A control-oriented two-zone charge mixing model is developed to simplify, but to describe mixing of fresh charge and residual gas during the intake stroke. Engine valve timing has a strong influence on the realization of stable homogeneous charge compression ignition (HCCI), since it affects turbulent flow that promotes mixing of fresh charge and residual gas. Controlled auto-ignition of a HCCI engine is achieved by good mixing of fresh charge and residual gas. Therefore, it is useful to develop a mixing model that can be executed in real-time to help extend the operational range of HCCI. For model derivation, the cylinder volume is artificially divided into two zones with a fictitious divider between them. First, the mixed zone consists of fresh charge induced by opening intake valves and some residual gas transferred from the unmixed zone. They are assumed to have been mixed homogeneously so that cold fresh charge gains thermal energy from hot residual gas. Second, the unmixed zone contains the rest of hot residual gas. Mass transfer between them which is forced by a fluid jet is directed from the unmixed zone to the mixed one. Based on the definitions of two zones and interaction between them, a two-zone charge mixing model is derived. To investigate phasing effects of valve timing on charge mixing, comparative simulation was carried out with different valve timings. For experimental validation and calibration of the proposed model, optical engine tests were performed with an infrared (IR) camera, together with GT-power simulation. From good agreement between the model temperature and the estimated temperature from IR images, the model turns out to be useful to describe mixing of fresh charge and residual gas. [DOI: 10.1115/1.4026660]

1 Introduction

During the past couple of decades, advanced combustion technologies have been extensively investigated to reduce harmful pollution and improve fuel efficiency of internal combustion engines (ICEs) [1–3]. In this perspective, HCCI engines have drawn keen attention as one of the most competitive alternatives. HCCI engines have been shown to produce much less nitrogen oxide and soot emissions while improving fuel efficiency [4]. Unlike spark ignition (SI) or compression ignition (CI) engines, there is no explicit initiator of combustion for HCCI engines. Instead, HCCI is attained by controlled auto-ignition occurring at multiple places at a time, when a homogeneous mixture is compressed so that its density and temperature reach appropriate conditions.

However, auto-ignition is difficult to control due to the absence of a direct initiator. And the operational range of HCCI engines is restricted. It is constrained at low load by lean combustion limit and at high load by in-cylinder peak pressure restriction. Variable valve actuation (VVA) turns out to be effective for resolving such challenges. VVA provides fine control over the temperature and pressure of a mixture by manipulation of effective compression ratio [5] and amount of hot residual gas retained. Quantitative regulation of hot residual gas facilitates temperature control of the mixture for auto-ignition. In particular, VVA with a negative valve overlap (NVO) strategy offers great advantages for HCCI combustion control [6,7]. NVO strategy enables effective control of the amount of residual gas to be trapped. However, not all of residual gas which is trapped will affect active reactant. Therefore, appropriate charge mixing model between the fresh charge (fuel + air) and the residual gas is essential for HCCI combustion control and extension of its available range.

Most modeling works for HCCI engines are based on assumption of a lumped system [8–10]. The mean value HCCI engine model is developed in continuous time domain [8] for control purpose. Cycle-to-cycle based model [9,10] were also proposed. Reference [10] employed mass fraction of gas species included in fresh charge and residual gas to predict peak pressure and start of combustion. These approaches commonly assume that thermodynamic properties of in-cylinder gas are uniform without regard of charge mixing.

Actual fluidic behavior in a cylinder during the intake stroke is extremely complicated due to turbulence. The turbulence is generated by intake flow which comes through intake valves. Thus, thermofluidic characteristics in a cylinder is not uniform at all throughout the cylinder volume. But, turbulence enables mixing of fresh charge and residual gas in a cylinder. It means that mixing of fresh charge and residual gas is strongly influenced by valve timings. Recently, the multizone model for HCCI is investigated in the framework of computational fluid dynamics [11] to simulate charge mixing in a cylinder. Unfortunately, such a model is not
suitable for use of real-time control because it requires extremely heavy computational load. Therefore, it is desired to develop a model with logical and plausible assumptions for simplification, but it still should retain significant physical features for model-based control design. Such a model allows real-time model-based determination of valve timing in order to facilitate stable HCCI combustion in a broader working range.

As a compromise between the single- and multizone models, two-zone HCCI models are proposed [12,13]. In Ref. [12], the mass transfer between the mixed- and unmixed zone is predetermined with respect to a positive valve overlap strategy. Reference [13] assumed the cylindrical shapes of two zones which actually rarely happen. Reference [14] presents a control-oriented two-zone model to determine the zone size in real-time. This paper extends the results from Ref. [14] by evaluating its impact for different valve timing and strategy as well as developing a complete calibration method for the model. In the proposed two-zone charge mixing model, the cylinder volume is segmented into the unmixed- and mixed zones. The unmixed zone includes residual gas only. This zone serves as a temporary repository of hot residual gas which plays a role of the heat source. Whereas, in the mixed zone, it is assumed that fresh charge and some portion of residual gas, expelled from the unmixed zone, are mixed homogeneously. Mass transfer from the unmixed zone to the mixed zone is driven by a fluidic impingement on the unmixed zone. Mass transfer between two zones represents mixing of fresh charge and residual gas in the cylinder.

To validate and calibrate the developed model, optical engine tests were performed with an IR camera. And GT-power simulation is carried out to further assist the model validation. Since an optical engine enables direct access inside a combustion chamber, it is broadly used in ICEs, particularly for diagnostic applications such as detection of fuel spray injection and combustion flame propagation [15,16]. In this research, mean temperature of two zones is estimated by radiation measurement with an IR camera. Image postprocess [17] is done to filter out inevitable noise and to extract features of interest. After postprocess, the IR image is segmented into two zones using the k-means cluster technique. Then, the mean temperature of each zone is estimated and compared with the model temperature.

The paper is organized as follows. In Sec. 2, definitions of two zones are established, and a charge mixing model is derived by associating two zones with mass transfer between them. Comparative simulation studies were carried out to investigate effects of valve timings on charge mixing in Sec. 3. Then, in Sec. 4 the optical engine used and theoretical background are presented briefly. And the methodologies for model calibration and validation with the measured IR images are demonstrated. The model temperature and the estimated temperature from IR images are compared. Finally conclusion is provided in Sec. 5.

2 Charge Mixing Model

Two zones for the charge mixing model are conceptually described in Fig. 1. Region 2 in the figure indicates a portion of residual gas which is burned in the previous cycle and retained by the early closing of exhaust valves. This burned gas is not blended with fresh charge at all, but plays a significant role as a temporary storage of heat source. Region 1 shows a mixture of fresh charge which enters a cylinder and some residual gas, which is transferred from the unmixed zone. There is a fictitious divider that separates the two zones. Heat transfer between two zones through a fictitious divider does not occur. All possible heat transfer between fresh charge and residual gas occurs in the mixed zone through effective collision. So fresh charge and residual gas are assumed homogeneously to be mixed in the mixed zone. Mass transfer between two zones is forced by a fluid jet (the arrow directing to Region 2) acting on the fictitious divider at high speed. So, mass is transferred from the unmixed zone to the mixed zone. The transferred residual gas in the mixed zone provides the fresh charge with crucial thermal energy, and enables the mixture to prepare for auto-ignition. In this research, charge mixing during the intake stroke is concerned. There exists charge mixing during the compression stroke due to inertial fluidic motion. However, comparably weak turbulent flow exists in a cylinder during the compression stroke [18,19]. Therefore, it can be inferred that charge mixing during the intake stroke is more significant.

Piston motion dynamics is first described with engine speed. Then valve flow is modeled as a function of the valve motion. Impact pressure acting on a fictitious divider is derived based on the kinetic energy of the inlet flow. Based on this impact force, mass transfer from the unmixed zone to the mixed zone is modeled. Finally, the thermodynamics of both zones are presented.

2.1 Piston Motion Dynamics

From a geometric configuration in Fig. 2, the position of the piston pin and its rate are determined as Eqs. (1) and (2) [20].
\[ s = (l + a) - \left( \sqrt{2 - a^2 \sin^2 \theta + a \cos \theta} \right) \]  
\[ \dot{s} = \dot{\theta} \left( a \sin \theta + \frac{a^2 \sin \theta \cos \theta}{\sqrt{2 - a^2 \sin^2 \theta}} \right) \]  

The cylinder volume and the volume rate change with the location and the speed of the piston head as Eqs. (3) and (4). The displacement volume \( V_d \) is the volume swept by the piston while it moves between top dead center (TDC) and bottom dead center (BDC) [20].

\[ V_{cyl} = V_d \left( \frac{1}{r_e} - 1 + \frac{s}{2a} \right) \]  
\[ \dot{V}_{cyl} = \frac{V_d}{2a} \dot{s} \]

2.2 Valve Flow Dynamics. All working fluids concerned are assumed to be ideal. Flow rate through the intake/exhaust valves are shown in Eq. (5) with the orifice equation. From \( \dot{m}_v = \rho_{up} A_v v_r \), flow velocity is shown in Eq. (6).

\[ \dot{m}_v = \frac{C_d A_v}{\sqrt{RT_{up}}} \rho_{up} \Phi(r_p) \]  
\[ v_r = \frac{C_d}{\sqrt{RT_{up}}} \Phi(r_p) \]

\( A_v \) is determined by the valve lift depending on \( \theta \). It is noted that \( \dot{m}_v \) would be negative if the gas in the cylinder is expelled out of the cylinder. Thus, flow direction through the orifice area should be determined in accordance with pressure gradient through the valve. \( \Phi(r_p) \) is determined with pressure ratio of up- and down-stream as below

\[ \Phi(r_p) = \begin{cases} \frac{1}{r_p} \sqrt{\frac{2}{\gamma - 1} \left( 1 - r_p^\gamma \right)} & \text{if } r_p > \left( \frac{\gamma}{\gamma + 1} \right)^\frac{1}{\gamma} \\ \frac{1}{\gamma^\frac{1}{\gamma}} & \text{if } r_p \leq \left( \frac{\gamma}{\gamma + 1} \right)^\frac{1}{\gamma} \end{cases} \]

It is mentioned that the turbulent flow, which is generated by intake flow, is the fundamental driving force for mixing and consequently such turbulence provokes pressure imbalance during the intake stroke [18,19]. From the energy conservation law of a fresh charge in motion, when the fluid jet hits the fictitious divider, its impact pressure is calculated as

\[ p_{jet} = \frac{p_{in} v_r^2}{2RT_{in}} \]  

Since \( p_{cyl} \) is close to \( p_{in} \) during the intake stroke, the ratio of \( p_{jet} \) and \( p_{cyl} \) is approximated as \( C_d^2 \Phi(r_p) \gamma^2 / 2 \). Therefore, \( p_{jet} \) is much smaller than \( p_{cyl} \) during the intake stroke. Thus, the pressure imbalance is nearly negligible compared to the mean cylinder pressure, although it is still critical for mass transfer between two zones. Therefore, the pressure is assumed to be uniform without zonal difference.

2.3 Cylinder Volume System. Equations (9) and (10) show mass and temperature rate of the cylinder volume system.

\[ m_{cyl} = \dot{m}_v \]  
\[ \dot{T}_{cyl} = \frac{\gamma - 1}{m_{cyl} \cdot R} \left( -p_{cyl} \dot{V}_{cyl} + \dot{Q}_{cyl} \right) \]

\[ \begin{align} \dot{m}_v &= \frac{m_{cyl}}{m_{cyl}} (\gamma T_{im} - T_{cyl}) \quad \text{if inflow via an IV} \\ \dot{Q}_{cyl} &= -h_{cyl}A_{cyl}(T_{cyl} - T_{wall}) \quad \text{if outflow} \end{align} \]  

Equations (11) and (12) describe ideal gas law and heat transfer rate from/to the cylinder wall for the cylinder volume system.

\[ p_{cyl} = \frac{1}{V_{cyl}} \left( \dot{m}_{cyl} \cdot RT_{cyl} + \dot{m}_{cyl} \cdot \dot{Q}_{cyl} \cdot \dot{V}_{cyl} \right) \]  
\[ \dot{Q}_{cyl} = -h_{cyl}A_{cyl}(T_{cyl} - T_{wall}) \]

where \( T_{wall} \) is assumed to be constant, \( h_{cyl} \) is determined by Woschni’s correlation [20,21] with the scalable factor \( x \) depending on engine geometry as

\[ h_{cyl} = x L^{0.2} p_{cyl}^{0.8} T_{cyl}^{0.55} \cos^{0.8} \]

2.4 Mixed Zone: Fresh Charge + Residual Gas. The mixed zone has two inlets or one outlet depending on the flow direction and associated valves. During outflow from the cylinder to the manifold (intake or exhaust), escape rate from each zone is determined according to its mass fraction. Mass and energy equations of the mixed zone are shown in below equations

\[ \dot{m}_1 = \begin{cases} \dot{m}_cyl + \dot{m}_1 & \text{if inflow} \\ \dot{m}_cyl \times \frac{m_{cyl}}{m_{cyl}} & \text{if outflow} \end{cases} \]

\[ \frac{d}{dt}(c, m_{cyl} T_{cyl}) = -p_{cyl} \dot{V}_{cyl} + \dot{Q}_{cyl} \]

\[ \dot{Q}_{cyl} = -h_{cyl}A_{cyl}(T_{cyl} - T_{wall}) \]

Based on the zonal uniformity in pressure, ideal gas law, and heat transfer rate for the mixed zone are shown in below equations

\[ -p_{cyl} \dot{V}_{cyl} + m_{cyl} \cdot RT_{cyl} = p_{cyl} V_{cyl} - \dot{m}_1 \cdot RT_{cyl} \]  
\[ \dot{Q}_{cyl} = -h_{cyl}A_{cyl}(T_{cyl} - T_{wall}) \]

2.5 Unmixed Zone: Residual Gas Only. For the unmixed zone, there is one outlet when fresh charge inflows. The unmixed zone consistently shrinks until it fades away completely. Similarly, dynamics for the unmixed zone is described in below equations

\[ \dot{m}_2 = \begin{cases} -\dot{m}_1 & \text{if inflow} \\ \dot{m}_cyl \times \frac{m_{cyl}}{m_{cyl}} & \text{if outflow} \end{cases} \]

\[ \frac{d}{dt}(c, m_{cyl} T_{cyl}) = -p_{cyl} \dot{V}_{cyl} + \dot{Q}_{cyl} + c_p \dot{m}_2 T_{cyl} \]

\[ -p_{cyl} \dot{V}_{cyl} + m_{cyl} \cdot RT_{cyl} = p_{cyl} V_{cyl} - \dot{m}_2 RT_{cyl} \]

\[ \dot{Q}_{cyl} = -h_{cyl}A_{cyl}(T_{cyl} - T_{wall}) \]

Because total heat transfer sum of two zones is identical to the heat transfer of the cylinder volume, following assignment rule is made:
It is noted that the multiplication of the heat transfer coefficient and the contact area can be determined only, not individual values. It is because the contact area cannot be determined due to fairly complex shapes of zones. Using Eqs. (14)–(23), the two-zone charge mixing model is derived in a matrix form of Eqs. (24)–(27). It can be seen that all variables of the cylinder volume system are treated as the exogenous signals of the two-zone charge mixing model.

\[
\Sigma Y = \Pi \\
X = \begin{bmatrix} \dot{T}_1, \dot{T}_2, \dot{V}_1, \dot{V}_2 \end{bmatrix}^T \\
\Sigma = \begin{bmatrix} c_v m_1 & 0 & p_{cyl} & 0 \\
0 & c_v m_2 & 0 & p_{cyl} \\
m_1 R & 0 & -p_{cyl} & 0 \\
0 & m_2 R & 0 & -p_{cyl} \end{bmatrix} \\
\Pi = \begin{bmatrix} (i) \text{ inflow through IV}, \\
-\dot{m}_1 c_v T_1 - m_2 c_v T_2 + \dot{m}_1 c_p T_{in} - \dot{Q}_1 \\
-\dot{m}_1 R T_1 + p_{cyl} V_1 \\
-\dot{m}_2 R T_2 + p_{cyl} V_2 \\
(ii) \text{ inflow through EV}, \\
-\dot{m}_1 c_v T_1 - m_2 c_v T_2 + \dot{m}_1 c_p T_{in} - \dot{Q}_1 \\
-\dot{m}_1 R T_1 + p_{cyl} V_1 \\
-\dot{m}_2 R T_2 + p_{cyl} V_2 \\
(iii) \text{ outflow}, \\
\dot{m}_1 R T_1 - m_2 c_v T_2 - \dot{Q}_1 \\
\dot{m}_2 R T_2 - \dot{Q}_2 \\
-\dot{m}_1 R T_1 + p_{cyl} V_1 \\
-\dot{m}_2 R T_2 + p_{cyl} V_2 \end{bmatrix}
\]

### 2.6 Mass Transfer Between Two Zones

The two-zone charge mixing model is still indeterminate because of unknown \( \dot{m}_c \). It is reminded that residual gas in the unmixed zone is transferred to the mixed zone by impact pressure exerting on the fictitious divider while keeping pressure of both zones identical (or nearly similar). It means that compressive work by a fluid jet is consumed by enthalpy outflow, in other words, \( \dot{m}_c \dot{V}_l = c_p m T_2 \). \( \dot{V}_l \) is approximated with \( \dot{V}_l = \dot{V}_c / \rho_0 \). Finally, mass transfer rate between two zones is determined with a constant mixing parameter (\( \eta \)) as

\[
\dot{m}_c = \frac{\gamma - 1}{\gamma} \frac{\dot{m}_c V_l^2}{2RT_2}
\]

From Eq. (28), it can be inferred that mass transfer rate between two zones is proportional to the kinetic energy rate of intake flow. Since turbulent flow boosts charge mixing more than laminar flow [20], \( \eta \) is used to represent the engine geometry-depending turbulent effect. With Eq. (28), the two-zone charge mixing model, Eq. (24) becomes determinate. It is noted that even though the mixing model is derived with the negative valve overlap strategy, it can be modified to accommodate other valve strategies.

### 3 Model Calibration and Comparative Analysis

#### 3.1 Global Parameters Calibration

In this section, global parameters are calibrated to identify states of the cylinder volume system using optical engine tests and GT-power simulation that provides a high fidelity engine model. Since mean temperature of the cylinder volume system for the actual engine is not available, the value from GT-power simulation is used. GT-power simulation is carried out to track the measurement values of optical engine tests such as IMEP, AFR, \( P_cyl \), and fuel mass with identical conditions. It is noted that the proposed two-zone charge mixing model does not require any specific combustion type. Thus, optical engine tests and GT-power simulation are done with SI combustion for consistent use of residual gas under stable combustion. The parameters include \( C_d \) for intake and exhaust valve flow in Eqs. (5) and (6), and \( \eta \) for the heat transfer coefficient in Eq. (13).

The engine runs at steady speed of 1500 RPM. The compression ratio is 9.8 and the displacement volume is 0.499 l. AFR is kept at the stoichiometric ratio, and indoline is used as the fuel for optical engine tests. Experimental conditions are shown in Table 1. EVO, EVC, intake valve opening (IVO), and intake valve closing (IVC) are −168, −20, 32, and 180 crank angle degree (CAD) with respect to the TDC position during the intake stroke. Zero CAD indicates TDC position during the intake stroke. The calibrated values of \( C_d \) and \( \eta \) are 0.85 and 2.00, respectively. Figure 3 shows the valve profile used and mean value comparisons of pressure, temperature, and mass for the in-cylinder gas. With calibrated parameters, acceptable agreements are achieved.

With the calibrated global parameters, additional comparative simulations are performed with different \( \eta \). With manipulation of \( \eta \) in Eq. (28), its influence on temperature of each zone is investigated. The mixing degree is defined as the mass ratio of transferred residual gas to the mixed zone at IVC and total retained residual gas at IVO. With the given valve profile, the mixed zone is generated after 40 CAD because of back-flow in the beginning of IVO. In Fig. 4, it is found that the mixing degree does not affect temperature of the unmixed zone. However, temperature of the mixed zone is obviously influenced by the mixing degree. This dependence of the mixed zone is utilized for calibration of \( \eta \) with optical engine test results. Details will be presented later.

#### 3.2 Phasing Effect: Exhaust Valve Timing

In this section, phasing effect of exhaust valve timing on two zones is investigated. Here, the calibrated global parameters (\( C_d, \eta \)) and the tuned mixing parameter (\( \eta \)) are used. Exhaust valve timings with phasing of ±10 CAD are shown in Fig. 5(a). For three cases, same initial conditions of temperature, mass, and pressure at –200 CAD are used. From Figs. 5(b) and 5(c), it is observed that 10 CAD retard of exhaust valve timing enhances mixing of fresh charge and residual gas. It is because the decreased pressure in a cylinder generates strong intake flow of high kinetic energy. However, since mass and temperature of retained residual gas at IVO are reduced by retard of exhaust valve timing, retard angle is limited.
3.3 Phasing Effect: Intake Valve Timing. Similarly, phasing effect of intake valve timing is investigated here. Three cases of intake valve timings with 6\(^\pm\)10 CAD phasing as shown in Fig. 6(a) are compared under same initial conditions. As can be seen in Figs. 6(b) and 6(c), 10 CAD retard of intake valve timing promotes mixing of fresh charge and residual gas. It is due to strong intake flow of high kinetic energy as phasing of intake valve timing. As a result, fresh charge gets more thermal energy from residual gas. However, since retarding intake valve timing makes unmixed zone temperature reduced, particularly when substantial residual gas is transferred to the mixed zone (60–100 CAD), the effect of retarding intake valve timing is limited. Also, the intake valve should be closed before BDC or slightly after BDC to avoid reduction of intake charge.

3.4 Intake Valve Timing: Synchronous Versus Asynchronous. In this section, influence of synchronous and asynchronous intake valve timings on two zones is investigated. Study on effect of independent intake valve control [6] motivates this comparative analysis. Figure 7(a) shows the valve timing used. Valve timing for one of two intake valves is shared, but the other is different. With asynchronous intake valve timing, better mixing is achieved (the less mass of the unmixed zone at IVC), as a consequence, fresh charge in the mixed zone gets more thermal energy from the transferred residual gas. In comparison with 10 CAD retard of intake valve timing, asynchronous intake valve timing
Timing enables making up for thermal loss due to reduced residual gas temperature by acceleration of charge mixing. Therefore, asynchronous valve strategy is a practical alternative of retarding intake valve timing.

4 Optical Engine Tests

The proposed model presents thermodynamic interaction between the mixed and unmixed zones, in terms of mass transfer between them. Unfortunately, there are no available sensors other than pressure transducers that are capable of measuring in-cylinder properties because of high pressure and temperature in a cylinder. Due to such restriction, an optical engine has drawn attention for ICE research, since it allows direct optical access to the inside of a cylinder so that significant features like fuel spray injection and combustion flame propagation can be captured by images [15,16,22]. In this research, an IR camera was used to estimate the mean temperature of two zones during the intake stroke. Then, the mixing parameter ($\eta$) is calibrated with estimated temperature. Thereby, suitability of the proposed model is evaluated.

4.1 Optical Engine Setup. Figure 8 shows the optical engine which is used for experimental validation. Specifications of the optical engine used are provided in Ref. [14]. Radiant emittance from gas in the cylinder passes through the optical window which is mounted on the Bowditch style piston. Because the diameter of the optical window is smaller than the engine bore, only some portion of gas in the cylinder can be observed. After passing through the window, the radiation is reflected from the mirror, then reaches the detector. The schematic setting is shown in Fig. 9. At 30, 40, 50, 60, and 70 CAD, images are taken ten times in consecutive cycles to see cyclic variation.
4.2 Background. Detailed theoretical background of radiation transfer can be found in Ref. [23]. Every material whose surface temperature is over the zero degree Kelvin emits radiation from its surface. For a blackbody, spectral radiant emittance from its surface is determined with Plank’s law as a function of wave length \((\lambda)\) and surface temperature \((T)\). A real object of identical surface temperature emits less radiation than an ideal blackbody, and their ratio is defined as emissivity.

In contrast with a solid surface, a majority of gases features nonuniform spectral distribution of emissivity. And most gases turn out to have extremely small emissivity. Thus instead of gas, soot which is one of combustion products is generally employed because it has much higher emissivity than gas. There have been many efforts to estimate soot emissivity for optical engine research [15,16,22].

If all radiant emittance from the object in the cylinder reaches the detector without any loss, the measured image output (i.e., intensity) is determined as

\[
U = \int_{\lambda_1}^{\lambda_2} \kappa(\lambda) \varepsilon(\lambda) E_\lambda(\lambda, T) d\lambda
\]

(29)

where \(\kappa\) is the normalized productivity of the detector which converts from the incident radiation to the actual image output. \([\lambda_1, \lambda_2]\) are the measurable spectral range of the IR camera. It is noted that the image output mainly depends on the surface temperature and emissivity of an object.

4.3 Image Postprocess and Segmentation. To estimate mean temperature of two zones, an IR image should be divided into two according to intensity distribution. Since raw IR images inherently have anisotropic noise [24], the median filter is applied in order to filter out undesired noise sufficiently without unnecessary distortion of structural features (for example, edges [17]). Besides noise suppression, the filter gets rid of isolated pixels in the mixed zone. If tiny portion of the residual gas is mixed with fresh charge, it appears as a small isolated region having high intensity. By applying median filter, such an isolated region could be included in the mixed zone.

After filtering, pixels outside the optical window are excluded because pixels showing intensity of in-cylinder gas are concerned for zone identification from IR images. And the pixels where the spark plug locates are ruled out either. As presented, soot is used for temperature estimation. Soot is likely to be accumulated on the spark plug and cylinder head. Because temperature of the spark plug remains high, excessively high image output around the spark plug is found. Therefore, pixels except these area are used for image segmentation.

To partition the images into two zones, the \(k\)-means cluster method is employed [25]. This technique partitions pixels into \(k\) clusters so that a predetermined cost function is minimized. The cost function in Eq. (30) takes accounts of location and intensity of the pixels.

\[
J^{(k)}_j = \beta |U_i - \bar{U}^{(k)}_j|^2 + (1 - \beta) |P_i - \bar{P}^{(k)}_j|^2
\]

(30)

where, \(U_i \in \mathbb{R}, P_i \in \mathbb{R}^2, \bar{U}^{(k)}_j \in \mathbb{R}, \) and \(\bar{P}^{(k)}_j \in \mathbb{R}^2\). The following describes the iterative algorithm of the optimal \(k\)-means cluster method.

iterative algorithm

1. determine initial values of \(\bar{U}^{(0)}_j, \bar{P}^{(0)}_j\) for \(\forall j\)th cluster.
2. for the \(i\)th cell, find the \(j\)th cluster to minimize Eq. (30), then assign the \(i\)th pixel to the \(j\)th cluster.
3. update mean values of new clusters: \(\bar{U}^{(k+1)}_j, \bar{P}^{(k+1)}_j\)
4. go to (2) until cluster assignment does not change.

Although this algorithm solves the local solution, an initial value turns out to be nearly insensitive to optimal image segmentation through numerical examination, if the solutions (the mean image output and pixel location of clusters) are constrained to integers.

It is found that soot distribution is rather nonuniform even in the unmixed zone (see the raw image measured at 30 CAD before the mixed zone appears in Fig. 10). Unless such nonuniformity in the unmixed zone is taken account, temperature estimation based on the image segmentation would not be accurate. Two clusters are used for the unmixed zone. After inflow occurs, including one for the mixed zone, three clusters appear in the images. Figure 10 shows one of the raw (left) and segmented (right) images at 30,
40, 50, 60, and 70 CAD. In each plot, the above bigger and the below smaller arcs indicate the intake and exhaust valves, respectively. The red cluster is the area where much soot is concentrated in the unmixed zone. The yellow cluster is the area where sparse soot exists in the unmixed zone. And the light blue cluster represents the mixed zone. The dark blue indicates the excluded area for image segmentation. It is noted that all images are scaled using the discontinuous color map. The optimal mean intensity for each cluster (\( C_{22} U_j \)) will be used for mean temperature estimation of each zone.

4.4 Temperature Estimation and Model Calibration. Since in-cylinder gas has very low emissivity, in other words it is transparent to radiation, the detector observes radiation emitted from the background parts such as the cylinder head, valve ports, and the spark plug, in addition to gas radiation. Figure 11 depicts the path of radiant emittance. Based on Eq. (29) and radiant path shown in Fig. 11, Eq. (31) describes the mean intensity of the \( j \)th cluster.

\[
\begin{align*}
\tilde{U}_j &= U_{gas,j} + U_{bg,j} \\
U_{gas,j} &= \int_{\lambda_1}^{\lambda_2} \kappa(\lambda) \tau_{amb} \tau_{win} e_{gas,j}(\lambda) E_b(\lambda, T_{gas}) d\lambda \\
U_{bg,j} &= \int_{\lambda_1}^{\lambda_2} \kappa(\lambda) \tau_{amb} \tau_{win} e_{bg,j}(\lambda) E_b(\lambda, T_{bg}) d\lambda 
\end{align*}
\]

(31)

Transmissivity of the ambient gap is variant to crank angle, because optical thickness of the ambient gap varies while the piston head moves down. To compensate a dynamic optical thickness, pixels outside the optical window is used. The intensity of these pixels represents radiation emitted from the piston head. It is supposed to remain nearly constant, because piston temperature is also controlled by engine coolant. However, the measured mean intensity of these pixels is found to change while the piston head moves downward. To compensate such varying transmissivity, the normalizing factor, \( N(\theta) \), is defined to cancel out decreasing transmittance while the piston head moves downward. In other words, \( \tau_{amb}(\theta) = \tau_{amb} \times N(\theta) \) with the constant \( \tau_{amb} \). Therefore, normalized mean values of \( \tilde{U}_j = \tilde{U}_j/N(\theta) \), \( U_{gas,j} = U_{gas,j}/N(\theta) \), \( U_{bg,j} = U_{bg,j}/N(\theta) \), and \( U_{piston} = U_{piston}/N(\theta) \) are used for temperature estimation. \( \tilde{U}_j \) of the \( j \)th cluster is shown in Fig. 12.

To estimate \( T_{gas,j} \) from \( U_{gas,j} \), \( U_{gas,j} \) should be extracted from \( \tilde{U}_j \) in Eq. (31). \( \kappa, e_{gas,j}, \) and \( \tau_{gas,j} \) of gas are assumed to be constant.
for computational simplicity. Constant soot emissivity of 0.5 is selected from the literature [15]. This emissivity corresponds to the soot of combustion gas. The normalized mean intensity of the unmixed zone (\(U_{\text{gas,2}}\)) is nearly identical to \(U_1\), because \(E_0(T_1, T_{\text{gas,2}})\) is much higher than \(E_0(T_2, T_{\text{bg}})\). It is because temperature of the background parts is controlled at relatively lower temperature by the engine coolant. With assumption of zero reflectivity, \(\tau_{\text{gas,2}}\) is equal to 1 - \(\varepsilon_{\text{gas,2}}\), then the normalized mean intensity of the mixed zone (\(\tilde{U}_{\text{gas,1}}\)) becomes the half of \(\tilde{U}_1\), since the soot emissivity of the background parts is very high (\(\varepsilon_{\text{bg}} \approx 1\)) due to high soot accumulation on these parts.

Relationship between the normalized mean gas intensity (\(U_{\text{gas,1}}, U_{\text{gas,2}}\)) and mean temperature (\(T_{\text{gas,1}}, T_{\text{gas,2}}\)) of two zones can be identified by an empirical approximation method, instead of direct employment Eq. (31). In this paper, the following situ-Plank function [26] is applied.

\[
T_{\text{gas,j}} = \frac{A_1}{\ln \left( \frac{A_2}{U_{\text{gas,j}}} + A_3 \right)}
\]  

(32)

To identify parameters in Eq. (32), temperature (\(T_{\text{gas,2}}, T_{\text{piston}}\)) and intensity (\(U_{\text{gas,2}}, U_{\text{piston}}\)) are used. The mean temperature of the cylinder volume system from GT-power simulation before intake flow begins is used as \(T_{\text{gas,2}}\). From Sec. 3.1, it is reminded that \(\eta\) does not affect the unmixed zone temperature. The piston temperature is assumed to be equal (or almost same) with the cylinder wall temperature of 400 K. With these sample data, situ-Plank function is identified using nonlinear fitting curve method.

The mixed zone temperature is estimated by applying (\(U_{\text{gas,1}}\)) into Eq. (32). In Fig. 4, mixed zone temperature is simulated with the different mixing parameters (\(n\)). To have the best agreement between the model temperature and the estimated temperature of the mixed zone, \(\eta\) is calibrated. With best tuned \(\eta\) of 8, the model and estimated temperature of two zones are shown in Fig. 13. Good agreements between them are achieved.

5 Conclusions

A two-zone charge mixing model during the intake stroke is developed to describe interaction between the fresh charge and the residual gas for HCCI combustion control. The cylinder volume is divided into two zones according to mixing status. The mixed zone is composed of the fresh charge and the transferred residual gas. And the unmixed zone is filled with the rest of the residual gas captured by early close of the exhaust valve. The model shows low computational demand while retaining significant physical features. From comparative studies using simulation, manipulation of valve timing can improve the mixing effect between the fresh charge and the residual gas. Validity of the proposed model is evaluated using optical engine tests and GT-power simulation. For experimental validation, IR camera is employed to estimate zone temperature during the intake stroke. IR images are segmented into the two zones with the mean intensity using the k-means cluster method. Then, the relationship between the mean gas intensity and the gas temperature is identified with the situ-Plank function. Then, based on the situ-Plank function, mean temperature of two zones are estimated from IR images. From acceptable agreements between the model temperature and the estimated temperature, the proposed two-zone model turns out to be useful for the description of charge mixing.

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Temperature Estimation From IR Images

V_1, V_2 = volume of the mixed-, unmixed zone (m^3)
\eta = mixing parameter
\rho_{ch} = fresh charge density coming from the intake manifold (kg/m^3)

Optical Engine
B = engine bore (m)
d_{win} = diameter of the optical window (m)

Radiation Background
E_b = spectral radiant emittance of a blackbody (W/m^2/\mu m)
\kappa = normalized productivity (m^2/W)
U = IR image output (i.e., intensity)
\epsilon = emissivity
\lambda = wavelength (m)

Image Segmentation
P_j = position of jth pixel
P^{th}_j = mean position of jth pixel
I_j = intensity of jth pixel
I^{th}_j = mean intensity of jth cluster at kth iteration
\beta = positive weighting parameter between 0 and 1

Temperature Estimation From IR Images
A_1, A_2, A_3 = parameters of the situ-Plank function
N = normalizing factor
\tau_{amb} = transmissivity of the ambient gap
T_{bg} = mean temperature of the background (K)
T_{gas,j} = mean temperature of gas in jth cluster (K)
T_{gas,j,k} = mean temperature of gas in jth cluster
T_{gas,j,k,l} = mean temperature of gas in the mixed zone (K)
T_{piston} = mean temperature of the piston (K)
T_{opt} = transmissivity of the optical window
U_j = normalized mean intensity of jth cluster
U_{bg,j} = mean intensity of background in jth cluster
U_{gas,j} = mean intensity of gas in jth cluster
U_{gas,j,k} = normalized mean intensity of gas in jth cluster
U_{gas,j,k,l} = normalized mean intensity of gas in the mixed zone
U_{gas,j,k,l,m} = normalized mean intensity of gas in the unmixed zone
U_{piston} = mean intensity of the piston
U_{piston,j} = normalized mean intensity of the piston
U_{piston,j,k} = normalized mean intensity of the mixed zone
U_{piston,j,k,l} = normalized mean intensity of the unmixed zone
\epsilon_{bg} = emissivity of the background
\epsilon_{gas,j,k} = emissivity of gas in jth cluster

Acronyms
AFR = air to fuel ratio
CAD = crank angle degree
EVO, EVC = exhaust valve opening, -closing
IMEP = indicated mean effective pressure
IV, EV = intake-, exhaust valve
IVO, IVC = intake valve opening, -closing
TDC, BDC = top-, bottom dead center

References