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Theoretical Analysis of a Spark Ignition Engine by the Thermodynamic Engine Model

Sung Bin Han
Department of Mechanical & Automotive Engineering, Induk University
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Abstract

Recent engine development has focused mainly on the improvement of engine efficiency and output emissions. The improvements in efficiency are being made by friction reduction, combustion improvement and thermodynamic cycle modification.

Computer simulation has been developed to predict the performance of a spark ignition engine. The effects of various cylinder pressure, heat release, flame temperature, unburned gas temperature, flame properties, laminar burning velocity, turbulence burning velocity, etc. were simulated. The simulation and analysis show several meaningful results. The objective of the present study is to develop a combustion model for a spark ignition engine running with isooctane as a fuel and predicting its behavior.

Key words : Spark ignition engine, thermodynamic engine model, compression, combustion, expansion, one-dimensional model, two-zone combustion model

1. Introduction

The engine combustion process is exceedingly complex. Even in the conventional spark-ignition engine, where under many operating modes the fuel and air can be treated as premixed, the combustion process is initiated in a three-dimensional time-varying turbulent flow, concerns a fuel which is a blend of hundreds of different organic compounds whose combustion chemistry is poorly understood, and takes place in a space confined by the combustion chamber walls whose shape varies with time and whose wall directly influence aspects of the process.

A number of engine combustion model classifications have been proposed. The most useful classification follows from one proposed by Hyewood [1]. Its utility stems from the fact that different classes of combustion model, because of their formalism, are generally useful in examining different kinds of combustion related engine problems. These categories of combustion model are: 1) zero dimensional (sometimes called thermodynamics) model, 2) quasi dimensional (sometimes called entrainment) models, and 3) multidimensional (sometimes called detailed) models.

Zero dimensional and quasi dimensional models are structured around a thermodynamics analysis of the contents of the engine cylinder during the engine operation cycle [1].

Simulation of internal combustion engine processes involves, developing a combustion model using the combination of assumptions and equations for predicting the engine performance during the open period which involves suction and exhaust strokes and the closed period which comprises compression, combustion, and expansion stroke [2].

The objective of the present study is to develop a combustion model for a spark ignition engine run-

†To whom corresponding should be addressed.
Department of Mechanical & Automotive Engineering, Induk University, 12 Choansan-ro, Nowon-gu, Seoul 139-749, Korea.
Tel : 02-950-7545 E-mail : sungbinhan@induk.ac.kr

ning with isooctane as a fuel and predicting its behavior.

2. Thermodynamic Engine Model

Thermodynamic model of the gasoline fueled engine is considered the closed system that both inlet and exhaust valves remain closed. The important events such as compression, combustion, and expansion take place during this period. The engine is reduced to a thermodynamic system, which consist of a homogeneous mixture of air, gasoline, and residual gas from the previous cycle. The boundaries of the system are the cylinder walls, cylinder head, and top of the piston head. Work is added to or taken from the system through the motion of the piston and heat is transferred to or from the system through the boundary surfaces [2].

Thermodynamic models can be classified into two groups; one-zone model and multi-zone model. In one-zone model, the cylinder charge is assumed to be uniform in both composition and temperature and the first law of thermodynamics is used to calculate the mixture energy. One-zone model represents the unburned mixture ahead of the flame and one the burned mixture behind the flame.

In two-zone model, the cylinder mixture is divided into burned and unburned zones, which are separate from each other by a surface of dis continuity. The composition and temperature of the burned and unburned gases are different and the pressure is uniform through out the combustion chamber. Two-zone model is used to calculate the mass fraction burned profile from measured cylinder pressure data [3].

In this model, the following assumptions have been used.

- The burnt and unburned gases are assumed to be ideal and non-reacting. The heat transfer from the burned to unburned gases is assumed to be negligible. The pressure is constant for both the zones. There is assumed to be no dissociation in the unburned gases prior to combustion.

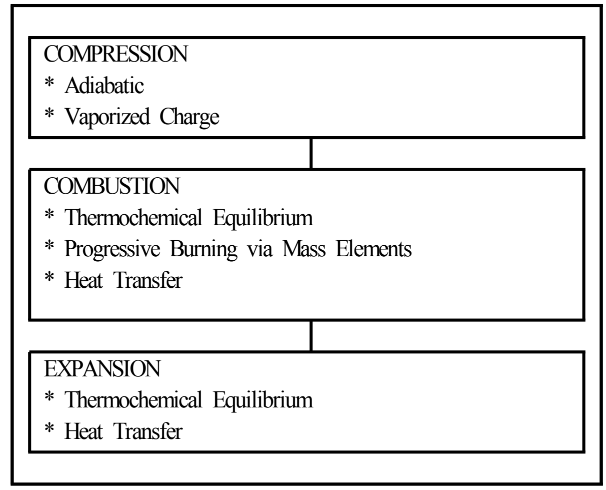

Figure 1. Block diagram of thermodynamic model.

- Cylinder volume at any instant consists of burned and unburned zones separated by a thin infinitesimal flame front. Flame propagates in a spherical pattern.
- The rate of heat transfer from gas to the wall depends on instantaneous heat transfer coefficient, concerned surface area, and difference in temperature between the gas and wall.
- The charge in the cylinder at any instant consists of fuel-air mixture and residual gases. Ideal gas equation is assumed to be valid for the mixture of gases.

Figure 1 shows the general structure of the thermodynamics model.

Thermodynamic engine model is based on the first law of thermodynamics expressed as [4, 5]:

$$dU = \delta Q - \delta W + h_{in} dm_{in} - h_{out} dm_{out}$$

The governing equation for the burned and unburned gas zones can be written as:

$$m = m_u + m_b$$

$$\frac{dm}{d\theta} = \frac{dm_u}{d\theta} + \frac{dm_b}{d\theta}$$

$$V = V_u + V_b$$

where the subscripts u and b denote the unburned and burned gases, respectively.

The following system of equations:

$$m_u C_{Vu} \frac{dT_u}{d\theta} = -p \frac{dV_u}{d\theta} - \frac{dQ_u}{d\theta} + h_u \frac{dm_u}{d\theta}$$

$$m_b C_{Vb} \frac{dT_b}{d\theta} = -p \frac{dV_b}{d\theta} - \frac{dQ_b}{d\theta} + h_b \frac{dm_b}{d\theta}$$

$$Q_u = h A_u (T_w - T_u)$$
$$Q_b = h A_b (T_w - T_b)$$

The unburned temperature is determined using the isentropic relationship in following equations:

$$T_u = T_{uo} (P/P_o)^{(k_u - 1)/k_u}$$
$$P_o = P_m [V_i / V(\theta_o)]^{k_u}$$

In the model, heat from combustion is supplied using a Wiebe function:

$$x = 1 - \mathrm{Exp}\left[-a\{(\theta - \theta_o)/\Delta\theta_b\}^{m+1}\right]$$

$$\frac{dx}{d\theta} = \frac{(m+1)a}{\theta_b{}^{m+1}} (\theta - \theta_o)^m \mathrm{Exp}\left\{-a\left(\frac{\theta - \theta_o}{\theta_b}\right)^{m+1}\right\}$$

With a=5 and m=2, θ_0 is the spark time at the beginning of combustion in crank angle and $\Delta\theta$ is the total combustion duration (x_b = 0 to x_b = 1).

The heat release rate is given by:

$$\frac{\delta Q}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta}$$

$$\frac{\delta Q}{d\theta} = P \frac{dV}{d\theta}\left(1 + \frac{C_{vb}}{R_b}\right) + \frac{C_{vb}}{R_b} \cdot \quad V \frac{dP}{d\theta}$$

The specific heat ratio is estimated using NASA interpolations of specific heats at constant pressure. The interpolating polynomials are given on a form [5, 6]:

$$\frac{C_p(T)}{R} = a_1 T^{-2} + a_2 T^{-1} + a_3 + a_4 T + a_5 T^2$$
$$+ a_5 T^2 + a_6 T^3 + a_7 T^4$$

The specific heat ratio can then be calculated

$$\gamma = \frac{C_p}{C_p - R}$$

Pressure is calculated from the equation:

$$\frac{dp}{d\theta} = \frac{T \sum R_i \frac{dm_i}{d\theta} + \sum R_i m_i \frac{dT}{d\theta} - p \frac{dV}{d\theta}}{V}$$

The total volume of the gases inside the cylinder at a particular crank angle is given by the expression:

$$\frac{dV}{d\theta} = \frac{\pi}{4} D^2 r \left(\sin\theta + \frac{1}{2\lambda} sin 2\theta\right)$$

3. Results

Figure 2 shows the cylinder pressure in a spark ignition engine as a function of crank angle. The pressure from this thermodynamics model reveals that it is shown the peak pressure 42 bar at around 18 degree after TDC (top dead center).

The calculation condition for compression, combustion, and expansion predictions is shown in Table 1.

The model assumes that crank angle is simulated. This implies that the diagnostic version of the heat release equation can be used to calculate the quantities of heat released at each crank angle [7, 8].

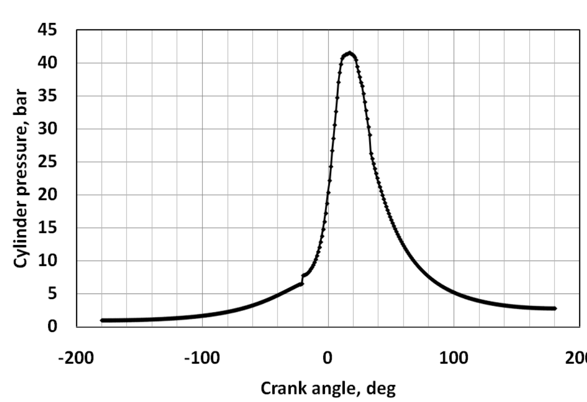

Figure 2. Variation of cylinder pressure with crank angle.

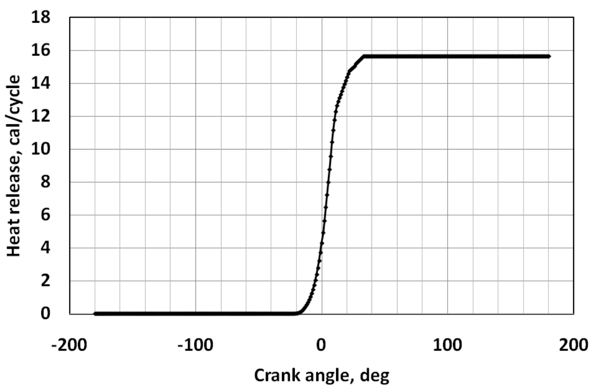

Figure 3. Variation of heat release with crank angle.

Table 1. Calculation condition for simulation.

Items	Specification
Bore x Stroke	90 x 70 mm
Displacement	445 cc
Compression ratio	4.5
Connecting rod	133 mm
Engine speed	1,600 rpm
Spark advance	23° (BTDC)
Air fuel ratio	13:1
Mixture	Isooctane and air

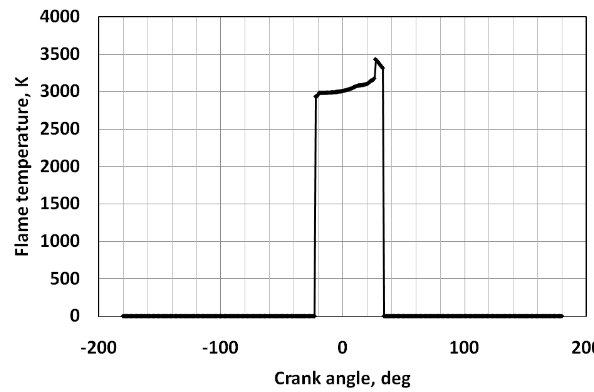

Figure 4. Variation of flame temperature with crank angle.

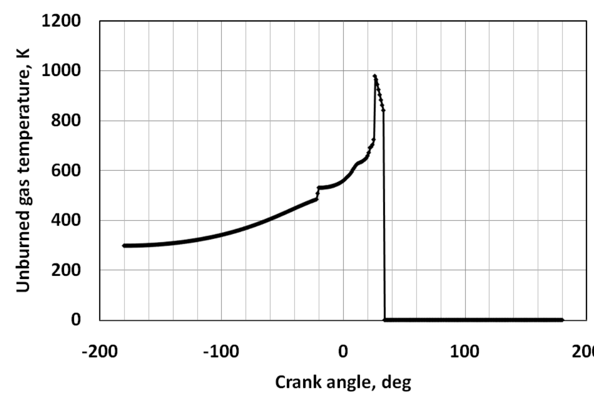

Figure 5. Variation of unburned gas temperature with crank angle.

Figure 3. shows the simulated heat release curve. Under simulated condition, the heat release is about 16 cal/cycle.

During the compression, combustion and expansion, the combustion period is assumed that the spark ignited flame front divides the in-cylinder gas mixture into two zones : the burned and unburned zones. To simplify the two-zone combustion model, the shape of the burned zone is assumed to be a circle centered at the cylinder [9]. For the spark ignition combustion the temperature of the unburned zone is quite different from that of the burned zone as seen in Figure 4 and Figure 5.

At any given flame radius, the geometry of the combustion chamber and the spark plug location govern the flame front surface area, the area of the approximately spherical surface corresponding to the leading edge of the flame contained by the piston, cylinder head, and cylinder wall. The larger this surface area, the greater the mass of fresh charge that can cross this surface and enter the flame zone [3].

The flame is assumed to propagate spherically from the one end of the cylinder. As the flame proceeds from one end to other the traveled portion of cylinder volume was assumed completely burned, while the other as unburned [2].

Figure 6, 7, 8, and 9 show the variation of volume area, flame radius, flame area, and side area with crank angle, respectively.

An important inherent property of a combustible fuel, air, burned gas mixture is its laminar burning velocity. This burning velocity is defined as the velocity, relative to and normal to the flame front, with which unburned gas moves into the front and is transformed to products under laminar flow conditions [3]. Figure 10, and 11 show the variation of laminar burning velocity, and turbulence burning ve-

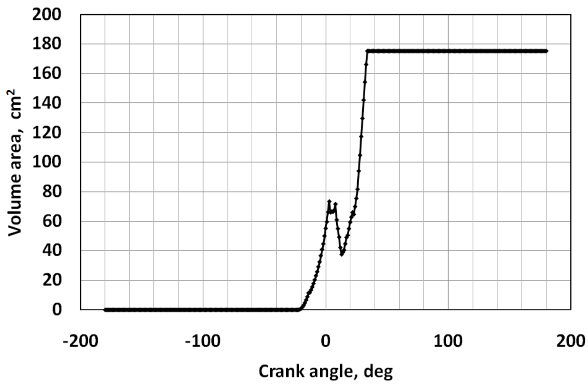

Figure 6. Variation of volume area with crank angle.

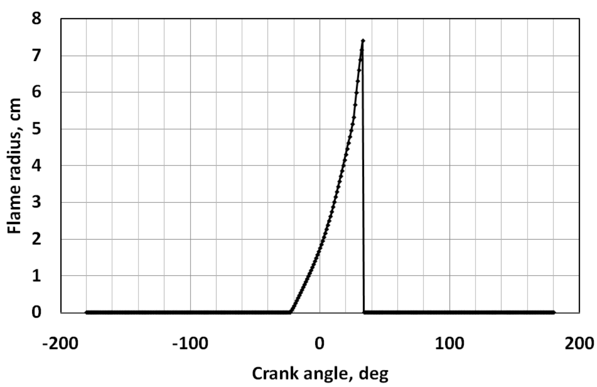

Figure 7. Variation of flame radius with crank angle.

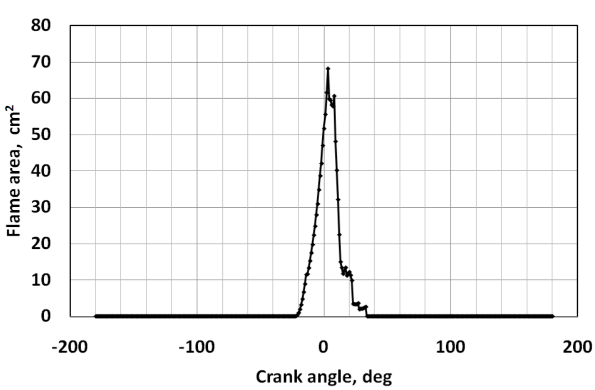

Figure 8. Variation of flame area with crank angle.

locity with crank angle, respectively.

4. Conclusions

The simulation has be come a powerful tool as it saves time and also economical when compared to experimental study. The following conclusions can

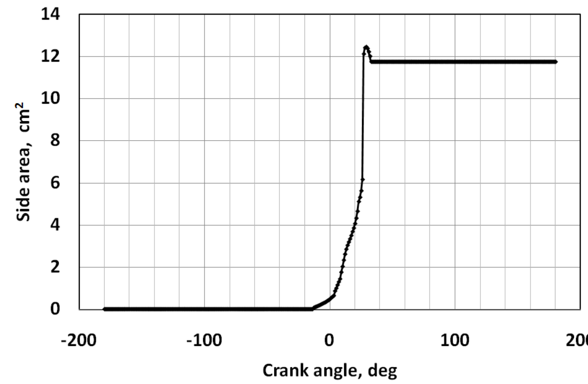

Figure 9. Variation of side area with crank angle.

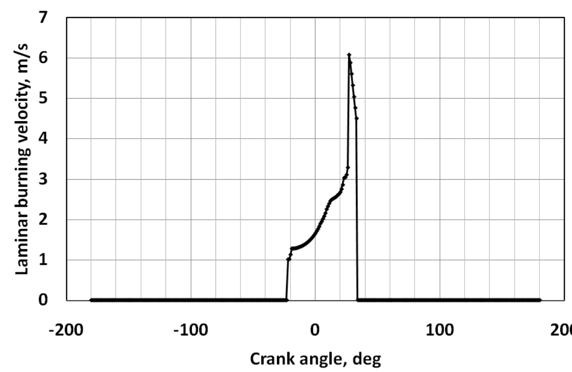

Figure 10. Variation of laminar burning velocity with crank angle.

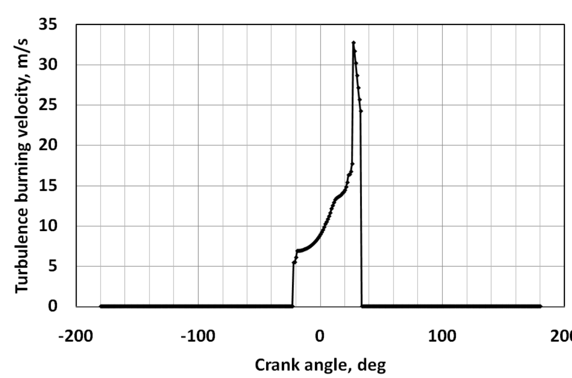

Figure 11. Variation of turbulence burning velocity with crank angle.

be made from above research;

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The effects of various cylinder pressure, heat release, flame temperature, unburned gas temperature, flame properties, laminar burning velocity, turbulence burning velocity, turbulence flame velocity, etc. were simulated. The simulation and analysis show several meaningful results.

Nomenclature

A	flame area
B	cylinder bore
D	diameter of piston
h	heat transfer coefficient
k	specific heat ratio
m	mass of mixture
P	gas pressure
Q	heat of the gases
R	universal gas constant
r	crank radius
S	stroke
SL	laminar flame speed
T	temperature
Θ	crank angle
Υ	specific heat ratio
ρ	density of mixture

subscripts

b	burned
u	unburned

References

1. Heywood, J. B., Combustion Modelling in Reciprocating Engines (Engine Combustion Modeling-An Overview), Plenum Press, 1980, 1-38
2. Sakthinathan, G. P. and Jeyachandran, K., Theoretical and experimental validation of hydrogen fueled spark ignition engine, Thermal Science, Vol.14, No.4, pp.989-1000, 2010.
3. Heywood, J. B., Internal Combustion Engine Fundamentals, Mc-Graw Hill, 1987.
4. Ribeiro, B., Martins, J. and Nunes, A., Generation of Entropy in Spark Ignition Engines, Int. J. of Thermodynamics, Vol. 10, No.2, pp.53-60, 2007.
5. Goodwin, A. R.; Sengers, J. V.; Peters, C. J., Applied Thermodynamics of Fluids, RSC Publishing, 2010.
6. Cengel Y. A.; and Boles, M. A., An Engineering Approach Thermodynamics, McGrow Hill, Second Edition, 1994.
7. Thompson, I. G., Spence, S. W., Thornhill, D., McCartan, C. D. and Talbot-Weiss, J. M., The technical merits of turbogenerating shown through the design, validation and implementation of a one-dimensional engine model, International of Engine Research, Vol.15, No.1, pp.66-77, 2014.
8. Muric, K, Stenlaas, O. and Tunestal, P., Zero-dimensional modeling of NOx formation with least squares interpolation, International of Engine Research, Vol.15, No.8, pp.944-953, 2014.
9. Yang, X. and Zhu, G. G., A control-oriented hybrid combustion model of a homogeneous charge compression ignition capable spark ignition engine. Proc IMech Part D: Journal of Automotive Engineering, Vol.226, No.10, pp.1380-1395, 2012.