에너지공학, 제24권 제3호(2015)
*Journal of Energy Engineering*, Vol. 24, No. 3, pp.61~66(2015)
http://dx.doi.org/10.5855/ENERGY.2015.24.3.061

# Effect of Compression Ratio on the Combustion Characteristics of a Thermodynamics-Based Homogeneous Charge Compression Ignition Engine

**Sung Bin Han**
Department of Mechanical & Automotive Engineering, Induk University
(Received 20 July 2015, Revised 14 September 2015, Accepted 19 September 2015)

## Abstract

Homogeneous charge compression ignition (HCCI) engine combines the combustion characteristics of a compression ignition engine and a spark ignition engine. HCCI engines take advantage of the high compression ratio and heat release rate and thus exhibit high efficiency found in compression ignition engines. In modern research, simulation has be come a powerful tool as it saves time and also economical when compared to experimental study. Engine simulation has been developed to predict the performance of a homogeneous charge compression ignition engine. The effects of compression ratio, cylinder pressure, rate of pressure rise, flame temperature, rate of heat release, and mass fraction burned were simulated. The simulation and analysis show several meaningful results. The objective of the present study is to develop a combustion characteristics model for a homogeneous charge compression ignition engine running with isooctane as a fuel and effect of compression ratio.

**Key words** : Compression ratio, homogeneous charge compression ignition engine, thermodynamic engine model

## 1. Introduction

Homogeneous charge compression ignition (HCCI) engine combines the combustion characteristics of a compression ignition engine and a spark ignition engine. HCCI engines take advantage of the high compression ratio and heat release rate and thus exhibit high efficiency found in compression ignition engines. Also, due to the lean air/fuel ratio, HCCI engines produce low NOx and Particulate Matter (PM) emissions even without the use of after treatment devices. Because HCCI combustion is not the result of a single spark, a traditional flame front does not exist, and in general, a localized high-temperature area is not present [1-3].

In the future, through the development of a HCCI fuel and advancements in the control of combustion phases, an air/fuel ratio and an increase in power similar to those of diesel engines are expected. For now, HCCI combustion applicable for automotive use should utilize a dual-mode engine. In other words, HCCI combustion would be used in low loads to increase fuel efficiency and emission reduction properties, and spark ignition or diesel combustion would be used under high loads to solve the power problem. In engines used for low load and part load conditions, HCCI combustion would be very useful [4].

However, regardless of the existing benefits, the control problems of ignition timing and burning rate are to be solved before the HCCI engines can be practically applied to commercial use. These two challenges are difficult to overcome firstly because HCCI is lack of ignition control mechanism like the spark and direct injection timing control. Secondly

†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

the HCCI combustion is dominated by the chemical kinetics based on the fuel properties therefore the occurrences of misfiring at low load and knocking at high load are usually noted which result in a limited operation range of HCCI engine. Another defect of the HCCI engine is the increase in HC and CO emissions. One of the reasons for such increase in HC and CO emissions is that the increase in EGR rate required for HCCI reduces the cylinder temperature. It is already well known that temperature reduction within the cylinder leads to lower post-combustion oxidation rate and higher HC and CO emissions [5].

In order to achieve considerable progress in reduction of fuel consumption of spark ignition engines, the combination of pressure charging and direct injection promises attractive possibilities. On the other hand pressure-charged engines provide additional advantages in terms of reduced friction losses compared to naturally aspirated engines with equivalent torque. One of the most important factors concerning the possibilities of pressure charging is the compression ratio. Lower ε usually provides advantages in knocking sensitivity but disadvantages in part-load efficiency. The results also depend on the manner of pressure charging: supercharging or turbocharging [6].

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 [7].

The objective of the present study is to develop a combustion model for a homogeneous charge compression ignition engine running with isooctane as a fuel and the effect of compression ratio.

## 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 [7].

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 [8].

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.
- 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 [7].

For the closed system, the basic energy equation combined with characteristic gas equation can be written as:

$$\dot{U} = -\frac{mRT}{V}\dot{V} + \sum Q_i = mC_v\frac{dT}{d\theta}$$

The internal energy at any instant is assumed to be a function of temperature only. It is given by:

$$\frac{dT}{d\theta} = [-P\dot{V} + h_i A_i (T_w - T)]\frac{1}{mC_v}$$

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

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

$$\frac{dV}{d\theta} = \frac{V_{disp}}{2}[\frac{1}{2}\frac{\sin 2\theta}{\sqrt{(\frac{2L}{S})^2 - \sin^2\theta}} - \sin\theta]$$

The method of computation of heat transfer coefficient due to convection is the key factor which controls the order of magnitude of the rate of heat transfer. Heat transfer by convection is given by:

$$Q_i = h_i A_i (T_w - T)$$

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

$$m = m_u + m_b$$

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

$$V = V_u + V_b$$

The following system of equations:

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

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

$$Q_u = hA_u(T_w - T_u)$$

$$Q_b = hA_b(T_w - T_b)$$

In the model, the mass fraction burned profile is often represented by the Wiebe function:

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

where $\triangle\theta$ is the total combustion duration, and a and m are adjustable parameters which fix the shape of the curve [8].

The heat release rate is given by:

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

The specific heat ratio is estimated using NASA interpolations of specific heats at constant pressure. The interpolating polynomials are given on a form [9]:

$$\frac{C_p(T)}{R} = a_1T^{-2} + a_2T^{-1} + a_3 + a_4T + a_5T^2$$
$$+ a_5T^2 + a_6T^3 + a_7T^4$$

## 3. Results and Discussion

The calculation condition for simulation is shown in Table 1.

Figure 1 shows the cylinder pressure in a spark ignition for a homogeneous charge compression engine as a function of crank angle according to the various compression ratios. The pressure from this thermodynamics model reveals that it is shown the peak pressure 40 kg/cm$^2$ at around 10 degree after top dead center and compression ratio 7.0. And it is shown the peak pressure 72 kg/cm$^2$ at around 5 degree after top dead center and compression ratio 9.5.

**Table 1.** Calculation condition for simulation.

| Items | Specification |
|---|---|
| Bore x Stroke | 90 x 70 mm |
| Displacement | 445 cc |
| Compression ratio | 7.0, 7.5, 8.0, 8.5, 9.0 and 9.5 |
| Connecting rod | 133 mm |
| Engine speed | 1,400 rpm |
| Spark advance | 14° (BTDC) |
| Air fuel ratio | 15:1 |
| Mixture | Isooctane and air |

Compression ratio has a great effect on the cylinder pressure. In the cylinder pressure, shown in Fig. 1, it can be seen that the increase of compression ratio results in a sharp increase of the cylinder's top pressure [10].

Figure 2 shows rate of pressure rise changing with compression ratios. From this figure, it could be found that the higher the compression ratio, the higher the top rate of pressure rise.

Figure 3 shows the variation of the cylinder temperature under various compression ratios. It is found With the increase of compression ratio, the cylinder temperature will be increased. The change of cylinder temperature with compression ratio can find some reason from the change of cylinder pressure with compression ratio.

Figure 4 shows that the rate of heat release versus crank angle according to the compression ratios. The heat release equation can be used to calculate the quantities of heat released at each crank angle [11]. Under simulated condition, the maximum rate of heat release is about 4 cal/degree at around 1 degree after top dead center and compression ratio 9.5. And maximum rate of heat release is about 2.6 cal/degree at around 6 degree after top dead center and compression ratio 7.0.

Figure 5 shows the mass fraction burned in a spark ignition for a homogeneous charge compression engine as a function of crank angle according to the various compression ratios. The flame development period is represented in Fig. 1 according to the compression ratios. The period of 10%~90% mass burned was shortened about 5 degree in 7.0 compression ratio as compared with 9.5 compression ratio.

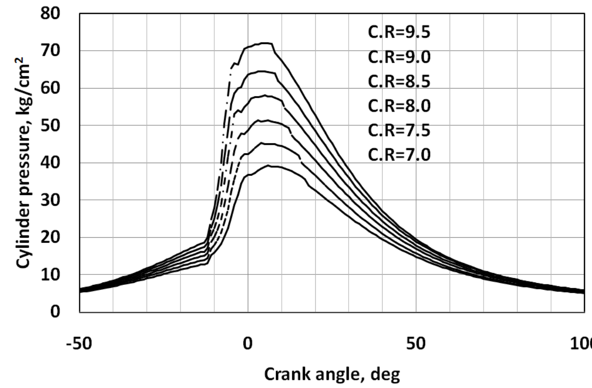


**Figure 1.** Variation of cylinder pressure versus crank angle.

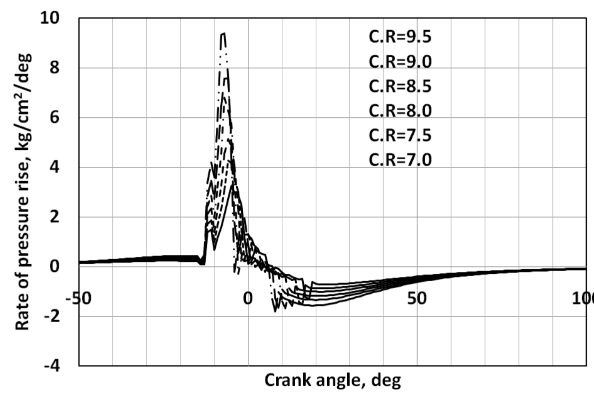


**Figure 2.** Variation of rate of pressure rise versus crank angle.

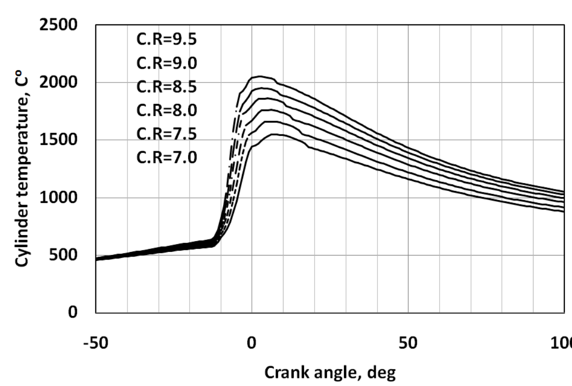


**Figure 3.** Variation of cylinder temperature versus crank angle.

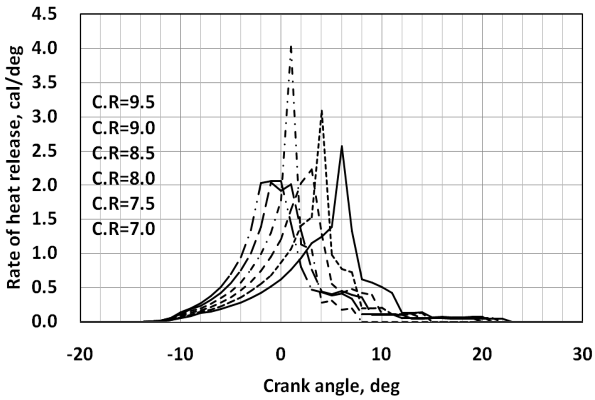


**Figure 4.** Variation of rate of heat release versus crank angle.

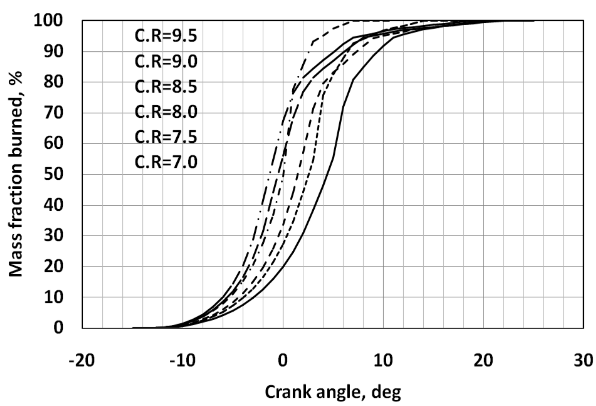


**Figure 5.** Variation of mass fraction burned versus crank angle.

## 4. Conclusions

A comprehensive computer simulation has been developed to predict the performance of homogeneous charge compression ignition engine. The simulation has be come a powerful tool as it saves time and also economical when compared to experimental study. The simulation and analysis show that compression ratio has a great effect on the cylinder pressure, rate of pressure rise, cylinder temperature, rate of heat release and mass fraction burned under simulated condition.

### Nomenclature

| | |
|---|---|
| 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 |
| T | temperature |
| $\Theta$ | crank angle |
| $\triangle$ | combustion duration |

### subscripts

| | |
|---|---|
| b | burned |
| u | unburned |
| o | start of combustion |

## References

1. Campbell, S., Lin, S., Jansons, M. and Rhee, K. T, In-cylinder liquid fuel layers, cause of unburned hydrocarbon and deposit formation in SI engines. *SAE Paper No. 1999-01-3579,* 1999.
2. Christensen, M. and Johansson, B., Homogeneous charge compression ignition with water injection, SAE Paper No. 1999-01-0182, 1999.
3. Choi, G. H., Han, S. B. and Dibble, R. W., Experimental study on homogeneous charge compression ignition engine operation with exhaust gas recirculation, Int. J. Automotive Technology Vol.4, No.3, 195－200, 2004.
4. Han, S. B., Experimental study on liquefied petroleum gas fueled homogeneous charge compression ignition engine, Int. J. Automotive Technology Vol.14, No.6, 837-842, 2013.
5. Au, M. Y., Girard, J. W., Dibble, R., Flowers, D., Aceves, S. M., Frias, J. M., Smith, R., Seibel, C. and Maas, U., 1.9-liter four-cylinder HCCI engine operation with exhaust gas recirculation. SAE Paper No. 2001-01-1894, 2001.
6. Kramer, F., Schwarz, C. and Witt, A., Effect of compression ratio on the combustion of a pressure charged gasoline direct injection engine, SAE Paper

No. 2000-01-0250, 2000.
7. 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.
8. Heywood, J. B., Internal Combustion Engine Fundamentals, Mc-Graw Hill, 1987.
9. Goodwin, A. R., Sengers, J. V., Peters, C. J., Applied Thermodynamics of Fluids, RSC Publishing, 2010.
10. Ma, J., Su, Y., Zhou, Y. and Zhang, Z., Simulation and prediction on the performance of a vehicle's hydrogen engine, International Journal of Hydrogen Energy, Vol.28, pp.77-83, 2003.
11. 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.