Vent Flows in Fire Compartments with Large Openings

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ABSTRACT: The Kawagoe vent flow model has been widely used to study fully developed compartment fires. It is well known that the Kawagoe model is only adequate for compartments with relatively small openings because it overestimates the vent flow rates for compartments with large openings. For a compartment with a very wide opening occupying a large fraction of one wall, experimental observations show that a uniformly distributed fuel load within could produce a line fire parallel to the opening, slowly progressing into the compartment. In this article, analytical analysis has been performed to study the vent flow due to a line plume fire within the compartment, with comparisons to the Kawagoe vent flow model. The analysis shows that for large openings, the vent flow is dictated by the plume entrainment, and as the size of the opening is reduced, the flow across the opening becomes restricted by the opening geometry, approaching the flow rate predicted by the Kawagoe model. Based on the line plume entrainment model, the analysis shows that for an opening occupying one entire wall, the line plume induced flow rate could be as low as 60% of the rate predicted by the Kawagoe model. The limiting vent opening geometry for applying the Kawagoe model is suggested. Simple approximate vent flow equations for large and small openings are also presented.

KEY WORDS: fully developed fires, compartment fires, vent flows, line plumes, large fire vents.

INTRODUCTION

IT IS KNOWN that during a naturally ventilated compartment fire, outside air will be drawn into the compartment through vent openings, such as

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doors or windows. This vent flow is a direct result of buoyancy forces generated by the fire. When carrying out computer modeling of a compartment fire, an adequate description of the amount of air flowing in through the vent is required. This is particularly important during the fully developed phase of the fire, where the rate of heat release is limited by either the available ventilation or the available fuel surface area. The rate of airflow into the compartment affects the fuel mass loss rate and the heat release rate inside the compartment, and hence the resulting gas temperatures for the duration of the fire.

The amount of air flowing into the fire compartment varies at different stages of the fire. During the early stages, a small fire could be represented as an axis-symmetric plume from a single point source rising toward the ceiling with a smoke layer underneath the ceiling level. The amount of air being drawn into the compartment is largely dependent on the entrainment of the fire plume, where the outside air enters through the opening before being entrained into the rising plume. As the burning progresses, the hot upper layer deepens with increasing temperatures. It is usually assumed that a hot layer temperature of 600°C represents the transition from a growing fire to a full room-involvement fire, termed 'flashover' [1]. After flashover, all available fuel surfaces will be ignited, creating more turbulence, and leading to a reasonably 'well-mixed' environment inside the compartment. At this stage, the induced airflow becomes dependent on the temperature and resulting buoyancy inside the compartment with the flow restricted by the size of the vent opening.

This 'well-mixed' condition has been the underlying assumption made by Kawagoe [2] during his pioneering work in vent flow modeling. Assuming a 'well-mixed' fire environment within the fire compartment and an outside space with a uniform ambient temperature, the vent flows are treated as driven by the buoyancy forces generated from the hydrostatic pressure difference between these two quiescent environments. Based on this assumption, the airflow induced into the compartment has been shown [3] to be weakly dependent on the temperature (for temperatures $> 500\,\mathrm{K}$) but strongly dependent on the geometry of the vent opening, which is characterized by the parameter, $A_v\sqrt{H_v}$. This parameter is generally known as the ventilation factor, or more appropriately, the geometrical vent parameter, where it describes the physical aspect of the vent opening with A_v (m^2) being the vent opening area and H_v (m) being the vent opening height. For moderate sized vent openings, the induced air inflow rate can be shown to be $\approx 0.5 \times A_v\sqrt{H_v}$ (kg/s) [1]. The value, 0.5 is regarded as the 'ventilation coefficient' that along with the parameter, $A_v\sqrt{H_v}$, describes the fire induced ventilation into the compartment. This approximation of induced airflow, based upon the 'well-mixed' assumption, has been widely used in the study of fully developed compartment fires. In particular, it confirms Kawagoe's burning rate correlation [4] of $0.09 \times A_v\sqrt{H_v}$ (kg/s) for burning of wood cribs in a fully developed compartment fire (as discussed in [1]).

However, several researchers have found that the Kawagoe vent flow model tends to overestimate the actual vent flow rate, particularly for an opening occupying a full wall. Babrauskas and Williamson [5] suggested that a factor of 0.5 should be applied to provide a satisfactory match with experimental data. Thomas et al. [6] in their analysis of post-flashover compartment fires with small and large openings have associated two flow behaviors for small and large openings. It was indicated that at small openings, vent flows are predominantly driven by the hydrostatic pressure differences between hot gases within and cold ambient air outside; whereas at large openings, vent flows become dominated by smaller pressure differences associated with entrainment. A recent small-scale experiment conducted by Thomas and Bennett [7] using trays of liquid fuel along the width of the compartment has shown different flow behaviors for partial and full wall openings. In their experiments, the opening height was approximately the compartment height, and the opening width was varied. They found that for a full width opening, the vent flows in and out were essentially two-dimensional; but as the opening width was reduced, the vent flow became three-dimensional due to flows around the vertical edges of the opening. Preliminary analysis by Thomas [8,9] has suggested that the fuel mass loss rate within the fire compartment can be roughly differentiated into two categories, for a full wall opening and a partial wall opening. Such an association between fuel mass loss rates and different ventilation openings has also been discussed by Thomas et al. [6]. These observations suggest that the fire behaviors resulting from induced airflow for a partial wall opening and for a full wall opening are different.

It is noted that many of the fully developed compartment fire experiments (including [7]) were conducted using fuels uniformly distributed over the entire floor area, with flames seen across the full width of the compartment. In such cases, the fuel closest to the opening will burn first, with flames forming a line across the width of the opening. Excessive amounts of fuel could be released within the compartment under radiation effects, where fuels unburned flow out of the compartment under convection and burn when mixed with air outside, resulting in external flaming. The burning will progress into the compartment as the fuel closest to the vent opening is consumed. This burning behavior occurs because the burning of fuel closest to the opening forms a line of flame, known as a 'line fire,' which denies the fuel deeper in the compartment any access to the incoming air.

This is particularly evident in a long and deep compartment with large openings, such as those described in [7,10].

The observation of a line fire offers some clues on the different vent flow characteristics between small and large openings. If a fire could be represented as a plume rising from a line source over the full width of a compartment with a full wall opening, all of the incoming air into the compartment will be entrained into the line plume. As the opening width is reduced, the line plume will try to entrain more air than the amount physically able to enter through the given opening, causing more circulation and mixing. This can explain the two- and three-dimensional flow behavior observed by [7].

A study of vent flow for 'large' and 'small' vent openings and the resulting mass loss rate due to fuel burning has shown a mirroring trend between vent flow behavior and fuel mass loss rate characteristics [11].

This article presents an analytical study of vent flows induced by line plumes using a stratified 'two-zone' model. The analysis gives the rate of airflow in terms of relevant geometrical ratios and the corresponding effects. Comparison is made with the Kawagoe vent flow model.

LINE PLUME ANALYSIS

Background

As an example of a burning line fire, Figure 1(a) shows the photograph of a fire in a compartment with the opening occupying the entire side of one wall [12]. In the region visible in the photograph, the fire environment inside the compartment can be approximated as a stratified two-layer system, with ambient cold air being entrained into the fire plume and a hot layer flowing

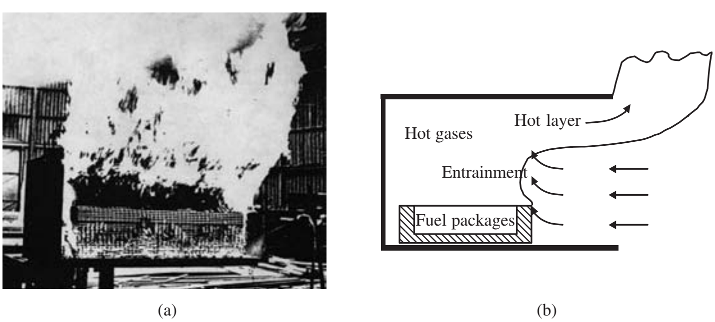

Figure 1. *Compartment fire with full wall opening: (a) compartment fire with full wall opening [12] (Courtesy of BRE) and (b) schematic representation of compartment with full wall opening.*

out, as depicted in Figure 1(b). If the fuels are uniformly distributed across the width of the compartment, the flame engulfs the plane of the opening, such that the fire cuts off access of airflow from the opening to the region behind the fire, leading to a two-dimensional line plume spanning the width of the compartment.

Extension of Thomas' Theory

Thomas [13] has performed analytical studies on vent flows resulting from a two-dimensional line plume as a fire source in a compartment with a full wall opening. In the analysis below, Thomas' treatment is extended and expressed in a generalized form for various vent opening geometries.

Consider a two-layer flow system in a compartment with a single rectangular opening having an opening height of H_v and a width of W_v (into the page) as depicted in Figure 2. This stratified two-layer flow system is represented with a hot upper layer and a cold lower layer separated by a thermal discontinuity, and a line plume acting as a link between the two layers. Air from the outside flows into the compartment due to the pressure difference and is entrained into the plume. The mass flux from the plume entrainment and the fuel flux from the plume cross the interface into the hot layer. A positive pressure is generated pushing the hot layer out of the compartment to maintain the mass balance. The pressure distributions between the compartment and the outside ambience are also shown.

In Figure 2, H'' represents the height from the neutral plane in the opening to the soffit of the opening, H' is the thickness of the cold lower layer inside the compartment (with respect to the sill height), and Δ is the vertical distance from the interface between the two layers to the neutral plane in the opening. The depth of the downstand at the soffit of the opening (i.e., the overhang above the opening) is not relevant in this stratified two-layer flow system with a single opening, because the outflow is dependent on the vertical distance from the neutral plane to the soffit. However, the sill

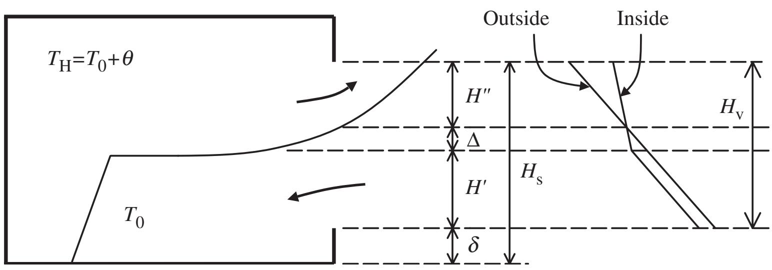

Figure 2. *Schematic representation of hot layer flowing across vent opening.*

height of the opening is important as it determines the amount of air that could be induced through the opening and entrained into the plume. As shown in Figure 2, δ represents the height of the sill above the floor. For an opening having no sill (extending down to floor level) the term δ becomes zero. The effect of sill height is discussed later in the analysis.

For a stratified two-layered environment, the equations for mass flow through a rectangular vent have been given by Rockett [3]. Using a slightly different form than used by Rockett, the inflow and the outflow rates per unit vent width, $\dot{m}'$, at the vent opening, (W_v), are given in Equations (1) and (2), respectively.

$$\dot{m}'_{\rm in} = \dot{m}_{\rm in}/W_{\rm v} = \rho_0 C_{\rm d}\sqrt{2g\frac{\theta}{T_{\rm H}}}\left(\frac{2}{3}\Delta^{3/2} + \Delta^{1/2}H'\right) \tag{1}$$

$$\dot{m}'_{\rm out} = \dot{m}_{\rm out}/W_{\rm v} = \frac{2}{3}\rho_0 C_{\rm d}\left(\frac{T_0}{T_{\rm H}}\right)\sqrt{2g\frac{\theta}{T_0}}(H'')^{3/2} \tag{2}$$

where ρ_0 is the density of the ambient air, T_0 is the ambient air temperature (lower layer), $T_{\rm H}$ is the hot layer temperature, θ is the temperature difference between the lower cold and upper hot layers, g is the gravitational constant, $9.81\,{\rm m/s^2}$, and $C_{\rm d}$ is the 'discharge coefficient' or the 'flow coefficient' of the vent opening.

The discharge coefficient is a value experimentally determined to correct for the non-idealistic flow behaviors that occur in actual flows for application in theoretical models. Analogous to orifice flows, the discharge coefficient is to account for the contraction of streamlines as they converge toward the orifice due to the fact that an instantaneous change of direction is impossible in actual flow. Experimental studies of the flow coefficient for fire induced flow through openings have been reported in the literature. Prahl and Emmons [14] performed small-scale kerosene/water experimental studies on flows through a single window and a single door opening configurations. Their results indicate that both the inflow and outflow coefficients approach a common value of 0.68 as Reynolds number increased. Steckler et al. [15] performed a full-scale fire experiment to study flow coefficients through room openings. They reported a mean value of 0.68 for the inflow coefficient and 0.73 for the outflow coefficient. Similar results have been reported by Nakaya et al. [16] where they found both the inflow and the outflow coefficients to be 0.68 during their full-scale compartment tests using a bigger fire source (which generated gas temperatures up to $\sim$1000°C). In this analysis, the inflow and outflow coefficients are assumed to have the same value.

To account for the mass loss rate of fuel, $\dot{m}_\mathrm{p}$, from the fire, let

$$\dot{m}_\mathrm{p} = (\gamma - 1)\dot{m}_\mathrm{in} \tag{3}$$

where $\gamma = 1 + s$ and s is the fuel–air mass ratio

$$s = \frac{\dot{m}_\mathrm{p}}{\dot{m}_\mathrm{in}}$$

Then, if the amount of airflow induced through the opening (through hydrostatic pressure differences or entrainment) is much greater than the amount of fuel supplied or released, then $s \approx 0$ (or $\gamma \approx 1$); and if there is substantial fuel being released compared to the air inflow, then $s > 0$ (or $\gamma > 1$).

Hence

$$\dot{m}_\mathrm{out} = \gamma \cdot \dot{m}_\mathrm{in} \tag{4}$$

A two-dimensional line plume will have entrainment on both sides. In a compartment with a single opening, the two-dimensional line plume will be entraining air from the 'front' side (at the vent opening end) and flow entrained by the 'rear' side will simply stir the gas behind the fire. This one-sided line plume is considered as having an imaginary frictionless wall at the plane of symmetry of the plume, where the entrainment by the line plume per unit of its line source is represented by half of its plume strength of $2\dot{Q}'$ [13]. This treatment is simplistic as it ignores the 'wind effect,' which could cause an increase in the plume entrainment particularly for plumes just inside the opening as the plume is blown over by the door jet, as observed by Steckler et al. [17]. The entrainment by the 'one-sided' line plume per unit length of its line source is given as (after Thomas [13]):

$$\dot{m}'_\mathrm{plume} = \dot{m}_\mathrm{plume}/W_\mathrm{plume} = \dot{m}_\mathrm{plume}/W_\mathrm{c} = \rho_0 J \left(\frac{g\dot{Q}'}{\rho_0 c_\mathrm{p} T_0} \right)^{1/3} (H' + \delta) \tag{5}$$

where J is the entrainment constant that varies with plume 'types'. Thomas [13] reported that $J = 0.365$ for the free line plume (after Lee and Emmons [18]) and $J = 0.214$ for the wall line plume (after Grella and Faeth [19]). The lower J value in the wall line plume was attributed to the wall stabilizing the plume and to a lesser extent by the wall friction effect [13]. Note that an opening with a sill height δ would provide the plume with an entraining height increased from H' to $H' + \delta$.

A line fire plume spanning across the full width of the compartment in fully developed fires is considered for analytical purposes, where the length of the line plume is taken to equal the compartment width, i.e., $W_{\mathrm{plume}} = W_{\mathrm{c}}$.

The heat balance equation within the compartment can be written by equating the fire heat release rate to the heat loss terms from the compartment. These heat loss terms include the convective heat loss due to the gas outflow, the radiative heat loss via the opening, and the heat losses to the enclosing boundaries. In an approximate form, the heat loss terms can be expressed as a function of temperature rise, θ, so that the heat loss to the enclosure boundaries becomes $h_{\mathrm{w}}A_{\mathrm{t}}\theta$, where h_{w} is the overall heat transfer coefficient to the ceiling and walls, A_{t} is the area associated with the heat transfer; and the radiation loss through the vent opening, as $A_{\mathrm{v}}\sigma T_{\mathrm{H}}^3\theta$, where $T_{\mathrm{H}}^4 \approx T_{\mathrm{H}}^3\theta$ for $T_{\mathrm{H}} \gg T_0$. Equation (6) presents the heat balance equation within the compartment, in its approximate form.

$$\dot{Q} \approx \dot{m}_{\mathrm{out}}c_{\mathrm{p}}\theta + h_{\mathrm{w}}A_{\mathrm{t}}\theta + A_{\mathrm{v}}\sigma T_{\mathrm{H}}^3\theta \tag{6}$$

From the conservation of mass equation, the mass outflow equals the mass inflow plus the fuel mass loss rate, i.e., $\dot{m}_{\mathrm{out}} = \gamma \cdot \dot{m}_{\mathrm{in}}$ as given in Equation (4). Normalized by the convection term, $\gamma\dot{m}_{\mathrm{in}}c_{\mathrm{p}}\theta$, Equation (6) becomes

$$\dot{Q} \approx \gamma\dot{m}_{\mathrm{in}}c_{\mathrm{p}}\theta\left(1 + \frac{h_{\mathrm{w}}A_{\mathrm{t}}}{\gamma\dot{m}_{\mathrm{in}}c_{\mathrm{p}}} + \frac{A_{\mathrm{v}}\sigma T_{\mathrm{H}}^3}{\gamma\dot{m}_{\mathrm{in}}c_{\mathrm{p}}}\right) \tag{7}$$

or

$$\mu\dot{Q} \approx \gamma\dot{m}_{\mathrm{in}}c_{\mathrm{p}}\theta$$

where

$$\mu = \left(1 + \frac{h_{\mathrm{w}}A_{\mathrm{t}}}{\gamma\dot{m}_{\mathrm{in}}c_{\mathrm{p}}} + \frac{A_{\mathrm{v}}\sigma T_{\mathrm{H}}^3}{\gamma\dot{m}_{\mathrm{in}}c_{\mathrm{p}}}\right)^{-1} \tag{8}$$

The term μ can be regarded as a factor that describes the fraction of the convective heat loss in the gas outflow to the total heat released in the fire. For negligible energy transfer to the boundaries and no radiation loss via opening, μ is equal to unity; otherwise, μ is less than unity. The effects of these heat loss terms on the μ factor will be discussed later.

Expressing Equation (7) in terms of the heat release rate per unit length of the line source, $\dot{Q}'$, where the length of the line source is assumed to equal the width of the compartment, W_c, gives

$$\begin{aligned} \mu\dot{Q}' &= (\gamma\dot{m}_{in}/W_c)c_p\theta \\ &= (\dot{m}_{out}/W_c)c_p\theta \end{aligned} \tag{9}$$

Substituting Equation (9) into Equation (1) to eliminate θ, and using Equation (4) to express Equation (1) as outflow per unit vent width, $(\dot{m}_{out}/W_v)$, gives

$$\begin{aligned} (\dot{m}_{out}/W_v)\left(\frac{1}{\gamma}\right)^{2/3} &= \rho_0\left(\frac{g\dot{Q}'}{\rho_0 c_p T_0}\right)^{1/3}(2\mu)^{1/3}(C_d)^{2/3}\left(\frac{T_0}{T_H}\right)^{1/3}\left(\frac{W_c}{W_v}\right)^{1/3} \\ &\times\left(\frac{2}{3}\Delta^{3/2}+\Delta^{1/2}H'\right)^{2/3} \end{aligned} \tag{10}$$

Similarly, substituting Equation (9) into Equation (2) gives

$$(\dot{m}_{out}/W_v) = 2\left(\frac{C_d}{3}\right)^{2/3}\left(\frac{T_0}{T_H}\right)^{2/3}\rho_0\left(\frac{g\dot{Q}'}{\rho_0 c_p T_0}\right)^{1/3}(\mu)^{1/3}\left(\frac{W_c}{W_v}\right)^{1/3}H'' \tag{11}$$

Assuming that there is no mixing between the two layers, the conservation of mass requires that the sum of the entrainment into the plume and the fuel pyrolysis rate has to equal the outflows through the opening. This is a simplistic approach as it ignores the mixing at the interface between the two layers due to shear as observed by Quintiere et al. [20]. Expressed in terms of outflow per unit vent width, $(\dot{m}_{out}/W_v)$, Equation (5) can be expressed as

$$H' = \frac{(\dot{m}_{out}/W_v)}{\rho_0\left[g\dot{Q}'/(\rho_0 c_p T_0)\right]^{1/3}}\cdot\frac{1}{\gamma J}\cdot\left(\frac{W_v}{W_c}\right) - \delta \tag{12}$$

Defining a dimensionless parameter, K, as

$$K = \frac{(\dot{m}_{out}/W_v)}{H_v\rho_0\left[g\dot{Q}'/(\rho_0 c_p T_0)\right]^{1/3}} \tag{13}$$

Rewriting Equations (10)–(12) in terms of K as follows, one gets:

$$K = \gamma^{2/3}\left(\frac{1}{H_{\mathrm{v}}}\right)(2)^{1/3}(\mu)^{1/3}(C_{\mathrm{d}})^{2/3}\left(\frac{T_0}{T_{\mathrm{H}}}\right)^{1/3}\left(\frac{W_{\mathrm{c}}}{W_{\mathrm{v}}}\right)^{1/3}\left(\frac{2}{3}\Delta^{3/2} + \Delta^{1/2}H'\right)^{2/3}$$
$$= a\left(\frac{1}{H_{\mathrm{v}}}\right)(\mu)^{1/3}\left(\frac{W_{\mathrm{c}}}{W_{\mathrm{v}}}\right)^{1/3}\left(\frac{2}{3}\Delta^{3/2} + \Delta^{1/2}H'\right)^{2/3} \tag{14}$$

$$K = 2(\mu)^{1/3}\left(\frac{C_{\mathrm{d}}}{3}\right)^{2/3}\left(\frac{T_0}{T_{\mathrm{H}}}\right)^{2/3}\left(\frac{W_{\mathrm{c}}}{W_{\mathrm{v}}}\right)^{1/3}\left(\frac{1}{H_{\mathrm{v}}}\right)H''$$
$$= b(\mu)^{1/3}\left(\frac{1}{H_{\mathrm{v}}}\right)\left(\frac{W_{\mathrm{c}}}{W_{\mathrm{v}}}\right)^{1/3}H'' \tag{15}$$

$$K = \gamma J\left(\frac{W_{\mathrm{c}}}{W_{\mathrm{v}}}\right)\left(\frac{1}{H_{\mathrm{v}}}\right)(H' + \delta) \tag{16}$$
$$= c\left(\frac{W_{\mathrm{c}}}{W_{\mathrm{v}}}\right)\left(\frac{1}{H_{\mathrm{v}}}\right)(H' + \delta)$$

where

$$a = \gamma^{2/3}(2)^{1/3}(C_{\mathrm{d}})^{2/3}\left(\frac{T_0}{T_{\mathrm{H}}}\right)^{1/3}, \quad b = 2\left(\frac{C_{\mathrm{d}}}{3}\right)^{2/3}\left(\frac{T_0}{T_{\mathrm{H}}}\right)^{2/3} \text{ and } c = \gamma J.$$

Since the opening height, H_{v}, as shown in Figure 2 is

$$H_v = H'' + \Delta + H' \tag{17}$$

substituting Equations (15) and (16) into Equation (17) gives an expression for Δ, where

$$\Delta = H_{\mathrm{v}} + \delta - \left[\frac{K}{c}H_{\mathrm{v}}\left(\frac{W_{\mathrm{v}}}{W_{\mathrm{c}}}\right)\right] - \frac{K}{b}H_{\mathrm{v}}\left(\frac{W_{\mathrm{v}}}{W_{\mathrm{c}}}\right)^{1/3}\left(\frac{1}{\mu}\right)^{1/3} \tag{18}$$

Substituting Equation (18) into Equation (14) gives an expression for K in terms of geometry variables that include the opening width ratio, $W_{\mathrm{v}}/W_{\mathrm{c}}$,*

*$W\mathrm{v}/W\mathrm{c}$ is basically the ratio of the vent opening width to the line plume length. Since in this analysis, the length of the line plume is taken as equal to the width of the compartment parallel to the vent opening (assuming burning of a uniformly distributed fuel load), the term $W_{\mathrm{v}}/W_{\mathrm{c}}$ is referred as the 'opening width ratio'.

the sill height ratio δ/H_v, and the associated constants, a, b, and c for the given temperature ratio T_H/T_0, the discharge coefficient C_d, the fuel–air mass ratio γ, and the plume entrainment constant J (as given in Equations (14)–(16), respectively), where

$$K = a(\mu)^{1/3}\left(\frac{W_c}{W_v}\right)^{1/3}\left\{\frac{2}{3}\left[1+\frac{\delta}{H_V}-\frac{K}{c}\left(\frac{W_V}{W_C}\right)-\frac{K}{b}\left(\frac{W_V}{W_C}\right)^{1/3}\left(\frac{1}{\mu}\right)^{1/3}\right]^{3/2} + \left[1+\frac{\delta}{H_V}-\frac{K}{c}\left(\frac{W_V}{W_C}\right)-\frac{K}{b}\left(\frac{W_V}{W_C}\right)^{1/3}\left(\frac{1}{\mu}\right)^{1/3}\right]^{1/2}\left[\frac{K}{c}\left(\frac{W_V}{W_C}\right)-\frac{\delta}{H_V}\right]\right\}^{2/3} \tag{19}$$

For a uniform well-mixed fire environment, where the hot layer has reached the floor level (or the sill of the opening), $H'=0$ and the flow is at its maximum for the given layer temperature. Note that this is the assumption made by Kawagoe [2] in his vent flow modeling. Similarly, this maximum flow can be expressed in terms of K. Using Equation (14) with $H'=0$ to obtain the expression for Δ, and Equation (15) for H'', substituting these expressions into Equation (17), gives

$$K_{\max} = \frac{(W_c/W_v)^{1/3}}{(1/\mu)^{1/3}\left[(1/a)(3/2)^{2/3}+(1/b)\right]} \tag{20}$$

This is the condition for the maximum flow through a given opening, hence K is written as $K_{\max}$.

The term K is written in terms of the mass outflow, $\dot{m}_{out}$. Since the present analysis is to investigate the air inflow rate across the vent opening, the expression for K given in Equation (13) can be rearranged and reduced to give the air inflow per unit opening width, ($\dot{m}_{in}/W_v$), by substituting Equations (4) and (9) into Equation (13) giving

$$\dot{m}_{in}/W_v = \frac{1}{\gamma}\cdot(\rho_0 K)^{3/2}\cdot\sqrt{\frac{g\theta}{\mu\rho_0 T_0}}\cdot\sqrt{\frac{W_v}{W_c}}\cdot H_v^{3/2} \tag{21}$$

It can be seen that the air inflow rate into the compartment is a function of the geometrical vent parameter, $A_v\sqrt{H_v} = (W_v H_v^{3/2})$. Equation (22) expresses the air inflow rate per unit of the vent parameter ($\dot{m}_{in}/A_v\sqrt{H_v}$), represented by a ventilation coefficient C, which describes the amount of airflow through an opening at various temperatures, opening geometries, and entrainment constants in a stratified two-layer flow system ($H' > 0$).

$$\dot{m}_{\mathrm{in}}/A_{\mathrm{v}}\sqrt{H_{\mathrm{v}}} = C = \frac{1}{\gamma}\cdot(\rho_0 K)^{3/2}\cdot\sqrt{\frac{g\theta}{\mu\rho_0 T_0}}\cdot\sqrt{\frac{W_{\mathrm{v}}}{W_{\mathrm{c}}}} \tag{22}$$

For a uniform well-mixed fire environment, i.e., Kawagoe's analysis, where the hot layer is considered to reach the floor level (for a door) or the sill level (for a window), H' becomes zero, and the expression for the air inflow per unit vent parameter, ($\dot{m}_{\mathrm{in}}/A_{\mathrm{v}}\sqrt{H_{\mathrm{v}}}$), under this 'well-mixed' condition is obtained by substituting the $K_{\max}$ expression in Equation (20) into Equation (21). This result is given in Equation (23). Under this well-mixed condition ($H' = 0$), the flow through an opening is at its maximum, therefore the ventilation coefficient C is denoted as $C_{\max}$.

$$\begin{aligned}\left(\dot{m}_{\mathrm{in}}/A_{\mathrm{v}}\sqrt{H_{\mathrm{v}}}\right)_{H'=0} &= C_{\max} \\ &= \frac{2}{3}\cdot C_{\mathrm{d}}\cdot\rho_0\cdot\sqrt{2g}\cdot\sqrt{1-\frac{T_0}{T_{\mathrm{H}}}}\cdot\left\{\frac{1}{\left[1+(T_{\mathrm{H}}/T_0)^{1/3}\cdot(1+s)^{2/3}\right]}\right\}^{3/2}\end{aligned} \tag{23}$$

Note that the expression of the air inflow per unit vent parameter, ($\dot{m}_{\mathrm{in}}/A_{\mathrm{v}}\sqrt{H_{\mathrm{v}}}$), has become a common first approximation approach to describe the fire induced airflow rate through an opening. Therefore, in the section to follow, this term is used to compare the calculated airflow rate induced by a line plume in a stratified environment to that by a commonly assumed 'well-mixed' fire environment (Kawagoe's model).

Effects of Heat Losses

In the above analytical formulation, using the heat balance equation, the fire heat release rate is expressed in terms of the flow rate and the temperature rise, θ, with the introduction of a factor, μ (as per Equations (6) and (7)) to account for enclosure boundary heat losses and radiative heat losses through the opening. The μ factor is defined by an expression given in Equation (8), showing that for negligible energy loss to the boundaries as well as no radiation loss via the opening, μ is equal to unity, otherwise μ is less than unity.

Considering that the air inflow per unit vent parameter could generally be expressed in terms of the ventilation coefficient, C, i.e. ($\dot{m}_{\rm in}/A_{\rm v}\sqrt{H_{\rm v}} = C$), Equation (8) could be expressed as:

$$\mu \approx \left(1 + \frac{h_{\rm w}A_{\rm t}}{\gamma c_{\rm p} C A_{\rm v}\sqrt{H_{\rm v}}} + \frac{A_{\rm v}\sigma T_{\rm H}^3}{\gamma c_{\rm p} C A_{\rm v}\sqrt{H_{\rm v}}}\right)^{-1} \tag{24}$$

The overall heat transfer coefficient, $h_{\rm w}$, depends on the convection ($h_{\rm conv}$), radiation ($h_{\rm rad}$), and conduction ($h_{\rm cond}$) into the wall. For boundaries with semi-infinite thickness, the heat loss to the boundaries, $\dot{q}''_{\rm w}$, can be written as:

$$\begin{aligned}\dot{q}''_{\rm w} &= (h_{\rm conv} + h_{\rm rad})(T_{\rm H} - T_{{\rm w}i}) = h_{\rm cond}(T_{{\rm w}i} - T_0)\\ &= h_{\rm w}(T_{\rm H} - T_0)\end{aligned} \tag{25}$$

where

$$\frac{1}{h_{\rm w}} = \frac{1}{(h_{\rm conv} + h_{\rm rad})} + \frac{1}{h_{\rm cond}}$$

Quintiere [21] has given typical ranges for these heat transfer coefficients, where $h_{\rm conv} \approx 10\text{–}30\,{\rm W/(m^2K)}$, $h_{\rm rad} \approx 5\text{–}100\,{\rm W/(m^2K)}$, and $h_{\rm cond} \approx \sqrt{k\rho c/t} \approx 5\text{–}60\,{\rm W/(m^2K)}$, with $h_{\rm cond}$ decreasing over time. For the purpose of estimating the boundary heat loss term in Equation (24), the overall heat transfer coefficient, $h_{\rm w}$, is taken to be $20\,{\rm W/(m^2K)}$ as a first approximation. By taking $A_{\rm t} = A_{\rm T} - A_{\rm v}$ with $A_{\rm T}$ being the total internal surface area, for a square compartment with a height of 3m having a vent opening comprising one full wall, $A_{\rm t}/A_{\rm v}\sqrt{H_{\rm v}}$ ranges from $4\,{\rm m}^{-0.5}$ to $5\,{\rm m}^{-0.5}$ as the compartment length changes from 5 to 10 m.

The ventilation coefficient, C, which will be discussed in more detail later, can vary from 0.25 for large openings to 0.5 for small openings. As such, with $\gamma = 1.2$ and $c_{\rm p} = 1150\,{\rm J/(kg\,K)}$, the enclosure boundary heat loss term is approximately in the order of 0.1–0.2. The radiation heat loss term is dependent on the height of the openings. Typical wall opening heights in compartments could range from 0.5 to 3 m.

Figure 3 plots the factor, μ, from Equation (24), over various temperatures with $C = 0.25$ and $H_{\rm v} = 3$ to represent large ventilation openings, and $C = 0.\ 5$ and $H_{\rm v} = 0.5$ to represent small ventilation openings. From the figure, it can be seen that μ is of the order of 0.65–0.85 over temperature ranges between 750 and 1500 K. For the purpose of presenting the analytical results, $\mu = 0.75$ will be used below.

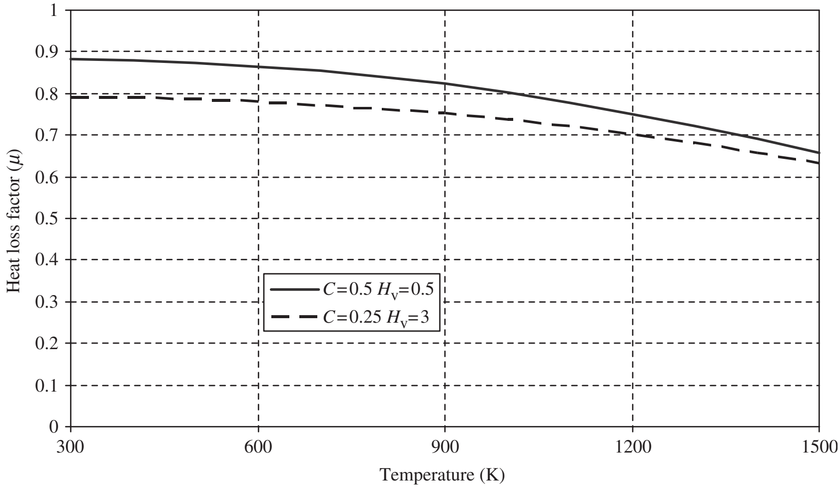

Figure 3. *The heat loss factor at various temperatures.*

ANALYTICAL RESULTS

The vent flow equations have been presented here in a generalized form, where the ventilation opening geometry is described in terms of opening width, opening height, and sill height; and the buoyancy flow is described in terms of flow temperatures, the air to fuel mass ratio, and the entrainment constant. In the sections to follow, sensitivity analyses are used to investigate how the air inflow rate is affected by different opening geometries (for both door and widow openings), with comparisons made against Kawagoe's vent flow model.

Vent without Sill (Door Opening)

Equation (22) is a generalized flow equation for a stratified two-layer flow system coupled by a line plume at various vent geometries. The line plume has been assumed to have a length equal to the compartment width ($W_{\mathrm{plume}} = W_{\mathrm{c}}$). As such, the vent flow across the opening is simply characterized by the opening width fraction, $W_{\mathrm{v}}/W_{\mathrm{c}}$, and the sill height ratio, δ/H_{v}.

Considering a wall with a door opening, i.e., no sill, hence $\delta = 0$, and using a layer temperature of 1173 K (900°C) as an example, Figure 4(a) plots the calculated air inflow per unit vent parameter, i.e., the ventilation coefficient ($\dot{m}_{\mathrm{in}}/A_{\mathrm{v}}\sqrt{H_{\mathrm{v}}} = C$), for the free line plume and the wall line plume as

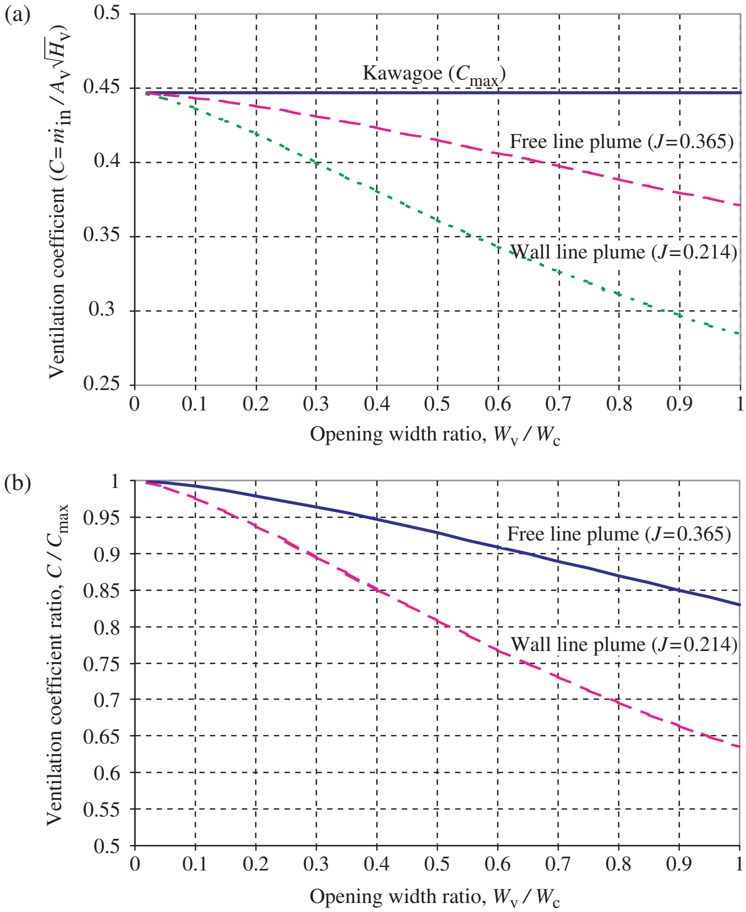

Figure 4. *The air inflow as a function of the opening width ratio, predicted using the line plume analogy and the Kawagoe model for a layer temperature of 1173 K (900°C): (a) value of ventilation coefficent, C vs opening width ratio,* W_v/W_c *and (b) value of ventilation coefficient ratio,* C/C_{max} *vs opening width ratio. (The color version of this figure is available online.)*

a function of the opening width fraction, W_v/W_c. The value of C_{max} for the maximum flow condition (i.e., the Kawagoe model per Equation (23)) is also shown.

The constants used in the calculations include: $\rho_0 = 1.2\,\text{kg/m}^3$, $g = 9.81\,\text{m/s}^2$, $\gamma = 1.2$, $C_d = 0.68$ and $T_0 = 293\,\text{K}$. For the purpose of this calculation, μ is taken as 0.75. It can be seen that at large W_v/W_c, the air inflow per unit vent parameter, C, from the line plume entrainment

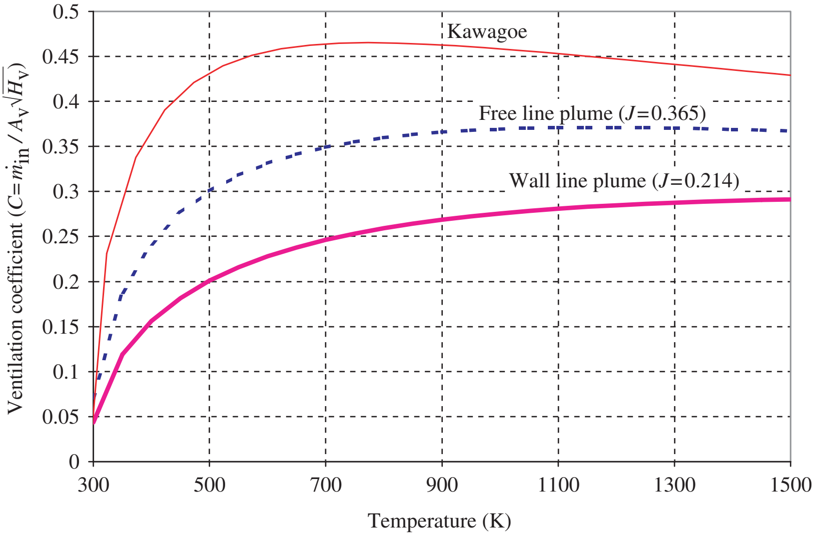

Figure 5. *Induced vent flow rates as a function of temperature for different types of flow across a vent with a full width opening and no sill. (The color version of this figure is available online.)*

is considerably less than the prediction made by the Kawagoe vent flow model.

Figure 4(b) shows the corresponding vent flow ratio $C/C_{\max}$ which is the ratio of the line plume entrainment from Equation (22) to the maximum value predicted by the Kawagoe model in Equation (23). It can be seen that for a full wall opening, i.e. $W_v/W_c = 1$, the free line plume only generates a flow rate 85% of the maximum flow. The wall time plume generates a flow rate as low as 65% of the maximum flow.

As the opening width ratio decreases, the vent flows for the line plumes approach the Kawagoe prediction. This is expected because with the reduction of the opening width ratio, the outflow through the opening is reduced. In order to maintain the mass balance in the system, which requires the entrainment into the plume plus the fuel generation rate to equal the outflow through the opening, the system adjusts itself by lowering the hot layer toward the floor level. This decreases the clear height, H', between the floor and the hot layer, hence the plume entrainment rate. Continuing to decrease the opening width will lead to the lowering of the hot layer toward the floor level approaching a uniform well-mixed environment.

Figure 5 shows the ventilation coefficient, C $(=\dot{m}_{air}/A_v\sqrt{H_v})$, calculated for the two types of line plume (Equation (22)) in a compartment with a full

Table 1. Approximate air inflow equations for a full wall opening, temperature range of 800–1500 K.

	Ventilation coefficient, C	Approximate air inflow equation
Kawagoe ($H' = 0$)	$C = 0.44$–0.46	$\dot{m}_{in} \approx 0.45 \times A_v\sqrt{H_v}$
Free line plume ($J = 0.365$)	$C = 0.36$–0.37	$\dot{m}_{in} \approx 0.36 \times A_v\sqrt{H_v}$
Wall line plume ($J = 0.214$)	$C = 0.26$–0.29	$\dot{m}_{in} \approx 0.27 \times A_v\sqrt{H_v}$

width opening having no sill, for various hot layer temperatures. Temperature has been chosen to be presented as a variable against the ventilation coefficient as typical post-flashover compartment fires reach temperatures over 600°C and beyond. The presentation of the ventilation coefficient against temperature would allow approximations to be made on the induced air flow through openings in a post-flashover compartment. The Kawagoe vent flows (Equation (23)) is also plotted for comparison purposes. The constants used in the equations are the same as described for the previous figures.

For the range of temperature (800–1500 K) of most interest, it can be seen that the vent flows across the opening for the three cases are reasonably constant. Thus, the induced airflow rate for these cases could be written in an approximate form as given in Table 1.

From Figure 5 and Table 1, it can be seen that airflow induced into the compartment through a full wall opening due to line plume entrainment is smaller than the flow rate estimated by the Kawagoe vent flow model. Table 1 shows that a wall line plume will have a flow rate of ~60% of the maximum flow rate predicted by Kawagoe's assumption of a uniform fire environment. This finding is in line with the observation by Babrauskas and Williamson [5], that for a large vent opening such as a window taking up one whole wall, the actual air inflows appear to be about 50% lower than that predicted by Kawagoe's model.

The lower flow rate estimated from a line plume in a stratified two-layer environment compared to Kawagoe's uniform well-mixed assumption can be explained by comparing the hydrostatic pressure distributions from the two assumptions. Figure 6 schematically depicts the hydrostatic pressure distribution for both the uniform ($H' = 0$) and the stratified ($H' > 0$) cases.

The pressure gradients of the inside gas layers for both cases are shown. From the mass balance, the mass flow above the neutral-plane height plus the fuel mass loss rate equals the mass flow below the neutral-plane height. The buoyancy head across the vent opening is represented by the hydrostatic

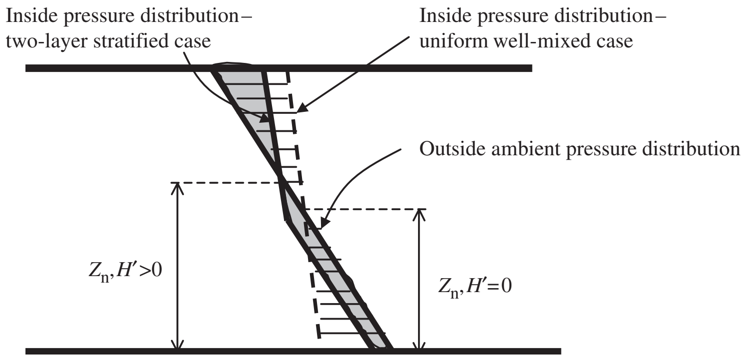

Figure 6. *Schematic representation of pressure distribution at the opening, for stratified and well-mixed cases.*

pressure difference between the inside and outside environments. It can be seen that the single zone uniform assumption (with $H' = 0$) produces the maximum buoyancy head across the opening, indicated by the larger area between the two hydrostatic pressure distributions. Since the vent flows are the result of buoyancy, the 'one-half factor' mentioned in [5] is simply used to adjust the flow rate so it matches the plume entrainment rate. It is noted that due to the nature of the assumption, the neutral-plane height (Z_n) from the uniform well-mixed assumption with $H' = 0$ is expected to be lower than the neutral-plane height evaluated from the stratified assumption with $H' > 0$.

Vent with Sill (Window Opening)

As discussed earlier, reducing the opening width will lower the hot layer toward the floor level. For a compartment with a window opening, hence a sill above the floor level, there is a limiting opening width ratio where the hot layer drops to the sill level. For any openings smaller than this limit, the flow across the opening is at the maximum, where the flow rate is restricted by the opening geometry. Any further decrease in the opening width would mean that the plume is trying to entrain more air than can get in through the opening, resulting in a lot of circulation and mixing.

Figure 7 plots the vent flow ratio as a function of the opening width ratio, W_v/W_c, for various values of sill height ratio, δ/H_v. It can be seen that for an opening with a high sill, the maximum flow condition ($C/C_{max} = 1$, per Kawagoe's model) is achieved for a much larger range of opening widths than for an opening with a low sill.

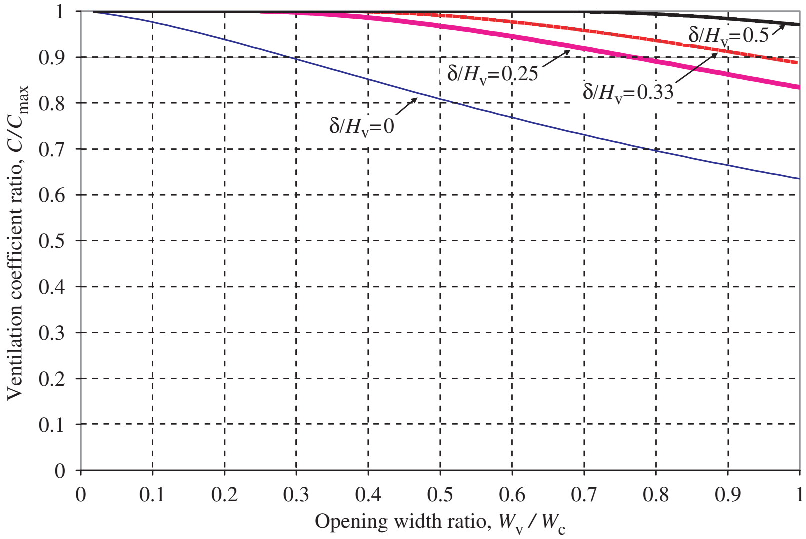

Figure 7. *The air inflow as a function of the opening width ratio, predicted using the wall line plume analogy for different sill height ratios, δ/H_v, for a layer temperature of 1173 K (900°C). (The color version of this figure is available online.)*

The limiting value of the opening width ratio, $(W_v/W_c)_{lim}$, for a given sill height ratio, δ/H_v, can be deduced by equating Equation (16) to Equation (20), with K_{max} describing the maximum flow condition at $H' = 0$. This is given in Equation (26).

$$\left(\frac{W_v}{W_c}\right)_{lim} = \frac{\gamma J}{K_{max}} \cdot \frac{\delta}{H_v} \tag{26}$$

Figure 8 plots the critical values of the opening width ratio, W_v/W_c against the sill height ratio, δ/H_v for the two different line plumes, using Equation (26). If the actual vent opening geometry (defined by the combination of W_v/W_c and δ/H_v) is smaller than the critical value, i.e., below and to the right-hand side of the appropriate line in Figure 8, the hot layer is expected to drop below the sill height of the opening. In such cases, the flow will be at the maximum, in which case the use of the Kawagoe vent flow model would be appropriate. This shows that the sill height is an important parameter that dictates the flow conditions through the vent opening.

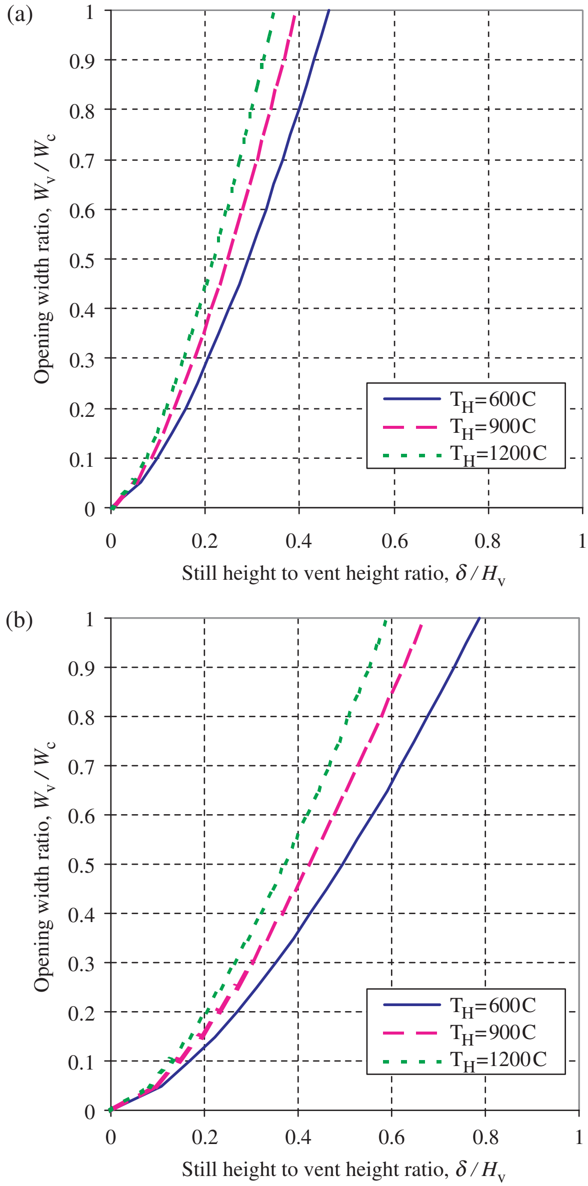

Figure 8. *Critical values of the opening width ratio W_v/W_c below which maximum vent flow will occur, as a function of the sill height ratio δ/H_v: (a) free line plume, $J = 0.365$ and (b) wall line plume $J = 0.214$. (The color version of this figure is available online.)*

The analytical results presented in Figure 7 show that the ventilation coefficient is at least 90% of the Kawagoe value for almost all opening widths ($W_v/W_c < 0.9$) if the sill height ratio δ/H_v is > 0.3. Considering that most typical windows in actual buildings are likely to have geometries within this range, the Kawagoe vent flow equation could be regarded as adequate, for engineering purposes, to describe vent flows through window-type openings.

For openings with no sill, or a small sill to opening height ratio, vent flow equations are simply characterized by the existence of a window sill, as given in Equations (27) and (28).

For window-type openings, with $\delta > 0$, the approximate vent flow equation is given by Kawagoe as:

$$\dot{m}_{in} = 0.45 \times A_v\sqrt{H_v} \tag{27}$$

For door-type openings, with no or very low sill height, the approximate vent flow equation based on the wall line plume analogy is:

$$\dot{m}_{in} = [0.45 - 0.18(W_v/W_c)] \times A_v\sqrt{H_v} \tag{28}$$

The analyses presented here used a constant flow coefficient, C_d, of 0.68 for all openings. A two-dimensional theoretical study on flow coefficients by Steckler et al. [15] indicate that the flow coefficient approaches a value of 1.0 when the opening width approaches the enclosure width. However, because their analysis has been restricted to a two-dimensional situation, the top (i.e., soffit or overhang above the opening) and bottom (i.e., sill) effects have been ignored. Analogous to flow over a weir or notch, even at full width over the enclosure width, the fluid flowing over these obstructions is expected to contract in the vertical direction and not horizontally. Therefore, the flow coefficient is expected to be less than unity. For the case where the opening constitutes one entire wall, the flow approaches the condition similar to a uniform channel flow where the flow coefficient could approach unity with minimal obstruction effects. However, it remains unclear what the effect of the opening edges are on the flows and what are the other effects neglected in the theoretical model, such as turbulence, viscosity, and three dimensionality on the fluid motion under such a condition, which determines the flow coefficient value. There appears to be very limited study for this circumstance.

Assuming that the opening edges are frictionless and those other effects remain secondary, the potential flow through a full wall opening due to a wall line plume evaluated using a discharge coefficient C_d of 1.0 is: $\dot{m}_{in} \approx 0.34 \times A_v\sqrt{H_v}$. This is higher than $\dot{m}_{in} \approx 0.27 \times A_v\sqrt{H_v}$ (Equation (28)) evaluated using $C_d = 0.68$.

Considering that most compartment opening constructions generally involve soffit/lintel above and side edges to allow for the installation of doors or windows, the analysis carried out using a discharge coefficient of $C_d = 0.68$ is considered to be representative for most openings in practical conditions.

COMPARISON WITH EXPERIMENTS

The above description of fire behavior has been based on a theoretical analysis. The vent flow behavior described by the analytical model needs to be confirmed with experimental observations. Unfortunately, vent flows induced by line plume fires under high temperatures have not been widely investigated. The closest experimental study was performed by Quintiere et al. [22] who used a line burner consisting of four contiguous sand burners supplied with methane, placed against the rear wall opposite the vent opening to simulate a wall line plume. In their experimental program, the length of the line burner (W_{plume}), the height (H_v), width (W_v), and sill height (δ) of the vent were systematically varied between experiments. Steady-state temperature and vent flows measurements were made during the experiments.

Their data show that the upper layer temperature ranged from 73 to 383°C with the majority of experiments having temperatures between 100 and 300°C. The geometrical variables tested were in the following ranges: δ/H_v ranged from 0 to 2.85 (door openings to windows at high levels); W_v/W_{plume} ranged from 0.125 to 2.15.

Figure 9 plots the measured mass flow rate of air per unit of the vent geometrical parameter, $\dot{m}_{air}/A_v\sqrt{H_v}$, i.e., the ventilation coefficient C, against the opening width to the length of the line plume ratio, W_v/W_{plume}. The data points with different sill height to opening height ratio, δ/H_v, are distinguished accordingly. Superimposed on the graph are the calculated values from the model described in Equation (22), where the following inputs are used: $\gamma = 1$ (negligible fuel mass), $\rho_0 = 1.2\,\text{kg/m}^3$, $T_0 = 293\,\text{K}$, $C_d = 0.68$, $J = 0.214$ (wall line plume), $\mu = 0.8$, with an upper layer temperature $T_H = 473\,\text{K}$.

Note that in the model development described, it was assumed that the line plume has a length equal to the width of the compartment, hence $W_c = W_{plume}$. In the reported experiments [22], the line plume length is not equal to the compartment width, $W_{plume} \neq W_c$, and therefore the term W_c in the model is replaced by W_{plume} for the calculation.

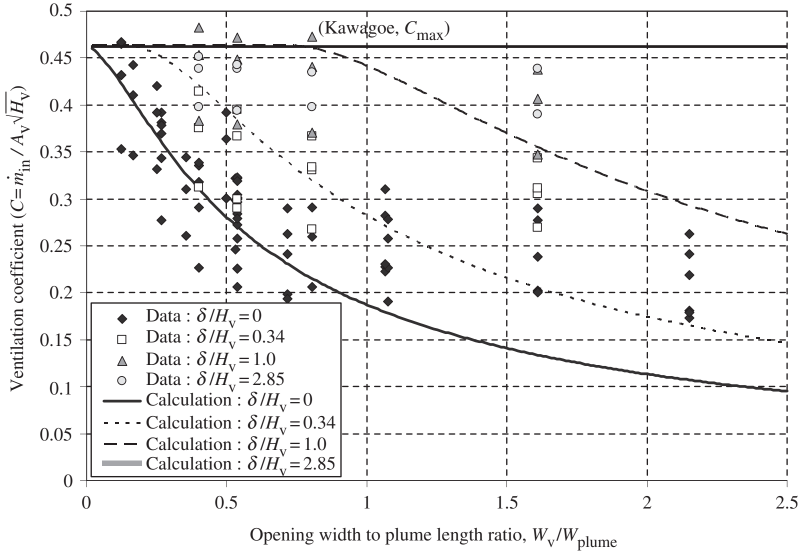

Figure 9. *Comparison between the model calculations and the experimental measurements of Quintiere et al. [22].*

Despite a lot of scatter in the experimental results, Figure 9 shows that the model produces favorable comparisons with the experiments, supporting the general trend for vent mass flows associated with different geometrical combinations, i.e., δ/H_v and W_v/W_{plume}. It confirms the vent flow behavior observed in the analytical study (Figure 7) where the vent flow rate approaches the Kawagoe vent flow limit at small opening widths and high sills.

CONCLUSIONS

This article is concerned with fully developed fires in compartments with uniformly distributed fuel load. The Kawagoe vent flow model, particularly in the form of $0.5 \times A_v\sqrt{H_v}$ (kg/s), has been widely used in the analysis of such fires. It is known that this expression is appropriate only for small openings, where the induced flow through the opening is restricted by the opening geometry. For large ventilation openings, this expression often overpredicts the actual vent flow. Considering that typical compartments often have fuels uniformly distributed over the floor area, burning after flashover will cover the full width of the compartment with the fuel closest to the opening producing a line of flames across the width of the opening.

The burning will progress away from the vent opening as the fuel closest to the opening is consumed. This burning behavior is likely to deny the fuel behind the flame access to the incoming air. This is particularly evident in a long and deep compartment with large openings as observed during experimental studies [7,10].

This article has analyzed the applicability of the Kawagoe model under different vent geometries. It presents a vent flow analysis based on line plume fires, to represent more realistic fire behaviors as reported in the literature. The analytical model developed, which is based on a number of simplifying assumptions, can be used to predict the gas flows in the ventilation opening of a fully developed fire. It produces results in agreement with findings and experimental results reported in the literature.

The analysis shows that for an opening occupying one whole wall, the vent flow is essentially entrainment driven. As the opening size decreases, the upper layer descends toward the sill level and more mixing occurs, approaching the well-mixed scenario where the size of the vent opening restricts the flow, leading to the Kawagoe vent flow limit. Hence, when analyzing fully developed compartment fires with large openings, it is necessary to consider the effect of line plume entrainment, noting that the Kawagoe vent flow equation will overestimate the induced vent flow rate.

The analysis provides a framework to understand the effect of large openings on fully developed compartment fires. Further analytical and experimental research in this area is necessary.

NOMENCLATURE

English Symbols

$A_T =$ total internal area of the enclosure including the vent opening area (m^2)

$A_t =$ area of the enclosing boundaries subjected to heat transfer (m^2)

$A_v =$ area of the vertical wall opening (m^2)

$C =$ ventilation coefficient ($= \dot{m}_{in}/A_v\sqrt{H_v}$)($kg\,s^{-1}\,m^{-2.5}$)

$C_d =$ discharge/flow coefficient of the wall opening including door or window (–)

$c =$ specific heat of the enclosure (J/(kg K))

$c_p =$ specific heat of gases (J/(kg K))

$g =$ gravitational constant, 9.81 m/s^2

$H_s =$ height of the soffit evaluated from the floor level (m)

$H_v =$ height of the vertical wall vent opening (m)

$H' =$ the lower cold layer depth inside the compartment with respect to the sill height (m) (Figure 2)
$H'' =$ the height distance above the neutral plane to the soffit of the opening (m) (Figure 2)
$h_{cond} =$ conductive heat transfer coefficient to the enclosure (W/(m^2 K))
$h_{conv} =$ convective heat transfer coefficient to the enclosure (W/(m^2 K))
$h_{rad} =$ radiative heat transfer coefficient to the enclosure (W/(m^2 K))
$h_w =$ the overall heat transfer coefficient to the ceiling and walls (W/(m^2 K))
$J =$ line plume entrainment constant
$K =$ dimensionless parameter (Equation (13))
$k =$ thermal conductivity (W/(mK))
$\dot{m}_{in} =$ mass flow rate of air into the compartment (kg/s)
$\dot{m}_{out} =$ mass flow rate of hot gases out of the compartment (kg/s)
$\dot{m}_p =$ mass loss rate of fuel inside the compartment (kg/s)
$\dot{Q} =$ heat release rate (W)
$s =$ Fuel–air mass ratio
$T_0 =$ temperature of the ambient air (K)
$T_H =$ temperature of the hot upper layer (K)
$t =$ time (s)
$W_c =$ width of the compartment (m)
$W_{plume} =$ length of the line plume (m)
$W_v =$ width of the vertical wall opening (m)
$Z_n =$ height of the neutral-plane evaluated from the floor level (m)
$Z_{n,\ H'>0} =$ height of the neutral-plane evaluated from the floor level based on the stratified (hot upper and cold lower layers) condition within the compartment (m) (Figure 6)
$Z_{n,\ H'=0} =$ height of the neutral-plane evaluated from the floor level based on the uniform temperature ('well-mixed') condition within the compartment (m) (Figure 6)

Greek Symbols

$\Delta =$ distance between the neutral plane and the base of the hot upper layer (m) (Figure 2)
$\delta =$ sill height of the vertical wall vent opening (m)
$\gamma = 1 + s$
$\mu =$ fraction of the convective heat loss in the gas outflow to the total heat released in the fire (Equation (8))
$\theta =$ temperature difference (K)

$\rho =$ density (kg/m^3)
$\rho_0 =$ density of ambient air (kg/m^3)
$\sigma =$ Stefan–Boltzmann constant (W m^{-2} K^{-4})

Superscript

$' =$ per unit length

Subscript

max = maximum flow condition (per Kawagoe's model)
lim = limit

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