ESTIMATION OF COMBUSTION EFFICIENCY FOR CEILING VENTED COMPARTMENT FIRES USING ZONE MODEL CONCEPT

by

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Combustion efficiency is difficult to measure or estimate for ceiling vented compartment fires because of the comprehensive interaction between vent flows and burning environment. Special phenomena suggesting very poor burnings were observed in such limited ceiling vented compartment fires, like ghosting flames and oscillating flames. An estimation method for the combustion efficiencies of pool fires, which was important to characterize the incompleteness of the combustion, was established in the scope of zone models, and was validated using a sealed compartment fire case. In this method, oxygen concentrations, fuel mass loss rates can be also predicted based on temperature predictions. Small scale ceiling vented compartment pool fires were studied using this method. Temperature distributions, fuel mass loss rates, and oxygen concentrations were compared in order to partially validate the mathematical model. The ventilation conditions of the poor fire were considered and quantified by introducing equivalence ratio. The present study may provide a practical way to understand the incomplete burning of such ceiling vented compartment fires.

Key words: *combustion efficiency, ceiling vent, zone model, enclosure fire*

Introduction

Ceiling vented compartments without any vertical opening are typical in ships, basements, and nuclear stations. Combustion in such compartments is often unstable and incomplete due to the relatively weak ceiling vent flow which can be corroborated by the occurrance of ghosting and oscillation flames. Here, ghosting flame is a flame that lifts off the pool surface and floats away from the burner surface and flame oscillation describes phenomenon that flame shrank to burning on part of the pool. The complicated interactions between horizontal vent flow and thermal made the resultant fire behavior difficult to determine. A serial of researches have been carried out by the State Key Laboratory of Fire Science with focuses on the temperature distributions, horizontal vent flow, and also the fire behaviors [1, 2], however, due to the difficulty of measurements, the studies were mainly on qualitative properties, lacking of quantitative analysis.

Combustion efficiency, defined as the ratio of the chemical heat of combustion to the net heat of complete combustion, is one of the most important characteristics of fire. Complete

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combustion is rarely achieved and products of incomplete combustion, such as CO and smoke, are quite common. The upper limit of the combustion efficiency is 1.00 and the lower limit is 0.46 corresponding to unstable burning leading to extinction [3]. The dominant influential factors for combustion efficiency are fuel type [4], pool size [5, 6] and also the ventilation [7]. In compartment fires, the ventilation condition is the dominant factor. The combustion efficiency would decrease to as low as 0.40 approaching flame extinction in enclosures [7]. Some analytical models have been established for combustion efficiency, such as Takeda's model [8] which was not practical. Based on a great number of hood experiments and compartment fire experiments, Tewarson *et al.* [7] found the strong dependence of combustion efficiency on the equivalence ratio (ER) and thus proposed a prediction model for combustion efficiency.

In order to describe the ventilation for compartment fires, serval parameters were proposed, such as opening factor for wall-vented compartments [9]. Similarly, He *et al.* [10] advanced a similar parameter to describe the ventilation for ceiling vented compartment, which is of the same dimension as the *opening factor*. Another concept, ER was also introduced referring to different control volume. When referring to the compartment, global equivalence ratio (GER) is then derived. Utiskul and Quintiere [11] adopted the GER to analyze the fire behavior of poorly wall-vented compartment fires. Also, Chen *et al.* [12] applied GER to ceiling vented compartment fires. Considering the possibility that the air-inflow cannot reach the flame to apply the control volume to the fire plume, plume equivalence ratio (PER) was then introduced. Tewarson's correlation is based on this concept and its usefulness was confirmed by Pitts [13].

However, it is almost impossible to measure PER in real fires due to the complex interactions among the vitiated environment, the vent flows and the flame. Yuan *et al.* [14] tried to estimate the PER for pool fires using fire plume models together with the oxygen concentration taken beside the fuel pan. This method had no direct relationship with the vent size, but the correlating gas temperature and the oxygen concentration which were both related to the vent size. This method is valid not only for ceiling vented enclosure fires, but also for closed enclosure fires. Unfortunately, this method was not validated due to the limitation on the measurements of the combustion efficiency.

To conclude, combustion efficiency of ceiling vented compartment fires is still not clearly stated, although several efforts have been conducted. Starting from existing theories and models, this paper aims to explore a way to estimate the combustion efficiency without any measurement, and to provide an approach to understand and predict the fire hazards in ceiling vented compartment fires.

Methods

The central aspect of this calculation method is the derivation of the PER which quantifies the ventilation of the flames. The combustion efficiency can be obtained through Tewarson's correlation accordingly. In order to determine the PER, oxygen concentration, fuel mass loss rate (MLR), together with the temperature distribution need to be given first.

Combustion efficiency based on the concept of plume equivalence ratio

Combustion efficiency, χ_c, is widely assumed as constant during the burning procedure. We tried to introduce a method to estimate the transient combustion efficiency reflecting the comprehensive effects of reducing oxygen concentration and the strengthened thermal environment. For a given fire, ventilation is the key influence factor for its combustion efficiency. Tewarson *et al.* [7] found the relationship between the combustion efficiency and the ventilation of the fire for non-halogenated fuels:

$$\chi_c = \chi_{c,\infty}\left[1-\frac{0.97}{\exp\left(\dfrac{\phi}{2.15}\right)^{-1.2}}\right] \tag{1}$$

where ϕ denotes the PER. By definition, PER is the ratio of fuel MLR to the mass of oxygen entrained into the fire plume. Referring to the flame, PER can be written:

$$\phi = \frac{\dfrac{\dot{m}_f}{\dot{m}_{O_2,act}}}{\dfrac{\dot{m}_f}{\dot{m}_{O_2,stoich}}} = \frac{\dot{m}_f}{\dot{m}_{O_2,act}}\Psi_O = \frac{\dot{m}_f}{\dot{m}_{ent}Y_{ox}}\Psi_O \tag{2}$$

where Ψ_O [gg^{-1}] is the stoichiometric oxygen-to-fuel mass ratio with the value of 3.52 for heptane, Y_{ox} – the oxygen concentration of the entrained gas, and $\dot{m}_{ent}$ [gs^{-1}] – the plume entrain rate. Measurement of oxygen consumption rate $\dot{m}_{O_2,act}$ is also impossible especially in ceiling vented compartment fires, thus Yuan *et al.* [14] tried to estimate the changing procedure of PER using fire plume models.

According to strong plume theory proposed by Heskestad and Hamada [15], entrainment rate is correlated to the convective heat release rate and the environment temperature. In the scope of zone models, environment temperature is regarded as the average temperature of the control volume, T_g, thus the entrainment rate below flame height, $\dot{m}_{ent,H_f}$, is given:

$$\dot{m}_{ent,H_f} = 0.878\left[\left(\frac{T_{H_f}}{T_g}\right)^{5/6}\left(\frac{T_g}{\Delta T_{H_f}}\right)+0.647\right]\frac{\dot{Q}_c}{c_p T_g} \tag{3}$$

For many kinds of fuel, ΔT_{H_f} is about 500 K. In Yuan's model, two parameters are required in order to give the value of PER, fuel MLR and oxygen concentration, which are both connected to the temperatures.

Fuel mass loss rate

The fuel MLR in compartment could be estimated in terms of free burning rate with the effects of oxygen concentration and temperature, as given by Rangwala [16]:

$$\dot{m}_f = \dot{m}_{f,\infty}\frac{Y_{ox}}{Y_{ox,\infty}} + \frac{\sigma A_f (T_g^4 - T_\infty^4)}{\Delta H_v} \tag{4}$$

where $\dot{m}_{f,\infty}$ is the free burning rate, $Y_{ox,\infty}$ – the ambient oxygen concentration, and T_∞ – the ambient temperature. For heptanes, ΔH_v is 0.48 kJ/g [11, 14]. The T_g is the average gas temperature in the compartment. Thus, $\dot{m}_f$ is correlated to the oxygen concentration and gas temperatures and the heat release rate is given as $\dot{Q} = \chi_c \dot{m}_f \Delta H_{th}$, in which ΔH_{th} is the theoretical heat of combustion.

Oxygen concentration

In the scope of zone models, oxygen concentration is assumed to be uniformly distributed in the control volume. The change of Y_{ox} is caused mainly by the fire and the horizontal vent flows, so we can get:

$$m\frac{\mathrm{d}Y_{\text{ox}}}{\mathrm{d}t} = -\frac{\dot{Q}}{\Delta H_{\text{ox}}} - \dot{m}_{\text{s}}Y_{\text{ox}} + \dot{m}_{\text{a}}Y_{\text{ox},\infty} \tag{5}$$

The MFR of smoke and fresh air are denoted as $\dot{m}_{\text{s}}$ and $\dot{m}_{\text{a}}$, respectively. Horizontal vent flows are difficult to quantify in real fire scenarios, and all the existing theories were established based upon salt-water experiments. In these theories, flow patterns and flow rates are related to pressure difference, ΔP_{o}, and density difference, $\Delta\rho$, of the two fluids across the vent, such as Cooper's [17] and Epstein's [18] models which provide a practicable tool to investigate the features of horizontal vent flows in ceiling vented compartment fires. For fire conditions, several studies have been conducted for the purpose of validating the theory [2], providing detailed flow fields of the ceiling vent flow and its interactions with the hot smoke inside the enclosure [19, 20]. Chow and Gao [21] successfully employed Cooper's theory to analyze the exchange flows of the ceiling vents in large atrium fires.

In fire cases, ΔP_{o} is caused by thermal expansion and $\Delta\rho$ is expressed by the local gas temperature following the ideal gas assumption. In the scope of zone model, ΔP_{o} is given in the terms of Q_{net} and T_{g}:

$$\Delta P_{\text{o}} = \frac{1}{2\rho}\left(\frac{\dot{Q}_{\text{net}}}{c_p T_{\text{g}} A_{\text{o}} C_d}\right)^2 \tag{6}$$

where Q_{net} is the heat absorption rate by the gas, given as $Q_{\text{net}} = Q(1 - \lambda)$, and λ is the heat loss fraction, A_{o} – the ceiling vent area, C_d – the flow factor with the value of 0.6 [18], and T_{g} – the average temperature in the enclosure. In addition, ignoring the temperature difference at the horizontal direction, the local gas temperature below the horizontal vent, T_{v}, is assumed as the average temperature of the ceiling jet. Following the ideal gas assumption, $\Delta\rho$ can be written:

$$\Delta\rho = \rho_\infty - \rho_v = \rho_\infty\left(1 - \frac{T_\infty}{T_{\text{v}}}\right) \tag{7}$$

With the knowledge of T_{g} and T_{v}, horizontal air-inflow rates and smoke outflow rates can be calculated according to Cooper's calculation model. Detailed calculation procedure can be found in [17].

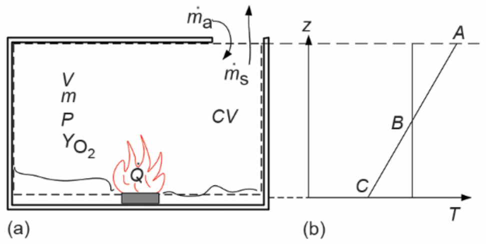

Figure 1. Control volume and basic assumption of temperature distribution

Gas temperature

The prediction of T_{g} and T_{v} would impact on the accuracy of oxygen concentration by the estimation of ceiling vent flows, fig. 1(a). For such ceiling vented compartment fires, smoke filling stage and quasi-steady stage can be observed, which are divided by the moment of the smoke layer descending to the floor. It has been widely observed in extremely limited ventilated compartment fires [10, 14, 22] that the temperature approximately rises linearly with height after the relatively short period of smoke filling stage, as illustrated in fig. 1(b).

In order to predict the average temperature, T_g, several types of zone models have been established referring to different types of ventilation [23, 24]. For fires in ceiling vented compartment fires, T_g can be basically derived from the energy conservation.

$$T_g = T_0 + \frac{\int_0^t \dot{Q}(1-\lambda)\mathrm{d}t}{\rho V c_p} \tag{8}$$

Yuan *et al.* [25] established a simple mathematical model to estimate the smoke filling time and the transient temperature distributions. The average temperature in the ceiling jet is assigned to T_v, which is also related to T_g using Heskestad's ceiling jet model [15]. The detailed computing method of the temperature distribution can be found in [25]. In the calculation, it should be emphasized that heat loss fraction, λ, changes from 0.70 [25, 26] in the smoke filling stage to 0.95 [25, 27] in the quasi-steady stage.

Description of experiment

Heptane pool fire experiments carried out previously by Chen [1] and Li [26] in a small-scale ceiling vented cabin were studied in depth using the prediction method. The dimensions of the small cubic enclosure were 1.00 m (L) × 1.00 m (W) × 0.75 m (H) with only vent located in one corner of the ceiling. The enclosure was constructed with 0.005 m thick steel boards except of one sidewall built of glass with the same thickness for the purpose of observing the fire behavior inside. Two sizes of heptane pool fires and vents were studied in the present work, with pool diameter, *D*, of 0.10 m and 0.141 m and vent sizes, *L*, of 0.05 m and 0.15 m.

An electronic balance, two arrays of thermocouples and one gas analyzer were used as shown in fig. 2. The electronic balance was set to monitor the total fuel consumption, thereby providing the time history of the fuel MLR. One array of thermocouples was set at

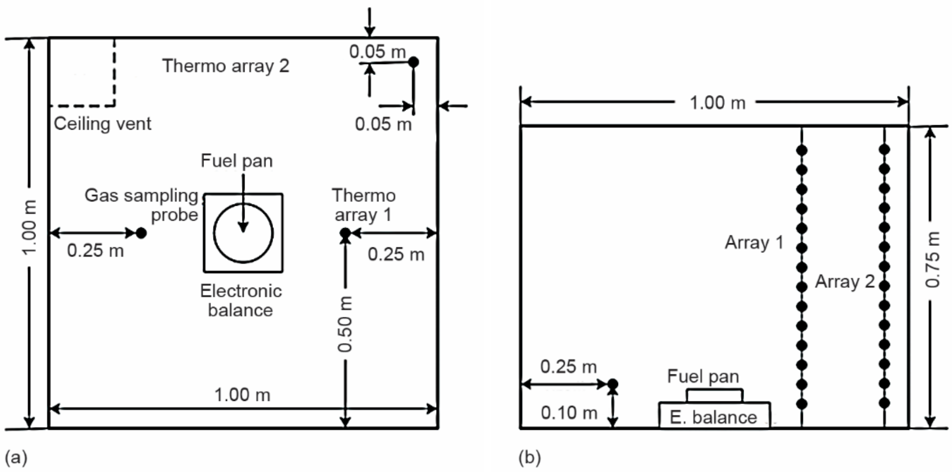

Figure 2. Experimental configuration [1, 26]; (a) top view, (b) side view

0.50 m and 0.25 m from two connecting walls and the other was located in the corner with distances from the neighboring walls of 0.05 m. Each array contained 14 thermocouples with equal spacing of 0.05 m from 0.10 m to 0.70 m height. In addition, gas sampling probe was installed near the fuel pan in order to measure the oxygen concentration of the entrained gas by the pool fire.

Some distinct flame behaviors were observed in the experiments. For experiments with L = 0.05 m and 0.15 m, flame would extinct due to oxygen starvation. Ghosting and oscillating flames occurred casually in the late period of burning in some conditions, which indicates the burning became very incomplete and unstable.

Results and discussion

In this part, the accuracy of the prediction model based on the concept of PER was studied firstly. Then, we tried to adopt this method to study the combustion efficiencies of ceiling vented compartment fires with different sizes of the pools and openings.

Validation of the combustion efficiency prediction model

Although Yuan *et al.* [14] proposed the calculation method for combustion efficiencies in ceiling vented compartment fires, comprehensive validation work has not been carried out owing to the difficulty of measurements of the transient distributions of oxygen concentrations, vent flows. Here, a simple validation was conducted using data taken from a sealed compartment fire test carried out by Zhang *et al.* [28], as illustrated in fig. 3.

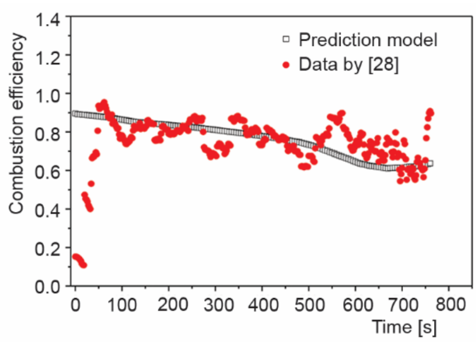

Figure 3. Comparison of predicted and measured combustion efficiencies of heptane pool fire in a sealed compartment

In fig. 3, Zhang *et al.* [28] measured the combustion efficiencies for heptane pool fires in a sealed compartment. The estimated combustion efficiencies was calculated using Yuan's model [14] in together with the recorded $\dot{m}_f$ and oxygen concentrations taken beside the fuel pan. In experiment, combustion efficiency fluctuated in the range of 0.6 and 1.0 except for the initial stage after ignition. The predicted combustion efficiency was initially 0.92, and then declined to as low as 0.61, and then slightly increased to 0.76 approaching the extinction. By comparison, although the model failed to capture the complex fluctuations, the range of the predictions was in good consistence with measurement and so was the trend. In general, prediction values matched the measurements very well.

Gas temperature

In the model, the estimations of oxygen concentration and fuel MLR were both correlated to the temperature predictions, here gas temperatures were studied first of all. Detailed validation work can be found in [25]. Here, taking D = 0.141 cm tests for example, both T_g and T_v of the predictions and measurements were compared as presented in Figure 4.

The average temperature, T_g, served as the ambient temperature in the calculation of T_v. Smoke filling time surved as the turning point of two stages of the termperature development. The comparisons suggested that the predicted smoke filling time is reasonable to divide the burning procedure into two stages. Since heat loss fraction, λ, had considerable impacts on T_g, two values, 0.70 and 0.95, were selected for the smoke filling and quasi-steady stages, which was in accordance with Chen *et al.* [27] findings. For D = 0.141 m tests, the smoke filling times were 39 seconds and 41 seconds, respectively. Generally, λ = 0.95 provided good predictions in the quasi-steady stage and the maximum error was 4.97%. While in the smoke filling stage, evident discrepancies were observed. In the calculation, a constant value of 0.70 was assigned to the heat loss fraction λ in the smoke filling stage. Indeed, λ kept increasing before the quasi-steady state as revealed by Chen *et al.* [27]. Also, λ is a function of the enclosure aspect ratio. The value of 0.70 was concluded from experiments carried out in an enclosure with aspect ratio 0.6 [29]. In the present cases, the aspect ratio of the model compartment was 2.5 suggesting that a larger value should be selected. Another reason for the inaccuracy was that the initial growing period of $\dot{m}_f$ after ignition was not taken into consideration in this model, the resultant heat release rates were somehow over-predicted consequently. Considering the fact that the growing stage was rather short, such discrepancies were still acceptable.

For T_v, as also shown in Figure 4, in the quasi-steady stage, prediction curves fit the measurements quite well, with a margin of error of 5.32%, and the major discrepancies also occurred in the smoke filling stage. Three reasons should be accounted for the divergences. Firstly, HRR was over-estimated to a large extent for ignoring the initial growth stage of the burning rate. Secondly, the predicted T_g was higher than its actual value, and this would further expand the disparities as Yuan *et al.* [25] elaborated. In addition, the measured profile taken from two thermocouples at z = 0.70 m and the computed profile were the average temperatures of the ceiling jet in the vertical direction.

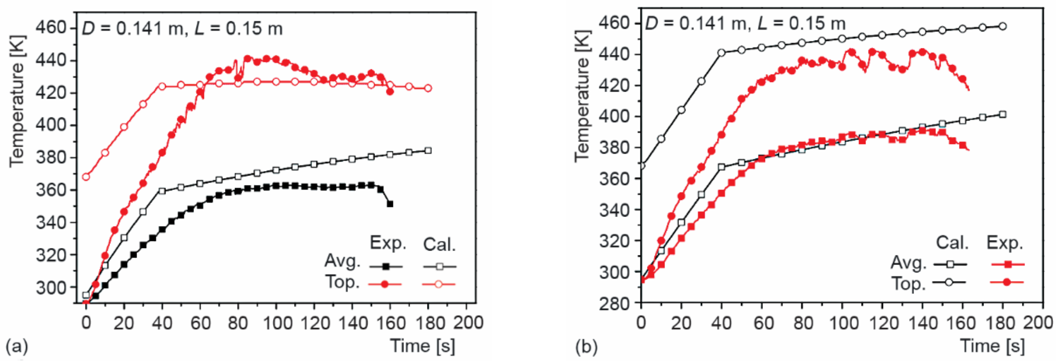

Figure 4. Comparison of predicted and measured average and top temperature for *D* = 0.141 m, *L* = 0.05 m, and *D* = 0.141 m, *L* = 0.15 m

Fuel mass lost rate and oxygen concentration

The predicted profiles of $\dot{m}_f$ and Y_{ox} were compared with the measurements to validate the feasibility of our model. In the cases, the free burning rate $\dot{m}''_{f,\infty}$ was assigned 0.01 kg/sm^{-2} [4]. The comparisons of the predictions (gray line) and measurements (dark line) of $\dot{m}_f$ in the four tests were presented in Figure 5. Obviously, the computed MLR changed differently right after ignition since the initial growth of MLR after ignition were not included in the model. This was the major inaccuracy source for this model.

Although both oxygen concentration and temperature have impacts on the computation of MLR. The computed MLR declined continuously along with the decreasing oxygen concentration as shown in fig. 5, suggesting the temperature factor rarely contribute to the value of MLR.

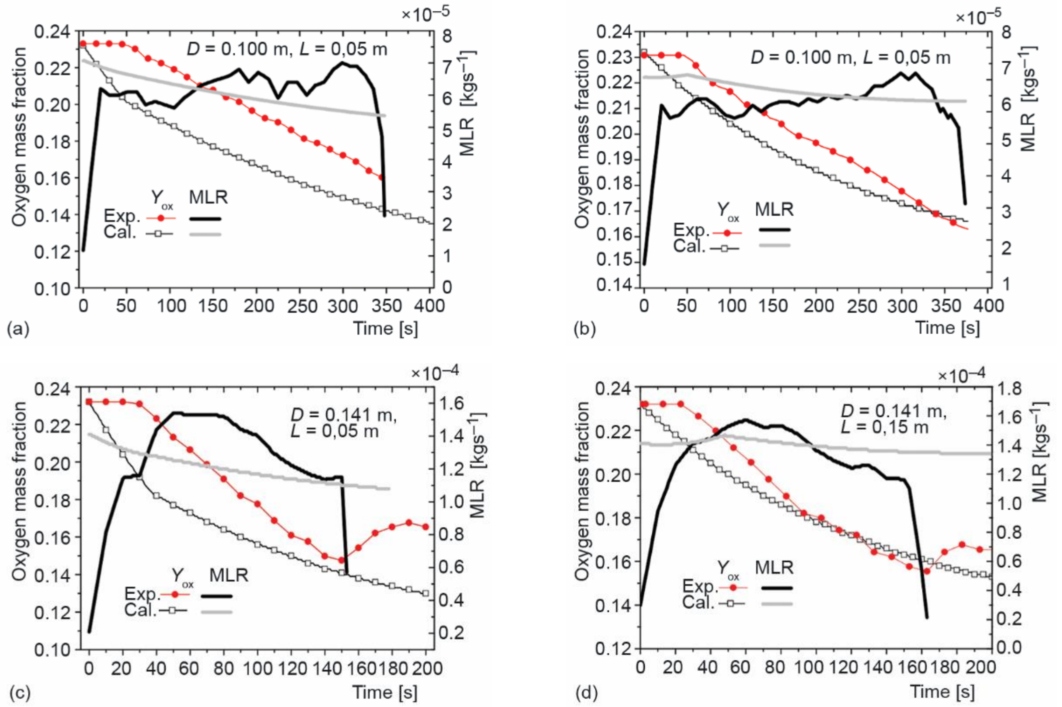

Figure 5. Comparison of predicted and measured oxygen mass fractions and MLR for (a) *D* = 0.100 m, *L* = 0.05 m, (b) *D* = 0.100 m, *L* = 0.15 m, (c) *D* = 0.141 m, *L* = 0.05 m, and (d) *D* = 0.141 m, *L* = 0.15 m

The computed Y_{ox} profiles were also compared with measurements in Figure 5. In the scope of zone model, Y_{ox} is the average oxygen concentration in the control volume, while the measurements were taken near the pool rim $Y_{ox,LL}$. Chen [1] pointed out that the average Y_{ox} Y_{ox} should be lower than $Y_{ox,LL}$. Moreover, the $Y_{ox,LL}$ kept unchanged before the smoke descended to the location of the sampling probe, and Y_{ox} began to decrease after ignition. In the figure, the largest discrepancies between the predicted and measured Y_{ox} were 13.9% and 12.3% when vent size was 0.05 m. And with larger vent size, the curves fit much better. Generally, this deviation was acceptable for engineering purpose.

Flow patterns at the horizontal vent had obvious impacts on the trends of the predicted Y_{ox} profiles. For small vent size as vent size of 0.05 m, the flows transferred from unidirectional to bidirectional and retarded the decreasing of Y_{ox} greatly because of the air--inflows. While for larger vent size, the tendency kept unchanged because the ceiling vent flow was bidirectional since very beginning.

Combustion efficiency

With the knowledge of MLR, oxygen concentration and gas temperatures, PER can be estimated accordingly, further combustion efficiencies can be given using Tewarson's function, as shown in fig. 6. The PER was below 1.0 at the ignition and this meant the ventilation was good. It began to rise and would exceed 1.0 quickly indicating the flame turned to under-ventilated condition. With larger fire size and smaller vent size, the increasing rate of PER was evidently raised. For 0.100 m pool fires, it took 43 seconds for ventilation changing from well-ventilated to under ventilated with vent size of 0.05 m while this time was prolongated to 80 seconds with vent size of 0.15 m. This was also seen in tests with pool size of 0.141 m, the moments were 21 seconds and 31 seconds, respectively. Till extinction, PER kept rising and reached 2.4 and 1.8 for 0.100 m pool fire tests and 2.8 and 1.8 for 0.141 m pool fire tests.

In fig. 6, χ_c kept decreasing during the burning procedure and dropped faster with larger fire size and smaller vent size. Theoretically, χ_c should keep at its initial value of 0.92 before the PER exceeding 1.0, however, using Tewarson's equation, χ_c began to drop from ignition. There should be fluctuations in the combustion efficiency curves according to Zhang's measurements [28]. The present method was not able to give such detailed information but gave good estimation for the range and trends.

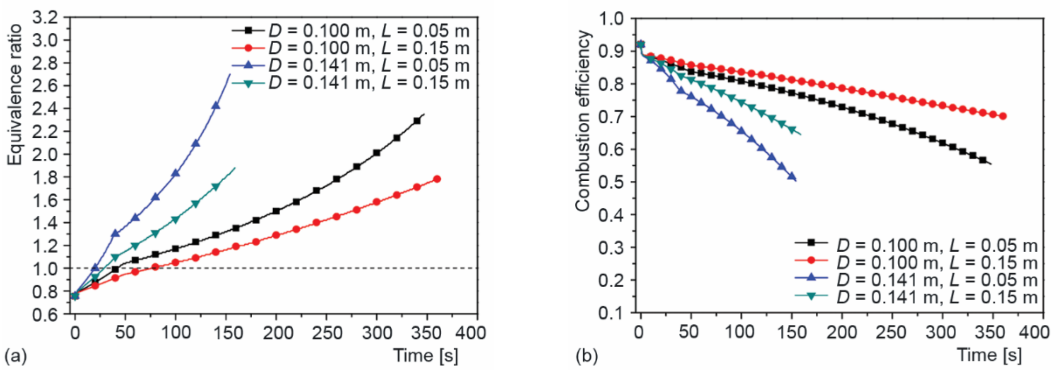

Figure 6. (a) Predicted ER (a) and (b) combustion efficiency for tests

Extinction occurred because of oxygen starvation in the four tests. The extinction time was slightly prolongated when increasing the vent size from 0.05 m to 0.15 m, while the combustion efficiency was raised evidently. In cases of 0.100 m pool fires, χ_c decreased steadily from 0.92 to 0.56 approaching extinction with vent size of 0.05 m, but this value raised to about 0.70 when the vent size increased to 0.15 m. Similarly, for the pool size of 0.141 m, χ_c was about 0.50 with vent size of 0.05 m approaching extinction, and increased to 0.65 when increasing vent size to 0.15 m. It should be noticed that in Zhang's sealed compartment fire experiment, the minimum value of χ_c was about 0.61, as shown in fig. 3. It can be inferred that fire size and compartment volume are more important than ceiling vent sizes.

Conclusions

The present study explored a comprehensive method for predicting the combustion efficiencies of ceiling vented compartment fires in the scope of zone models. Generally, in the scope of zone model, ceiling vent flow rates, oxygen concentrations, fuel mass loss rates can be estimated via temperature predictions based on the assumption of linearly inclined temperature distribution. Thus, plume equivalence ratio which characterized the ventilation of flame is derived by employing strong fire plume theories. Then, the combustion efficiency is determined according to Tewarson's model. This method provided an analytical and practical method for understanding the combustion efficiencies of ceiling vented compartment fires as their measurement was quite difficult.

Comparisons between calculations and measurements for some small scale ceiling vented compartment fires were also given in order to validate the feasibility of the method. The present model is successful in predicting the temperature distributions and oxygen concentrations with errors in acceptable range, for engineering purpose. While, when predicting the MLR, deviations can be observed after ignition because the initial growth period is ignored in the model. The main purpose of this model is to predict the combustion efficiency and the results showed that although it failed to predict the fluctuations, the value range and change trend can be predicted reasonably. In the case study, using the method proposed by this study, the transient plume equivalence ratio curves suggested the unsteady ventilation conditons for the pool fire as burning processed, and accordingly the combustion efficiency decreased along with time, which was proved by the observation of ghosting flames and oscillating flames indirectly.

Acknowledgment

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Nomenclature

A_f – area of fuel pool, [m^2]
A_o – area of ceiling vent, [m^2]
C_d – flow coefficient
c_p – specific heat
D – pool diameter, [m]
ΔP_o – pressure difference across the vent, [Pa]
L – vent size, [m]
m – mass, [kg]
$\dot{m}_f$ – fuel mass loss rate, [kgs^{-1}]
$\dot{m}_s$ – smoke outflow rate, [kgs^{-1}]
$\dot{m}_a$ – air-inflow rate, [kgs^{-1}]
$\dot{m}_{ent}$ – plume entrainment rate, [gs^{-1}]
P – pressure, [Nm^{-2}]
$\dot{Q}$ – heat release rate, [kW]
$\dot{Q}_{net}$ – net heat flux, [kW]
T_g – average gas temperature in the compartment, [K]
T_0 – initial room temperature, [K]
T_v – gas temperature at the ceiling vent, [K]
T_∞ – ambient temperature, [K]
t – time, [s]
Y_{ox} – mass fraction of oxygen
$Y_{ox,LL}$ – mass fraction of oxygen at the bottom of the compartment
$Y_{ox,\infty}$ – ambient mass fraction of oxygen
V – volume of the compartment, [m^3]
Y_{O_2} – mass fraction of oxygen, [–]

Greek symbols

ϕ – equivalence ratio
Ψ_o – stoichiometric oxygen demand, [gg^{-1}]
$\Delta\rho$ – densigy difference across the vent, [kgm^{-3}]
χ_c – combustion efficiency, [m^{2}s^{-1}]
λ – heat loss fraction
σ – Stefan-Boltzman constant, [Wm^{-2}K^{-4}]

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