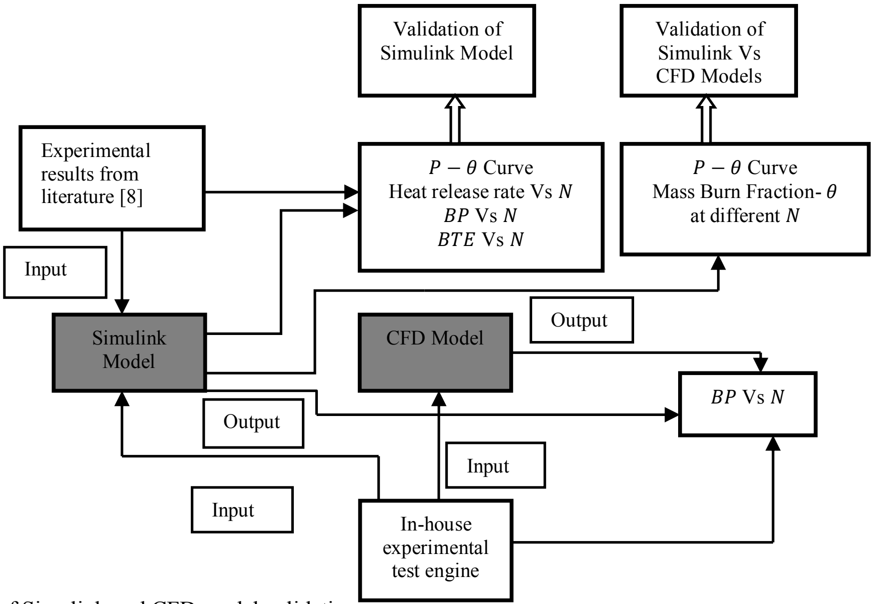


Fig.1. Flowchart of Simulink and CFD model validation process

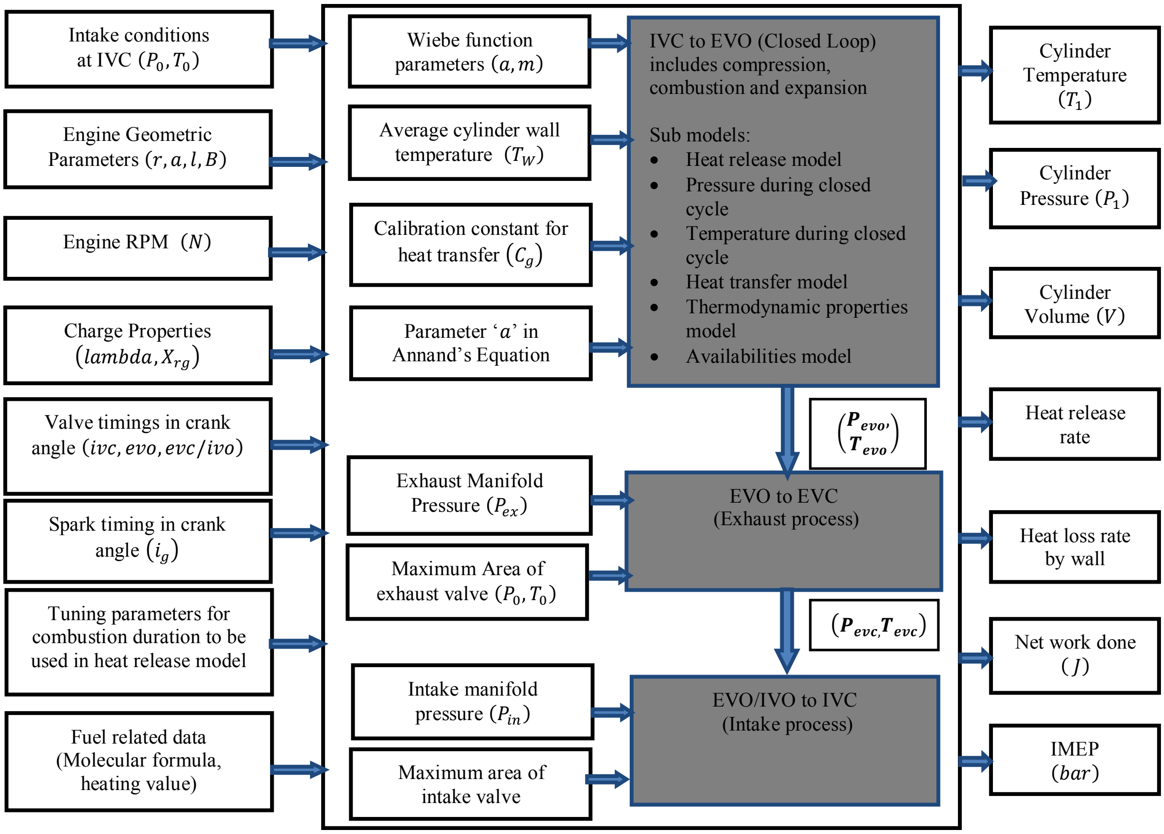


Fig.2. Flowchart of Simulink model.

The following reference parameters are obtained as below;

$$r_1 = 7.4, N_1 = 4000\ rpm, \emptyset_1 = 1.1, \theta_{01} = -25^0\ and\ \Delta\theta_1 = 45^0.$$

These values were taken as input to the main model for simulating the cycle for any operating condition. The parametric studies were done and results were compared with experimental data. The simulation results follow the experimental trend of decrease of peak pressure as intake pressure *(P_0)* is reduced. (Figs. 3 a-d).Also, they are able to follow the trend of pressure rise at the start of combustion and its decrease during expansion. Assuming the experiments were performed with care and that experimental data [12] are good, there is some mismatch in terms of peak pressure. The deviation in peak pressure found to be 6%, which reduces to 1% between Simulink and experimental results with decrease in intake pressure. This anomaly is always lies between simulation and experiments in case of complex system like IC engines especially when higher intake pressure. It is mainly because of the fact that combustion is a very complex phenomenon and the simple Wiebe function based model is not sophisticated enough to take into account many factors that affects the combustion process. However, the prediction of heat release during combustion from the model can be considered as an accurate estimate predicts it does its job well enough to emulate heat released during combustion and produce meaningful results.

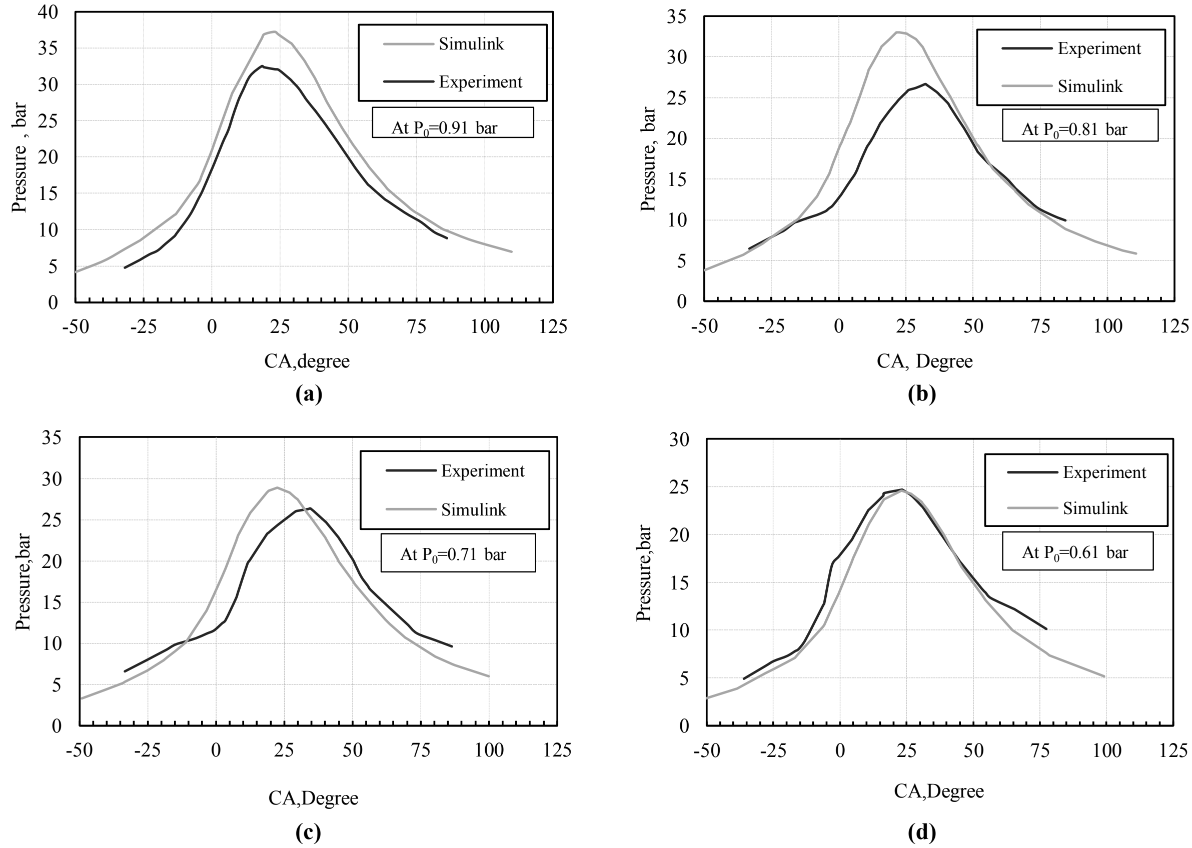


Fig.3. Pressure variation in the cylinder with crank angle at different intake pressures.

The operating experimental condition consist of the experiments with a 4-stroke, single cylinder, 5HP @ 4000 rpm petrol engine (Table 1). The engine runs at wide open throttle (WOT) condition with stoichiometric air-fuel ratio and the spark timing was maintained constant. Even at wide open throttle operation, the manifold pressure would not reach atmospheric value due to various restrictions in flow passage. The effect of efficiency variation was observed with the increase in speed from experiment with simulation result. The indicated efficiency is again higher.

However, the simulation curve follows the trend of decreasing efficiency as the rpm is increased (Fig. 4a).It can be explained from the fact that as speed of the engine (rpm) increases the piston speed also increases while the flame speed increases only marginally. It is mainly due to the increase in turbulence resulting from increased piston speed. The indicated power and brake power variation with speed is shown in Fig. 4 (b). The maximum indicated power for the Simulink model is 36 kW while the experiments had the brake power of about 29 kW for same operating conditions. The indicated power is higher than brake power and also follows the trend. It should be noted that the brake power obtained from experiment is lower due to the friction power lost. The heat released in combustion and the intake pressures at different rpm with experiments are plotted in Figs. 4 (c-d). The unexpectedly higher heat release in experiments could be because of the fact that the fuel intake in engine might be varying with respect to change in rpm. During simulation, fuel quantity remains same at all rpm because it is dependent only on intake pressure. Moreover the fuel-air ratio might be getting affected which is assumed to be constant in the Simulink model.

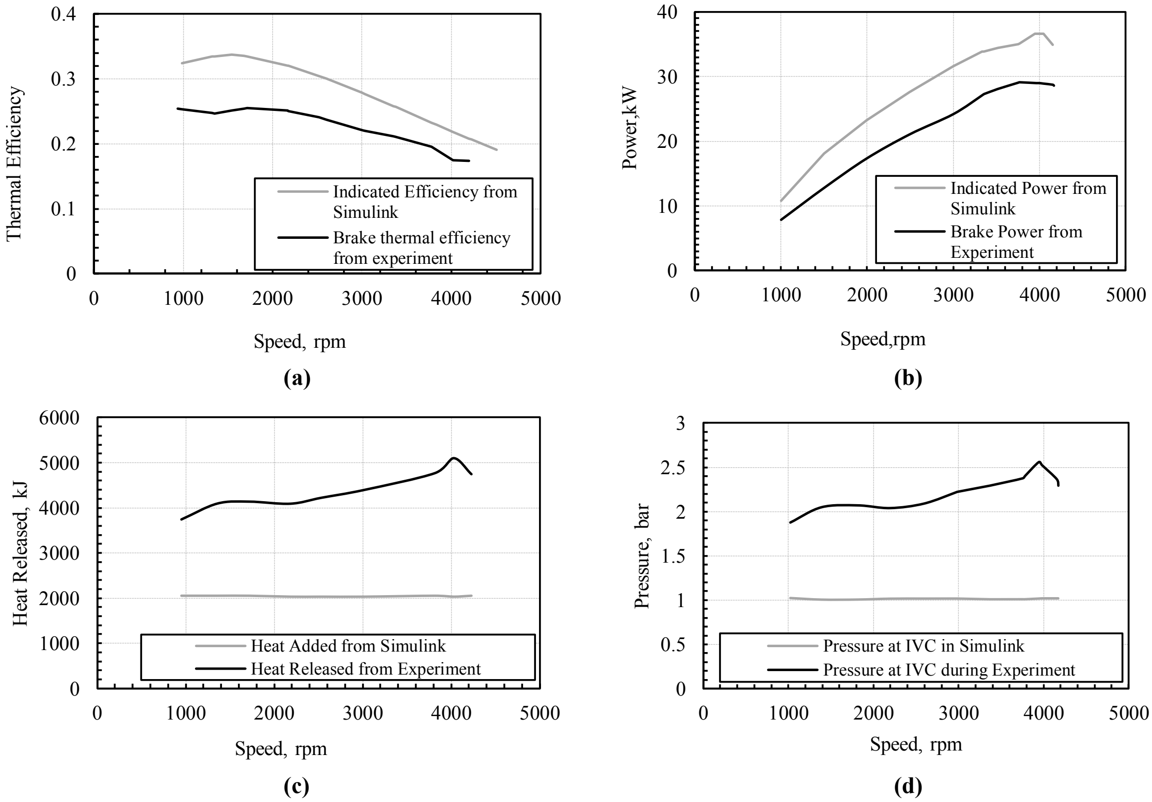


Fig.4. Performance parameter variation with speed(a) Thermal efficiency variation with speed;(b) Power Vs speed;(c)Heat released vs speed;(d)pressure at IVC Vs speed.

3.2. Validation of CFD model

CFD model has been developed for the combustion chamber of a test engine to predict its performance (Fig. 5-a). It is a four-strokes, air cooled, 1.8HP (HONDA make) engine with rated speed of 3600 rpm (Table 1). In this model, one should be able to perform transient simulation with dynamic/moving mesh (Fig. 5-b). For a fixed compression ratio, it uses a module "in-cylinder" using the commercial package Fluent 6.3. The premixed combustion model is used for simulating engine combustion for different speeds.

The CFD model prepared using FLUENT was verified for its results at the condition of rated power 1.8 HP at 3600 rpm (Table 1; Column 2). While creating the physical setup for IC engines, apart from using three user defined functions (UDFs) initialize.c, laminar flame velocity.c and work.c, several other parameters such as viscous flow $k-\varepsilon$ turbulence model is defined. For the premixed combustion, the properties are defined for laminar flame speed and input for the heat of combustion (4.4×10^{7} kJ for gasoline) and unburned mass fraction (0.0625) for stoichiometric gasoline-air mixture. Also, the location and size of spark plug (2mm radius) are taken for simulation. The other parameters are the spark timing (25° bTDC), 0.001 sec as spark duration (0.001s) and diffusion time (0.01 s).The boundary conditions are defined as the wall temperatures and axis setting defined to have boundary type axis. After the solution is converged result can be plotted from work.txt file obtained.

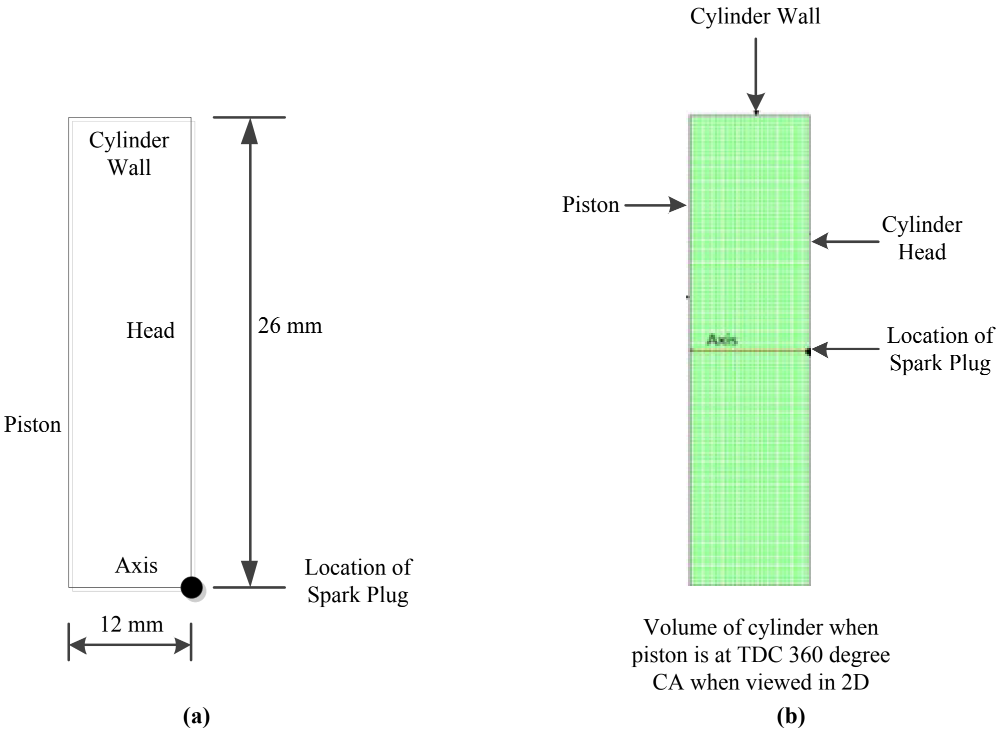


Fig.5. (a) Two dimensional geometry of combustion chamber (only half section above the axis) at TDC; (b) Dynamic mesh used for CFD simulation

The CFD model iterations are carried out to solve the problem. Fig.6 shown is the counter plots of progress variables after sparking. Flame initiation begins at the start of ignition at origin and as piston moves from TDC the flame front moves further towards piston with consuming more and more charge.

Further, the previously validated Simulink model is used for prediction of certain performance parameters for another in-house engine as specified in Table 1 (Column 4). The Simulink model is tuned for the reference parameters and then used to simulation of the engine at different speeds. The reference values for Simulink are

$$r_1 = 4.8, N_1 = 3600\ rpm,\ \emptyset_1 = 1, \theta_{01} = -25^0\ and\ \Delta\theta_1 = 140^0.$$

The CFD model will be more accurate as it calculates the flow at every point and then predicts combustion process. Simulink model which emulates the combustion could then be compared with results of CFD model.

Operating experimental conditions of 1.3 kW single cylinder, air-cooled, variable speed petrol engine having 3000 rated rpm was used for utilization to alternative fuels. A magneto ignition system was used on the crank shaftof the engine for supply of spark current to the spark plug installed at the engine head.

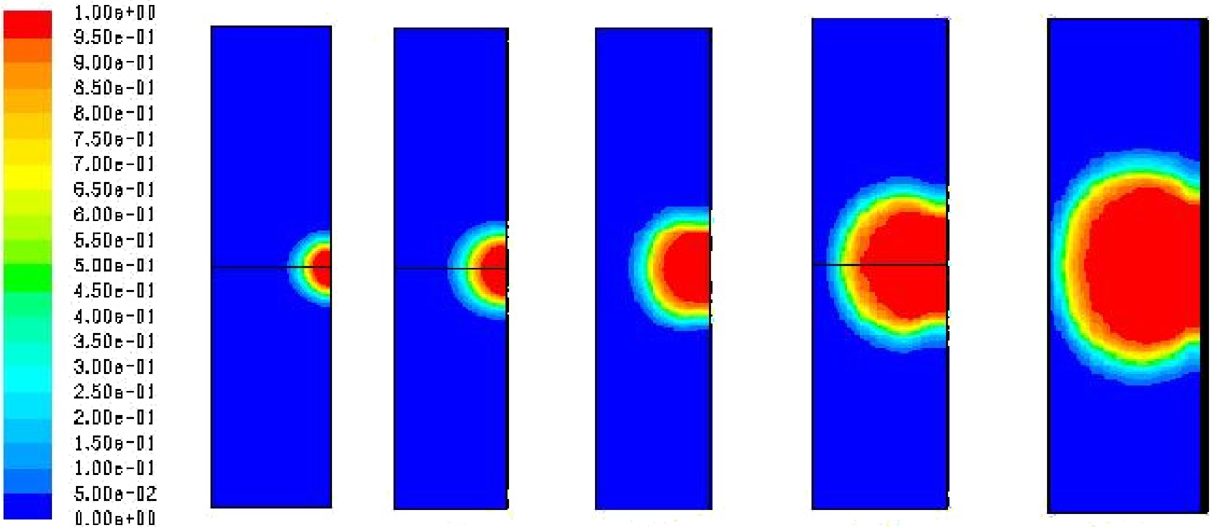
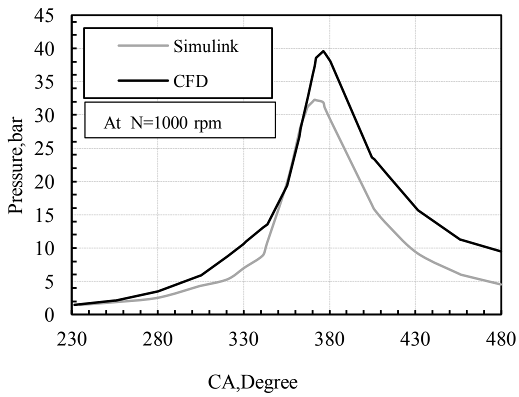
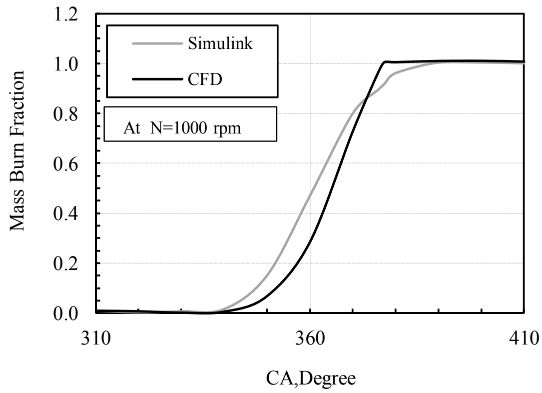


Fig.6.Counter plots for progress variables from 335-375^{0}CA

In Figs 7(a-c), it can be observed that the pressure curves of the Simulink and Fluent model do not match very well. At low speed 1000 rpm, the maximum pressure recorded by CFD model and relatively large error observed with lower speed (≥ 8%).But with increase in speed the variations reduces and mean error of pressure prediction remains below 3%.The mass burn fraction of the two models show Figs 7(d-f) almost same combustion duration, however there is slight mismatching of the slope and curvatures.



(a)



(d)

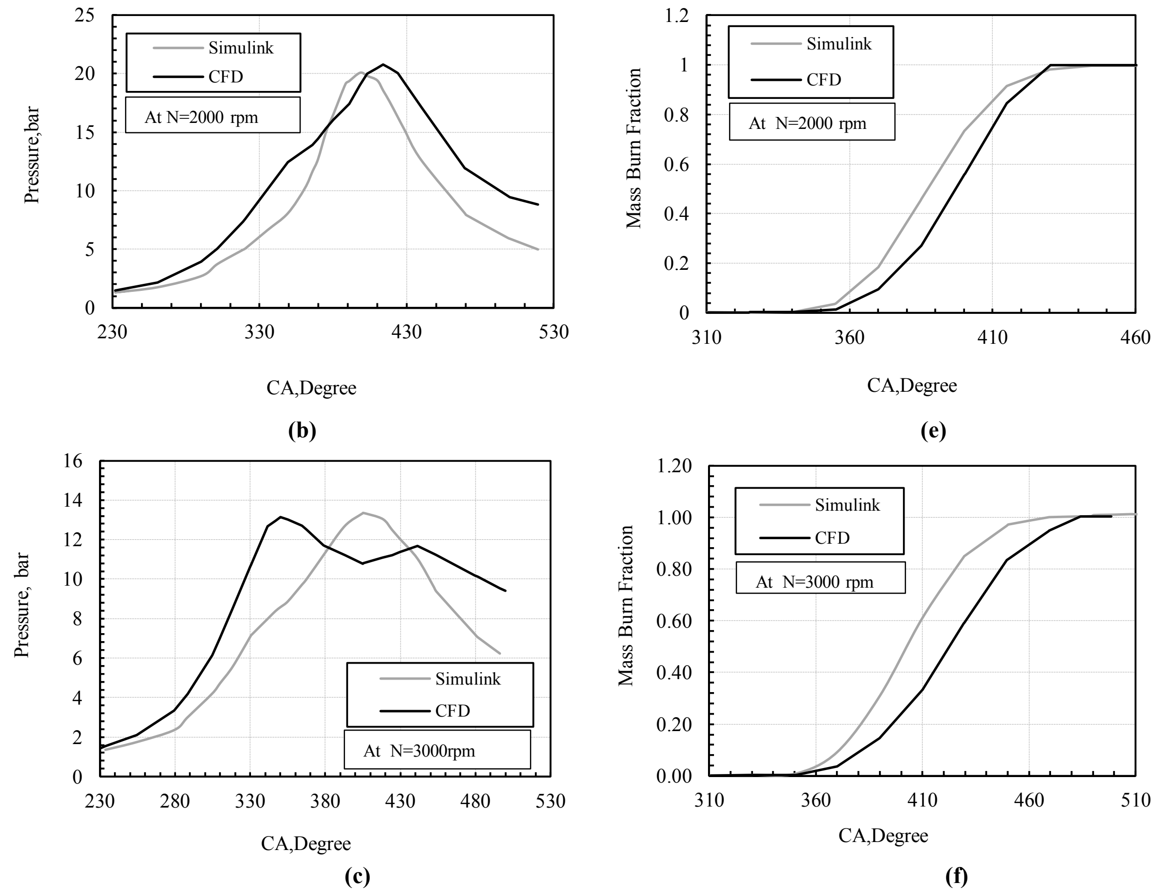


Fig.7. Comparison of Simulink and CFD models for experimental test engine at different speeds.

This may be due to discrepancy in mass burn fraction during CFD simulation. The experimental data of brake power (Fig 8) at different speeds for both the models are compared with experimental results and are found to be close agreement.

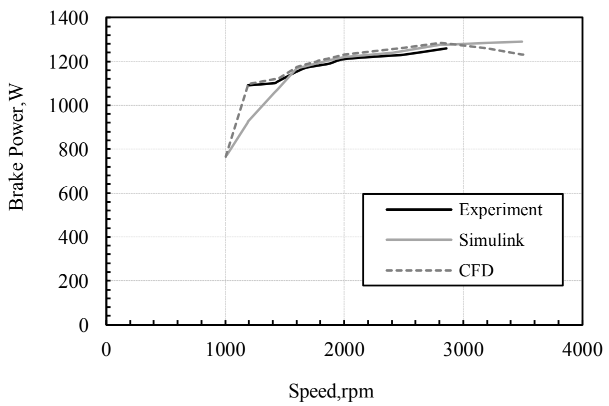


Fig.8.Brake power variation with speed for experiment test set up

4. Conclusion

In this paper, an attempt has been made to address all the thermodynamic processes occurring in engine through appropriate and suitable relations. A single-zone zero-dimensional Simulink model presents good interface to user to carry out parametric studies for prediction crucial parameters prior to the experiments. Side by side, the CFD analysis is also performed using commercial package (Fluent). In the present study, the Simulink model is validated from the experimental data and found to be well agreement with the reported results. The Simulink analysis is performed to predict the maximum pressure after combustion. The deviation of mean pressure is about 6% higher as compared to experiment. However, it reduces to 1% with reduction in intake charge pressure from 0.91 bar to 0.61 bar. During experiment, when the speed increases from 1000 to 4000 rpm, the heat release rate rises from 3800 kJ to 4800 kJ, whereas simulation shows constant heat release rate of 2000 kJ for all speeds.

After successful validation of Simulink model, the numerical analysis was carried out at rated rpm of engine (1.8 HP @ 3600 rpm) using CFD and subsequently both the results are compared. Here, the pressure curve shows slight variation. At low speed 1000 rpm, the maximum pressure recorded by CFD model showed relatively large deviation ($\geq$ 8%). But with increase in speed (at 3000 rpm), the variations reduces and deviation remains below 3%. Mass burn fraction shows almost same trend of combustion duration. The experimental brake power shows close match with Simulink and CFD results.

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