Estimation of Flame Propagation in Spark-Ignition Engine by Using Turbulent Burning Model

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ABSTRACT

This paper describes an estimation of the mass fraction burned profiles and flame propagation characteristics in a complex shape combustion chamber of side-valve type spark-ignition engine by applying the turbulent burning model originally proposed by Blizard and Keck. The instantaneous measurements of cylinder pressure, flame front position and wall heat flux were performed and compared with the model calculation. The model is modified to include effects of the density variation of the unburned gas and the gas flow in the chamber, and by adjusting empirical constants, it can successfully simulate the mass fraction burned profile for the engine geometry and the operating conditions which does not violate the spherical flame propagation condition.

INTRODUCTION

In order to increase the fuel efficiency and to reduce the pollution from spark-ignition engines, many investigators [1], [2] have proposed turbulent burning models for these engines to estimate the combustion processes, engine performance and exhaust gas emissions.

The turbulent burning model proposed by Blizard and Keck [3] is based upon the assumption that eddies produced by turbulent shear flow in a chamber are entrained by flame front at some turbulent burning velocity and subsequently burn in some characteristic time. The model equations consist of the mass conservation equation across the turbulent flame front and the eddy burning equation with a characteristic time defined by the laminar flame propagation velocity and the characteristic eddy radius. These equations coupled with the thermodynamic equation [4], [5] which is based on the first law of thermodynamics constitute a complete set of ordinary differential equations for the mass fraction burned and the cylinder pressure, provided that the turbulent flame speed velocity and the characteristic eddy radius (turbulent length scale) are evaluated. The flame geometry is assumed to be spherical with the ignition point as a center. The turbulent flame speed and the eddy radius are given by empirical formulæ with empirical constants. In this model, the turbulent flame speed is proportional to the inlet gas velocity and the eddy size is proportional to the inlet valve lift, and they remain constant during the combustion process. They showed that their experimental results of the measurements of cylinder pressure and flame geometry were well predicted by their model equations for all of their operating conditions. McCuliston et al. [6], Tabaczynski et al. [7]-[9] and Keck et al. [10], [11] developed a further refinement of the model to define exactly the turbulent length scale and the turbulent flame speed assuming that the turbulence in the cylinder is isotropic. Their models, still, restricted themselves to the symmetrical shape combustion chamber.

The present study aims at estimating the mass fraction burned profiles and flame propagation characteristics in a complex shape combustion chamber of side-valve type spark-ignition engine by applying and modifying the turbulent burning model originally proposed by Blizard and Keck. The mass fraction of burned gas and flame radius are measured as a function of crank angle to be compared with the calculated results using a modified version of the model to discuss its applicability on this type of engine because Blizard-Keck model may be applied only for the engines with simpler shaped combustion chamber.

EXPERIMENTAL

Experimental Apparatus

The present experiments are carried out in a four-cycle, side-valve type, single cylinder spark-ignition engine (Bore : 90 mm, Stroke : 70 mm). The relevant engine geometry and operating conditions are shown in Fig. 1 and Table 1 and 2, respectively. The combustion chamber of this engine is divided into the two sections, i.e., a pre-chamber and a main chamber. The area of port connected these divided two chamber can be varied to change a flow field in the main chamber. Three types of chamber are used which have a different area between a pre-chamber and a main chamber. These are called S0, S1 and S4 as decreasing the area. The ignition is performed on the center axis of the main chamber. Fuel is gasoline. The piston crown and cylinder head surface are flat and cylinder head is cooled by water.

Measurements

Gas flow patterns in the combustion chamber are observed by means of the oil film method and they are shown in Fig. 2 (operating condition: motoring). In the combustion chamber S0, gas flow
along the cylinder head surface is found to be very moderate. In the S1 chamber, a gas seems to flow from suction valve to cylinder center and its velocity is higher than that of S0. In the S4 chamber, a gas flows around the center of the chamber axis to form a swirl.

The cylinder pressure is measured by the diaphragm-type pressure transducer in continuous 100 cycles. Data are stored on magnetic tape and the average profile is calculated by digital computer. Typical measured cylinder pressures are shown in Fig. 3. In the S1 chamber, a crank angle when pressure increases is earlier than that of in the S0 chamber and maximum pressure is also higher. In the S4 chamber, a pressure rise in the initial

\[ N = 1300 \text{ rpm} \]
\[ IT = \text{BTDC} \ 25^\circ \]
\[ \phi = 1.0 \]
\[ \eta_v = 0.67 \]

Fig. 3 Pressure diagram for typical conditions

![Pressure diagram](image)

Fig. 4 Location of ionization probes, heat flux sensors and spark plug

![Location diagram](image)

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stage is smaller than that in the S1 chamber but in the late stage pressure rises rapidly to reach a maximum pressure higher than that of S1.

Position of the flame front, local instantaneous heat flux into the chamber wall are measured. The ionization-probes and compact-type wall heat-flux sensors embedded in the cylinder head at locations which are are shown in Fig. 4 are used for these measurements, respectively. The data of the wall heat-flux is utilized for discussing the validity of the thermodynamic model because it demands the exact information about the heat loss to the chamber wall. The local instantaneous heat flux at surface of combustion chamber is calculated by using the Fourier equation from the temperature at a surface and that at a point 0.7 - 0.8 mm deep inside. The details of the heat flux measurement are shown in Ref. [12]. Typical instantaneous heat flux distribution on the cylinder surface is shown in Fig. 5 for the chamber S0 and S4. Instantaneous heat loss of combustion chamber wall is estimated from distribution of heat flux at each crank angle. Because of the intensive swirl generated in the chamber S4, the wall heat flux is larger than that in the chamber S0 and the maximum area lies near the port where gas flows out from the pre-chamber while in the S0 chamber it lies near the ignition point.

The flame arrival time at each measuring point is measured in 20 cycles by each ionization probe. Distributions of flame arrival time are shown in Fig. 6. For each combustion chamber, flame propagates symmentrically around an ignition point plug in the combustion chamber. Dimensionless flame front radius $r_f/r_p$ as function of crank angle is shown Fig. 7. In the chamber S1, flame spreads more rapidly than than S0 because of the disturbed flow field in the chamber. But for the chamber S4, flame movement is very small in the initial stage and in the late stage flame accelerates.

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![Diagram](image)

Fig. 5 Distribution of instantaneous heat flux in the S0(left) and S4(right) chambers

Fig. 6 Distribution of flame arrival time in the S0(left) and S4(right) chambers

Fig. 7 Dimensionless flame radius versus crank angle from spark timing
Mass Fraction of Burned Gas

Mass fraction of burned gas is estimated by thermodynamic analysis using measured pressure and heat loss as a function of crank angle. This thermodynamic analysis involves the following basic assumption:

(i) the total mass in the combustion chamber can be divided into a mass fraction burned and a mass fraction unburned.
(ii) the pressure is spatially uniform throughout the combustion chamber.
(iii) both unburned and burned gases obey the equation of state of a perfect gas.
(iv) unburned gases in the combustion chamber is compressed isentropically.

Thermochemical parameters as a specific heat and a heat of formation are calculated under the assumption that a burned gas is in a chemical equilibrium state and unburned gas in a frozen state. The specific heats of both burned and unburned gas remain constant throughout the process. Experimentally obtained heat loss is utilized in this calculation. The results of these experimentally obtained mass fraction burned will be shown in RESULTS AND DISCUSSIONS with the calculated one by the turbulent burning model.

TURBULENT BURNING MODEL

Background

Combustion processes in a spark-ignition engines are very complex phenomena because they include unsteady flame propagation processes in a turbulent flow field with a high turbulent intensity. But, fortunately, the turbulent flow field in the combustion chamber in some engines may be regarded as homogeneous and isotropic so that the theoretical viewpoints developed by the theory of the isotropic turbulence can be used in estimating turbulent length scales and intensity as a function of the macroscopic characters of the flow field.

Blizard and Keck [3] have proposed a model based on the assumption that eddies entrained in a combustion zone burn with some characteristic time which is dependent on the length scale of turbulence and the laminar flame speed. The turbulent flame speed is described by the empirical formula in terms of an engine speed and the length scale in terms of a valve lift and a compression ratio at ignition. In addition to these fundamental physical assumptions, it is necessary to assume a flame geometry to obtain the relationship between a surface area of the flame front and a volume enclosed by the flame front. A spherical shaped flame front may approximate the case when it is not disturbed by a gas flow like a swirl and a squish in a chamber. Engines which were used to develop a turbulent combustion model as a phenomenological model by the groups of Keck [3], [10], [11] and Tabaczynski [7]-[9] are mostly OHV engines. In general, the OHV engines have combustion chambers which generate a considerable squish but not a swirl so that the assumption of spherical flame front may be valid if the shape of chamber is almost symmetrical. However, for SV engines (side-valve type engines) which have a more complicated shape combustion chamber than that of the OHV engines, it is necessary to consider the effects of squish as well as swirl. Recently, even for the OHV engines, a swirling gas flow is made intentionally for rapid combustion of the charge.

In these high-swirl chambers, it is not expected that the assumptions of the homogeneity of turbulence and the spherical flame propagation retain their validity. Then, the model mentioned above has a fundamental deficit from this point of view. Returning to the basic hydrodynamic conservation equations for the unsteady compressible flow with some appropriate turbulent models may be the best way to treat this problem [13] because of the incredible development of high-speed supercomputers. However, even with this fundamental approach, there remains an uncertainty in constructing the turbulence model. In addition, computing costs are not so small at present. Then, it is still valuable to develop the phenomenological turbulent combustion model which is available for the chambers with considerable gas flows. For this reason, in the present paper, the phenomenological model based on the Blizard-Keck model is developed to take into account of the gas flows in the chamber which are described in the previous section.

Model Equations

The Blizard-Keck model as well as the models that was developed after basically consist of two ordinary differential equations with respect to time. The first equation describes the mass conservation across the turbulent flame zone in defining the turbulent flame speed $u_\tau$ as:

$$\frac{dx}{dt} = \rho_0 A_f u_\tau,$$

where the explanations of the symbols used are listed in the NOMENCLATURE. Equation (1) represents the first stage of turbulent flame propagation when the unburned charge is entrained in the combustion zone. The second equation states that in the combustion process of the turbulent combustion zone small eddies entrained in burn in the laminar mode and is written as:

$$\frac{dx}{dt} = \frac{x_b - x_b}{\tau},$$

where $x_b$ denotes a mass fraction of burned gas and the characteristic time defined by using a turbulence length scale $l_\tau$ and a laminar flame speed $u_\tau$ as:

$$\tau = l_\tau / u_\tau.$$

Figure 8 illustrates the situation described by Eqs. (1) and (2).

![Fig. 8 Model of turbulent combustion zone](image-url)
In order to solve these equations and to find the variation of the mass fraction burned with time \( t \), five parameters, i.e., unburned gas density \( \rho_u \), area of the flame front \( A_F \), laminar flame speed \( u_1 \), turbulent flame speed \( u_t \), and turbulent length scale \( l_e \) should be given as functions of time.

The first parameter \( \rho_u \) can be calculated by using the thermodynamic model that has been described in the previous section where the mass fraction burned was calculated from the experimentally obtained cylinder pressure. In this section, the same equation is used in the following form:

\[
\frac{dP}{dt} = \frac{\Omega_1}{\Omega_2} ,
\]

where the functions \( \Omega_1 \) and \( \Omega_2 \) are defined as:

\[
\Omega_1 = m_u \frac{d\rho}{dt} \left[ C_{v, u} \left( \frac{1}{\rho} \right) \frac{1}{u_1} (T_0 - T_F) + (\tau_u - \tau_0) (T_0 - T) \right]
- \tau_0 \rho \frac{dV}{dt} (T_0 - T_0) \frac{dQ}{dt},
\]

\[
\Omega_2 = V \frac{\rho_0 - \rho_u}{\rho_0} (1 - x_0) V \left( \frac{1}{\rho_0} \right) \frac{1}{u_1},
\]

Then, \( \rho_u \) is obtained from \( p \) by using the isentropic relation:

\[
\rho_u = \rho_0 \left( \frac{p}{P_0} \right)^{\frac{1}{\gamma}},
\]

where the subscript \( i \) denotes the condition at the bottom dead center in the suction.

The second parameter \( A_F \) can be evaluated by assuming the spherical flame propagation. Some geometric calculations derive the relationship between \( A_F \) and \( V_t \). The latter is a volume enclosed in the flame front, and is calculated as

\[
V_t = V - (m - m_t) \rho_u,
\]

where \( m_t \) denotes the mass entrained in the flame front.

The third parameter \( u_1 \), the laminar flame velocity, is estimated by the empirical formula,

\[
u_1 = u_{0l} \left( \frac{\rho_u}{\rho_0} \right)^{\alpha_1} \exp \left[ \frac{E}{RT_0} - \frac{1}{h_{b0}} - \frac{1}{h_0} \right],
\]

with the empirical constants being evaluated to fit experimental data.

The last two parameters \( u_t \) and \( l_e \) are most important in this modeling. Here, the original Blitzard-Keck model is just modified to take into account of their variations in course of the flame propagation. The proposed equations are as follows:

\[
\nu_t = \nu_t \left( \frac{\rho_u}{\rho_0} \right)^{\gamma},
\]

\[
l_e = C_l \left[ \nu_t \left( \frac{\rho_u}{\rho_0} \right)^{\gamma} \right]^{\delta},
\]

where

\[
u_t = C_t u_1 \left( \frac{\rho_u}{\rho_0} \right)^{\gamma},
\]

for conditions that do not violate drastically the spherical flame propagations. The characteristic velocity \( u_1 \) is defined as

\[
u_1 = \tau_0 u_0 A_p / A_t,
\]

which represents an inlet velocity into a main chamber of SV engine. The last part in the right-hand side of Eq. (12) expresses the dependency of the turbulent burning velocity on the local flow velocity if it is proportional to \( r_e^{-4} \). This term should be considered when the intensive swirl appears in the main chamber.

Method of Calculation

Equations (1) through (14) were solved with initial conditions at the time of spark, that is,

\[
x_0 = x_t = 0,
\]

\[
p = p_0,
\]

but it is necessary to give a small value for the radius of flame front \( r_F \) at the ignition for initiating the flame propagation.

The values of parameters that appeared in the equations are given assuming that the fuel is octane \( (C_8H_{18}) \) and the burned gas reaches immediately chemical equilibrium condition where, for simplicity, six species \( \{CO_2, CO, H_2, H_2O, C_2O, N_2\} \) exist.

With taking into account for the residual gases that exist in each cycle, the thermochemical parameters like \( h_{f0}, h_{f_b}, \gamma_u, \gamma_b, C_{p_u}, \) and \( C_{p_b} \) are calculated for each operating condition. For the laminar flame velocity \( u_1 \), the activation energy \( E \) and the exponent \( \alpha \) are evaluated to fit the experimental data [14] and the adiabatic flame temperature \( T_b \) is calculated assuming chemical equilibrium. The values which are used in this calculation are:

\[
E = 491 \text{ MJ/kmol}, \alpha = 0.4, u_{10} = 0.316 \text{ m/s},
\]

\[
p_{u_0} = 1.05 \text{ kg/m}^3, T_{b0} = 2440 \text{ K}.
\]

The ordinary differential equations (1), (2), and (3) are transformed into the appropriate non-dimensional form and numerically integrated by the Runge-Kutta-Gill method.

RESULTS AND DISCUSSIONS

The empirical constants \( C_1, C_2, C_3 \) and \( C_4 \) which appeared in the correlation formulae (10) through (13) for the turbulent length scale and the turbulent flame speed are evaluated to match...
the calculated mass fraction burned profile with the experimental results for the standard operating condition, i.e., \( N = 1300 \text{ rpm}, \phi = 1.0, \) IT = BTDC 25 deg, \( \tau_v = 0.67 \) for the chamber S0 and S4. The values adopted are \( C_1 = 0.32, C_2 = 0.24, C_3 = 2.2 \) and \( C_4 = 0 \) for the chamber S0 or \( C_4 = 1.2 \) for the chamber S4.

Figure 9-a shows the calculated mass fraction burned curves with the experimental plots. Agreement between calculated and experimental data is quite good while the model is quite simple. The effects of spark timing can be seen from this figure. Figures 9-b and 9-c show the time history of length scale and flame speed respectively. As the spark advance angle decreases, the non-dimensional turbulent length scale increases because the period when the eddies are compressed decreases. The turbulent flame speed increases rapidly with crank angle for large spark advance but is almost constant for small advance which might be caused by the density dependence of the flame speed. In Table 3, the thermochemical parameters obtained in this calculation as well as the length scale and the flame speed at the time of ignition are listed.

Figure 10 represents the effects of the engine speed on the mass fraction burned curve. Only a little effect appeared because there seems to be a similar relation between the flame speed and the engine speed.

In Fig. 11, the effects of the volumetric efficiency are shown and it is found that the larger \( \eta_v \) results in the shorter burning time as is expected.

Figure 12 expresses that in the lean mixture the flame speed is smaller than that in a stoichiometric mixture.

Figure 13 shows the effects of the gas flow in the chamber. It should be noted that for this figures the calculated curves for the chamber S4 the constant \( C_4 \) is set equal to zero. For the moderate gas flow appeared in the chamber S4, the correlation formula for the turbulent flame speed

Table 3  Thermochemical parameters, length scale and flame speed at ignition

<table>
<thead>
<tr>
<th>IT BTDC</th>
<th>( \gamma_v )</th>
<th>( \gamma_0 )</th>
<th>( C_{pb} )</th>
<th>( C_{ph} )</th>
<th>( \mathbf{h}_{\mathbf{ph}} )</th>
<th>( \mathbf{h}_{\mathbf{ph}} )</th>
<th>( \mathbf{h}_{\mathbf{ph}} )</th>
<th>( \mathbf{m} )</th>
<th>( \mathbf{w} )</th>
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<td>1.22</td>
<td>1166</td>
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<td>3.17</td>
<td>0.69</td>
<td>4.35</td>
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<tr>
<td>25°</td>
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<td>1.23</td>
<td>1161</td>
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<td>3.30</td>
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<td>4.38</td>
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</tbody>
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Fig. 9-a  Mass fraction burned gas profile

Fig. 9-b  Length scale profile (Calculated)

Fig. 9-c  Turbulent flame speed profile (Calculated)
Fig. 12 Mass fraction burned profile

Fig. 13 Mass fraction burned profile

need not take into account of the effects of spatial inhomogeneity, while for the high-velocity swirl generated in the chamber S4, it is necessary to consider the spatial inhomogeneity created by the swirl. This fact can be seen in Fig. 13 as a great discrepancy between the predicted curve and the experimental points. In Figs. 14 a, b, and c the results with $C_2=1.2$ are shown for the chamber S4. In this chamber, the flame propagates very slowly at the initial stage and accelerates rapidly as it propagates through the accelerating swirl velocity field. This successful results may be due to the fact that even in this high-swirl chamber S4 the spherical flame propagation is realized because the center of the swirl is identical to the ignition point.

CONCLUSIONS

The mass fraction burned in the combustion chamber was predicted as a function of time for various operating conditions and chamber geometries in the side-valve type spark ignition engine. In order to do this, the turbulent burning model was developed to include the effects of gas flows generated in the chamber and of density variation of the unburned gas. The calculated results were compared with the experimental data obtained by using the single-cylinder side-valve engine. The model proposed well simulated the turbulent flame propagation process and the following conclusions are derived.

(1) Turbulent flame speed $u_f$ can be derived as a sum of a laminar flame speed and a term which depends on the mean gas velocity between the divided chambers and also depends on the instantaneous unburned gas density.

Fig. 14-a Mass fraction burned gas profile

Fig. 14-b Length scale profile (Calculated)

Fig. 14-c Turbulent flame speed profile (Calculated)

(2) Turbulent length scale $l_u$ depends on the valve lift and the instantaneous unburned gas density.

(3) For the high-swirl chamber with ignition at the swirl center, the turbulent flame speed depends on the flame position because of the spatial inhomogeneity of the velocity field.

NOMENCLATURE

- $A_f$: flame front area, m²
- $A_t$: area connected two chamber, m²
- $A_p$: piston surface area, m²
- $A_{in}$: area of inlet valve, m²
- $C_p$: specific heat at constant pressure, J/kg·K
- $C_v$: specific heat at constant volume, J/kg·K
- $E$: activation energy, J/mol
- $h_f$: specific enthalpy of formation gas mixture, J/kg
\( \text{IT} \) = spark timing, BTDC deg
\( l_s \) = stroke, m
\( l_vl \) = inlet valve lift, m
\( l_e \) = turbulent length scale, m
\( m \) = total mass of gas in combustion chamber, kg
\( m_b \) = mass of burned gas, kg
\( m_f \) = total mass of flame front, kg
\( N_e \) = engine speed, rpm
\( p \) = pressure, Pa
\( -Q \) = heat loss to the chamber wall, J
\( r_f \) = radius of flame front, m
\( r_p \) = radius of piston, m
\( T \) = temperature, K
\( T_i \) = inlet gas temperature, K
\( u_e \) = turbulent flame speed, m/s
\( u_i \) = mean inlet gas velocity
\( u_l \) = laminar flame speed, m/s
\( u_p \) = mean piston speed, m/s
\( u_T \) = characteristic velocity related to turbulence, m/s
\( V \) = volume of combustion chamber, m³
\( V_c \) = clearance volume, m³
\( V_f \) = volume enclosed by flame front, m³
\( x_b \) = mass fraction of burned gas
\( x_f \) = mass fraction of burned gas
\( \alpha \) = exponent in laminar flame speed correlation
\( \gamma \) = ratio of specific heat
\( \xi \) = compression ratio
\( \eta \) = volumetric efficiency
\( \phi \) = crank angle, deg
\( \rho \) = gas density, kg/m³
\( \tau \) = characteristic reaction time, s
\( \Phi \) = equivalence ratio

Subscripts
\( 0 \) = reference conditions
\( b \) = burned gas
\( i \) = conditions at bottom dead center
\( s \) = conditions at spark time
\( u \) = unburned gas

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