International Journal of Current Engineering and Technology
ISSN 2277 - 4106

Research Article

Simulation of Performance Parameters of Spark Ignition Engine for Various Ignition Timings

Varunkumar Singh[Ȧ*]

[Ȧ]Department of Mechanical Engineering, Veermata Jijabai Technological Institute, Mumbai, India

Accepted 25 November 2013, Available online 15 December 2013, **Vol.3, No.5 (December 2013)**

Abstract

The combustion process in a spark ignition engine is a progressive reaction, and thus certain factors such as the timing of ignition, combustion duration, the end of combustion etc. have considerable effect on performance of an engine. These parameters affect the maximum pressure that is developed inside the cylinder during combustion. Thus the output of the engine can be controlled to a certain extent by variation of important engine parameters mentioned. The effect of ignition timing on parameters such as maximum cylinder pressure, power, torque, fuel consumption is studied in this paper. Thus the optimum value of ignition angle is simulated by single zone model and verified with experimental data.

Keywords: *Ignition timing, peak cylinder pressure, combustion duration, FMEP, BSFC.*

1. Introduction

When the ideal Otto cycle is considered, one of the prime assumptions made is that the combustion process is instantaneous and begins at TDC. But in an actual engine, combustion is a progressive phenomenon. Thus if ignition begins when the piston reaches the TDC, the maximum pressure achieved by the engine is reduced. This is because by the time combustion is completed the piston has already surpassed the TDC. Also when the combustion is started at time much before the piston reaches top dead centre, there is inadequate compression which again results in loss of work. Thus the selection of a proper ignition timing is the compromise between the maximum pressure achieved and loss of work. There has to be a certain value of ignition angle before TDC when maximum power will be achieved.

In the present work, the modelling of IC engine is done on Matlab®. The model aims to plot the PV curve for 4 stroke spark ignition engine using simulation models provided by V. Ganeshan(V. Ganeshan , Computer Simulation of Spark Ignition Process, 1996). The Fuel Air cycle of 4 stroke IC engine is modelled considering the progressive combustion and including the effects of intake manifold temperature loss, effect of residual gases in clearance volume, heat transfer loss to cylinder walls and the loss of work due to friction which arises due to speed. The engine specifications, given in Table.1, are used for simulation are same as the one used in an experiment to study the effect of different ignition timing on SI engine parameters performed by J. Zareei & A.H. Kakaee (. Zareei & A.H. Kakaee,2013).

The ignition angle is varied from 40^o bTDC to 10^o aTDC and the value for which the optimum performance could be achieved is concluded.

2. SI Engine Cycle Simulation

2.1 *Engine Specification*

Table 1 Engine Specifications

Engine type	TU3A
Number of strokes	4
Number of cylinders	4
Cylinder diameter, mm	75
Stroke, mm	77
Compression ratio	10.5:1
Maximum power, kW	50
Maximum torque, N-m	160
Maximum speed, rpm	6500
Displacement, cc	1360
Fuel	97-octane

For experiment and simulation, the engine is assumed to run at 3400 RPM and fully open throttle.

2.2 *Compression Process*

The P-V curve for compression process is determined from the equation

$$P_1V_1^k=P_2V_2^k. \quad (1)$$

where,
P = pressure.

*Corresponding author: **Varunkumar Singh**

V = cylinder volume.
k = polytropic coefficient.
Here, V is volume of cylinder at certain value of crank angle.

$$V(\emptyset)=V_{disp}\left[\frac{r}{r-1}-\frac{1-\cos\emptyset}{2}+\frac{L}{S}-0.5\sqrt{\left(\frac{L}{S}\right)^2-\sin^2\emptyset}\right] \quad (2)$$

where,
r = compression ratio
∅ = crank angle
L = connecting rod length
S = stroke
V_{disp} = displacement volume

2.3 Combustion Process

Combustion Model

The combustion process is modelled as progressive combustion. The pressure change for different crank angle values as combustion progresses is calculated by equation (V. Ganeshan , Computer Simulation of Spark Ignition Process, 1996):

$$\Delta P = \Delta Pp + \Delta Pc \quad (3)$$

where,
ΔP = pressure change during small interval of time
ΔP_P = pressure change during that time as a result of piston movement
ΔP_c = pressure change during that time interval as a result of combustion

The first part of the equation is derived by differentiating the logarithm of equation :

PV^k= constant.

where, k = Cp/Cv.

Thus $$\Delta P = -Pk\frac{\Delta V}{V} \quad (3a)$$

The second part of the equation is:

$$\Delta Pc = (P_3 - P_2)\frac{V_{tdc}}{V}\Delta n \quad (3b)$$

where,
V_{tdc}= clearance volume
P_3= pressure after combustion
P_2= pressure after compression
n= mass fraction of burned gas to total gas
and,

$$\Delta V = V(\emptyset + \Delta\emptyset) - V(\emptyset) \quad (3c)$$

where,
V = cylinder volume
∅ = crank angle

2.4 Expansion Process

The P-V curve for expansion process is determined from the equation:

$$P_3V_3^k=P_4V_4^k. \quad (4)$$

where,
P_4 = pressure before exhaust
V_3 = clearance volume
V_4 = total volume of cylinder

The working conditions during simulation are attempted to bring as close as possible to actual working condition by including following:

1) *Intake Manifold Temperature Loss*

In SI engines, the fuel air mixture is prepared in carburetor. During this process of mixing, the fuel evaporates by taking the latent heat of evaporation from the air. This results in drop in the temperature of the intake air fuel mixture. The temperature in manifold as a result of drop in temperature is given by:

$$T_m = T_a - \Delta T \quad (5)$$

where,
T_a = ambient air temperature.
T_m = temperature of mixture in manifold
ΔT = temperature drop due to fuel evaporation and is calculated as (V. Ganeshan , Computer Simulation of Spark Ignition Process, 1996).

$$\Delta T=\frac{h_{fg}}{\left[\frac{A}{F}\right]C_{pa}+C_{pf}} \quad (6)$$

where,
C_{pa} and C_{pf} denote the specific heats of air and fuel
h_{fg} is the heat of vaporization of fuel.

2) *Temperature Increase due to Exhaust Gas Residual*

During the intake valve, the air fuel mixture mixes with the residual exhaust collected in the clearance volume. Since the exhaust gas is at a high temperature compared to intake mixture, the temperature of the resultant mixture is given by (V. Ganeshan , Computer Simulation of Spark Ignition Process, 1996):

$$T_{mix} = \frac{r\,T_m}{r-1+(T_m/T_5)} \quad (7)$$

where, T_5 is the temperature of exhaust gas.

3) *Heat Transfer*

The pressure change Δp due to heat transfer is (V. Ganeshan, Computer Simulation of Spark Ignition Process, 1996):

$$\frac{\Delta p}{p} = \frac{h_c A(T_w - T)}{M C_V T}\Delta T \qquad (8)$$

where,
h_c= coefficient of heat transfer calculated by Annand's Equation(Annand WJD,1963).
A = interior surface area of cylinder volume.
T_w = interior surface temperature
M = mass of working fluid
Cv = working fluid specific heat
T = working fluid temperature

4) *Calculation of FMEP*

The Frictional Mean Effective Pressure is the measure of the pressure lost due to friction in reciprocating parts.

FMEP for a given speed N is given by following relation (Heywood, J., Higgins, J., Watts, P., and Tabaczynski, R,1979):

$$FMEP=0.97+10.5\left(\frac{Nrpm}{1000}\right)+0.05\left(\frac{Nrpm}{1000}\right)^2 \qquad (9)$$

5) *Effect of Ignition Timing on Combustion Duration:*

The combustion duration for the required engine parameters is calculated using the following empirical relation (Hakan Bayraktar, Orhan Durgun,2004):

$$\Delta\theta_b\,(r,Nrpm,\emptyset,\theta_s)=f_1(r)f_2(Nrpm)f_3(\emptyset)f_4(\theta_s)\Delta\theta_{b1} \qquad (10)$$

where,
$\Delta\theta_b$= angle of combustion and $\Delta\theta_{b1}$ *is* known angle of combustion for a set of parameters.
θ_s= spark advance angle
$\emptyset$ = equivalence ratio

$$f_1(r)=3.2989-3.3612(r/r_1)+1.0800\left(\frac{r}{r_1}\right)^2$$

$$f_2(Nrpm)=0.1222+0.9717\left(\frac{Nrpm}{Nrpm1}\right)-5.0510\times10^{-2}(Nrpm/Nrpm1)^2$$

$$f_3(\emptyset)=4.3111-5.6383\left(\frac{\emptyset}{\emptyset_1}\right)+0.2304(\emptyset/\emptyset_1)^2$$

$$f_4(\theta_s)=1.0685-0.2902\left(\frac{\theta_s}{\theta_{s1}}\right)+0.2545(\theta_s/\theta_{s1})^2$$

Complete Simulation Process

The algorithm for the complete simulation process is given in Fig. 1.

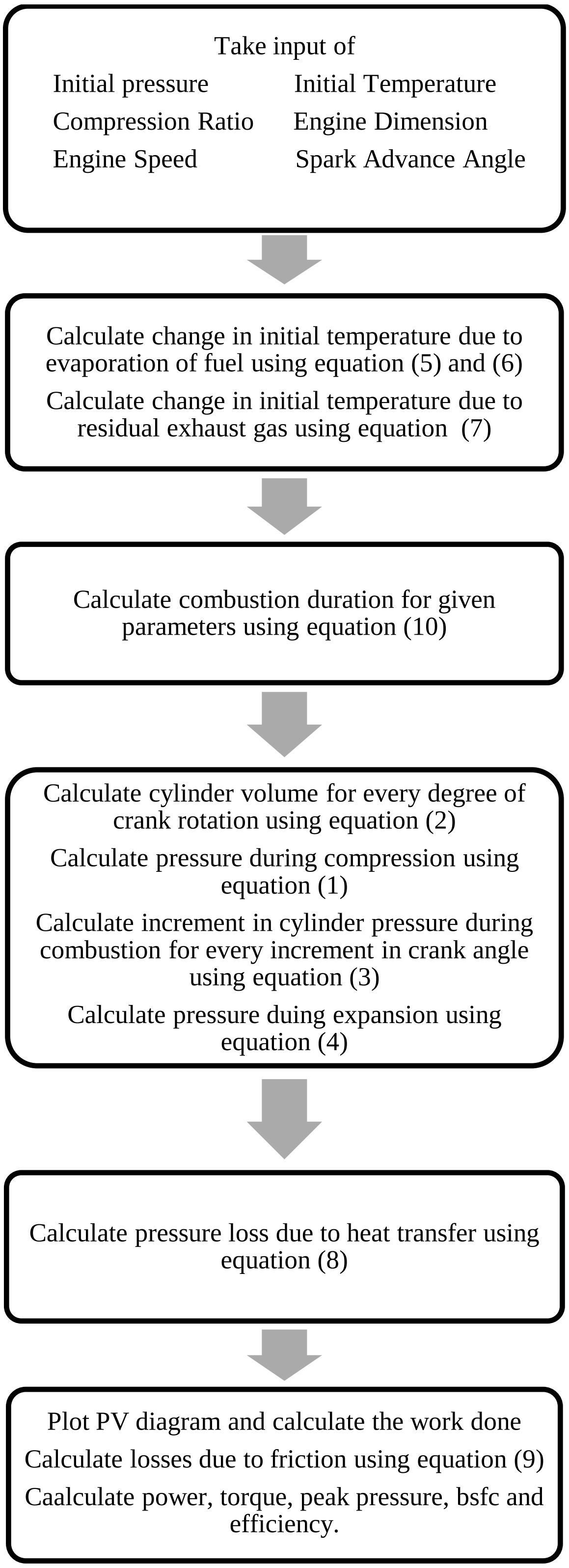

Figure 1 Simulation Algorithm

3. Results and Discussion

In this paper, the analysis is done by plotting the PV diagram for the given engine specifications and calculating the work done i.e. area of PV diagram using Matlab®. Different PV diagrams are plotted by varying the ignition timing and their respective combustion durations.The results are used to calculate various engine parameters and validated with the experimental values.

3.1 Peak Cylinder Pressure

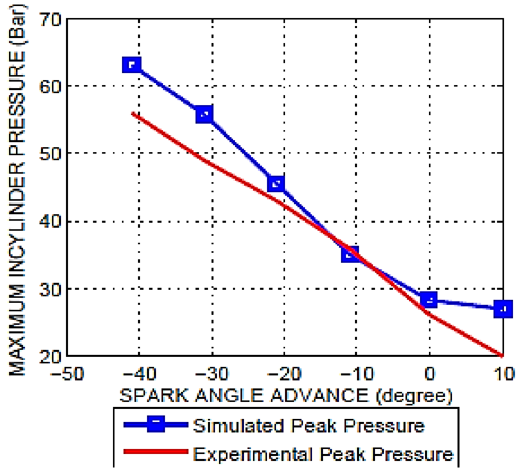

Figure.2 Effect of Spark Advance Angle on Maximum Cylinder Pressure.

In Fig.2, it is observed that maximum in-cylinder pressure increases with spark advance. When ignition is started at TDC, by the time maximum pressure is achieved by combustion process the piston has already crossed the TDC position and expansion process is started. Thus the maximum pressure that can be achieved is reduced .This reduces the mean effective pressure as a result of which the work done is decreased. With increase in spark advance, the duration between TDC and maximum combustion pressure is decreased and overall in-cylinder pressure increases.

3.2 Power

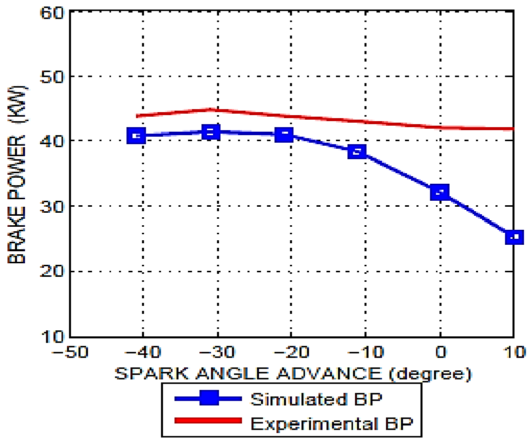

Figure 3 Effect of Spark Advance Angle on Brake Power

For ideal engine, the maximum power will be generated when there will be instantaneous combustion at TDC. But since the combustion is progressive, maximum pressure is not achieved instantaneously, and thus, spark ignition angle is advanced to get maximum pressure. When ignition is started at TDC, expansion process is already started. Thus the work generated from expansion of that region is lost. Also, if ignition is started much before the TDC, the energy from combustion is utilized for opposing the compression movement of piston. From the plotted result in Fig. 3, it can be seen that the maximum power is produced when spark advance angle is 31^{o}. Also simulation and experimental values show a similar variation with change in the spark advance angle.

3.3 Torque

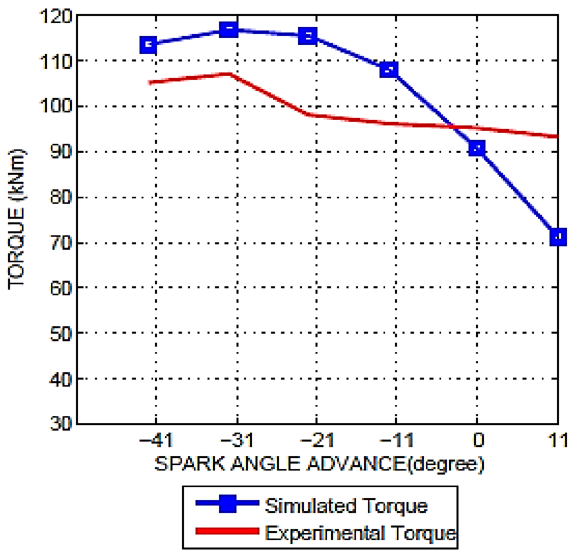

Figure.4 Effect of Spark Advance Angle on Torque

Torque can be calculated from power by the relation, T = P/ω where, $\omega = 2\pi N/60$. Variation of torque with ignition angle is similar to that of power. The variation of power generated and torque for different values of ignition angle is plotted. The experimental and simulated values are compared. In Fig. 4, it is observed that torque increases with spark advance till a certain value, and maximum braking torque is achieved at spark advance of 31^{o}. After this, with further increase in spark advance there is a decrease in the torque.

3.4 Brake Specific Fuel Consumption (BSFC)

BSFC decreases with spark advance. This could be attributed to the increase in spark advance. There is an increase in power output till a certain extent. Also, since combustion process is advanced, the exhaust temperature decreases and loss of energy is reduced. Thus losses are reduced which gives more power for same amount fuel. Thus BSFC decreases with spark advance as shown in Fig. 5.

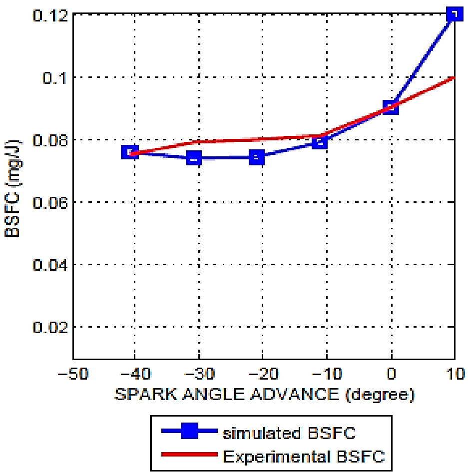

Figure 5 Effect of Spark Advance Angle on BSFC for unity equivalence ratio.

Conclusion

From the simulated and experimental results, it can be concluded that the ignition angle can be a parameter to control the performance of the engine. Also selection of an ignition angle is based on optimum performance. From Fig. 2, it could be seen that high ignition advance gives maximum pressure in cylinder, but performance parameters decrease beyond a certain value. One reason for this could be surpassing of the knock limit pressure for the fuel. For the given engine, the optimum performance is observed for a spark advance angle of 31^{o}.

References

V. Ganeshan (1996) , Computer Simulation of Spark Ignition Process, ISBN 81 7371 015 5 .Chapters 6,7,8,9.

J. Zareei ,A. H. Kakaee (2013), Study and the Effects of Ignition Timingon Gasoline Engine Performance and Emissions. *Eur. Transp. Res. Rev*.,5:109–116.

Annand WJD (1963), Heat Transfer in the Cylinders of Reciprocating Internal Combustion Engines. *Proc. Inst. Mech. Eng.*;177:973–90.

Heywood, J., Higgins, J., Watts, P., and Tabaczynski, R. (1979), Development and Use of a Cycle Simulation to Predict SI Engine Efficiency and NOx Emissions, SAE Technical Paper 790291,doi:10.4271/790291.

Hakan Bayraktar, Orhan Durgun (2004), Development of an Empirical Correlation for Combustion Durations in Spark Ignition Engines. *Energy Conversion and Management*, 45 (2004), pp. 1419–1431.