On the Significance of A Wall Effect in Enclosures With Growing Fires

U.S. DEPARTMENT OF COMMERCE
National Bureau of Standards
National Engineering Laboratory
Center for Fire Research
Washington, DC 20234

June 1983
ON THE SIGNIFICANCE OF A WALL EFFECT IN ENCLOSURES WITH GROWING FIRES

Leonard Y. Cooper

U.S. DEPARTMENT OF COMMERCE
National Bureau of Standards
National Engineering Laboratory
Center for Fire Research
Washington, DC 20234

June 1983

U.S. DEPARTMENT OF COMMERCE, Malcolm Baldrige, Secretary
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, Director
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>LIST OF FIGURES</th>
<th>iv</th>
</tr>
</thead>
<tbody>
<tr>
<td>LIST OF TABLES</td>
<td>iv</td>
</tr>
<tr>
<td>Abstract</td>
<td>1</td>
</tr>
<tr>
<td>1. INTRODUCTION</td>
<td>2</td>
</tr>
<tr>
<td>2. A DESCRIPTION OF THE WALL EFFECT AND A MEASURE OF ITS SIGNIFICANCE</td>
<td>2</td>
</tr>
<tr>
<td>3. SOME PROPERTIES OF THE WALL FLOW</td>
<td>7</td>
</tr>
<tr>
<td>4. IMPLICATIONS OF THE WALL EFFECT CRITERION</td>
<td>10</td>
</tr>
<tr>
<td>4.1 Applying Results When Wall Temperatures Are Close to Ambient</td>
<td>13</td>
</tr>
<tr>
<td>5. ESTIMATING THE WALL TEMPERATURE</td>
<td>17</td>
</tr>
<tr>
<td>5.1 Wall Heating Due to Radiation</td>
<td>18</td>
</tr>
<tr>
<td>5.2 Wall Heating Due to Convection</td>
<td>22</td>
</tr>
<tr>
<td>6.1 The Quasisteady Flow Assumption</td>
<td>25</td>
</tr>
<tr>
<td>6.2 Wall Layer Penetration</td>
<td>26</td>
</tr>
<tr>
<td>7. SUMMARY AND CONCLUSIONS</td>
<td>27</td>
</tr>
<tr>
<td>8. ACKNOWLEDGMENTS</td>
<td>28</td>
</tr>
<tr>
<td>9. REFERENCES</td>
<td>29</td>
</tr>
<tr>
<td>10. NOMENCLATURE</td>
<td>30</td>
</tr>
</tbody>
</table>
LIST OF FIGURES

Figure 1. Sketch of the significant features of the wall effect .... 33

Figure 2. Plots of $\zeta_1$ as a function of $X_1$ per eqs. (23) and (24) for different values of $(\hat{m}_w/\hat{m}_p)/X_2$ and for $m = 0, 1.0$ and $2.0$ under the assumption $\delta \ll 1$ 34

Figure 3. Plots of $I_1(x;m)/I_1(0;m)$ and $I_2(x;m)/I_2(0;m)$ per eqs. (43)-(44) and (48)-(49), respectively, for $m = 0, 1.0, 2.0$ and $5.0$ 35

Figure 4. Plots of eq. (51) and (54) criteria for wall layer quasisteadiness and penetration, as functions of interface elevation and for $t^m$ enclosure fires 36

LIST OF TABLES

Table 1. Thermal conductivity and diffusivity of selected materials 37

Table 2. Results of calculations on the significance of the wall effect in full-scale experiments of references 1 and 2 .... 38
ON THE SIGNIFICANCE OF A WALL EFFECT IN ENCLOSURES WITH GROWING FIRES

Leonard Y. Cooper

Abstract

This paper studies the significance of a wall effect that has been observed during the growth stage of enclosure fires. Relative to the two-layer phenomenon which tends to develop during such fires, the effect has to do with the near-wall downward injection of hot upper layer gases into the relatively cool uncontaminated lower layer. It is conjectured that these observed wall flows are buoyancy driven, and that they develop because of the relatively cool temperatures of the upper wall whose surfaces are in contact with the hot upper layer gases. For a growing fire (growth proportional to \( t^m \); \( t \) being time and \( m \geq 0 \)) in an enclosed compartment, an analysis of the conjectured mechanism for the wall flow leads to a time-dependent solution for the ratio of wall flow mass ejection rate from the upper layer, \( \dot{m}_w \), to the fire plume mass injection rate to the upper layer, \( \dot{m}_p \). The solution indicates that in practical fire scenarios \( \dot{m}_w / \dot{m}_p \) can be of the order of "several tenths" even prior to the time that the upper layer interface has dropped to an elevation midway between the ceiling and fire. In other words, the results of the analysis indicate the importance of taking the wall effect into account in two-layer zonal analyses of enclosure fire phenomena.
1. INTRODUCTION

The zone approach to mathematical modeling has received much attention in the analysis of enclosure fire environments. In recent years several such zone models have been developed to predict environments in rooms of fire origin as well as in single and even multiple adjacent spaces. Because of the fact that they have each tended to be developed for specific purposes and classes of problems, there is quite a variation from model to model in the amount of attention and detail paid to the simulation of different physical phenomena that come into play. In this regard, however, even the most sophisticated of the models have tacitly ignored a commonly observed and potentially important wall effect which can lead to significant features of enclosure fire environments not heretofore simulated.

After describing the wall effect in question this paper will then identify the conditions under which the effect is significant. Finally, the paper will propose a general purpose algorithm for including the wall effect in an overall zonal model analysis.

2. A DESCRIPTION OF THE WALL EFFECT AND A MEASURE OF ITS SIGNIFICANCE

The wall effect under consideration is first described and analyzed within the context of a generic fire scenario which develops within an enclosure lacking any significant ventilation openings (any leakage from the
enclosure is assumed to occur near the floor). Later, the relation of the wall effect to fire environments developing in enclosures with open doors or windows will be addressed.

A sketch of the significant features of all the conjectured wall effects to be described below is presented in figure 1.

A fire starts a distance \( H \) below the ceiling of an enclosure of floor area, \( A \), and releases energy at the rate \( Q(t) \), where \( t \) is time from the effective instant of ignition. Because of their elevated temperature, the products of combustion from the fire are driven upwards by buoyancy forces, and a turbulent plume is generated. All along the axis of the plume, quiescent ambient air is entrained laterally into and mixed with the plume gases. The plume eventually impinges on the ceiling and spreads radially outward along this surface forming a relatively thin, turbulent ceiling jet. The ceiling jet gases redistribute themselves across the entire ceiling area of the enclosure, start to fill the upper enclosure volume, and eventually submerge all the continuous ceiling jet flow activity. Below this jet activity the now reduced momentum plume gases form a relatively quiescent, elevated temperature, upper gas layer which continues to increase in depth as the plume gas upward filling process continues in time. The elevation above the fire of the lower interface of this gas layer is designated by \( Z_i(t) \).

Except for regions near the bounding ceiling and wall surfaces, it is reasonable to characterize the upper layer as being uniform in composition with an absolute temperature, \( T_u(t) \) and with concentrations of products of combustion \( i \), \( C_i(t) \) (units of product per unit mass of bulk upper layer mixture).
The ceiling surface temperatures are generally different from $T_u$. This surface is heated from the initial ambient temperature, $T_\infty$, by convection from the high temperature ceiling jet gases and by radiation from the fire's combustion zone. Later into the fire, radiation exchanges from the bulk upper layer gases at $T_u$ and from other bounding surfaces of the enclosure may also become significant.

In general the temperatures of the wall surfaces are also different from $T_u$. The radiant heating mechanism of these surfaces is similar to that for the ceiling. However, except for possible locations of strong ceiling jet-wall surface flow interactions (i.e., where wall surfaces are close enough to the plume-ceiling impingement point), the vigorous convective heat transfer at the ceiling is generally absent at the walls. Thus, one can anticipate that the wall surfaces will be heated from the initial temperature, $T_\infty$, at a significantly lower rate than the ceiling surface.

It is conjectured and seems to have been observed$^{1,2}$ that the condition of relatively cool walls, at absolute temperature, $T_w$, bounding the elevated temperature upper layer gases, at $T_u$, would lead to the development of a classic, two-dimensional, downward-directed, natural convection or buoyancy-driven boundary layer flow along the upper portion of the vertical wall surfaces. The phenomenon would occur in this basic form away from locations of vigorous ceiling jet-wall interactions. The boundary layer flow would originate near the ceiling, increase in mass and momentum flux with decreasing elevation, and, at the upper layer - lower layer interface, be injected into the lower layer of the enclosure as a downward-directed wall jet of upper layer gases. The injected gases would have a bulk temperature lower than the
upper-layer gases but higher than the relatively cool and uncontaminated lower-layer gases.

Once in the lower layer, the downward-directed wall jet of the now upwardly-buoyant, product of combustion laden gas would be buoyed back upward and away from the wall to either mix with and contaminate the lower layer air or to entrain additional (i.e., in addition to the fire plume) lower layer air into the upper layer.

Whatever the eventual disposition of the wall flow contaminants, the basic wall flow phenomenon could significantly alter the rate of development of life-threatening conditions in the enclosure. If, for example, the flow is buoyed upward as a wall plume and re-enters the upper layer, then, having entrained additional lower layer air, the depth of the upper layer will grow more rapidly than it would have otherwise, albeit at a reduced temperature and product concentration. On the other hand, if the wall flow is mixed with the lower layer gas, then the net effect would be a contamination of the lower layer, less rapid growth in upper layer depth and at increased temperature and concentration.

When the mass flux leaving the upper layer in the wall flow is large enough to significantly retard the velocity of descent, $dZ_1/dt$, of the upper layer - lower layer interface or to lead to significant increases in temperature and/or product of combustion contamination of the lower layer, then this flow must be taken account of in a mathematical simulation of the overall enclosure environment.
Let \( \dot{m}_p \) be the mass flux of the plume into the upper layer, i.e., at \( Z_4 \). Also, let \( \dot{m}_w \) be the total mass flux of the wall boundary layer flow ejected at \( Z_4 \) from the upper layer. Then, consistent with the above, an analysis of the enclosure environment which does not include an accounting of the wall flows can lead to reliable results only when \( \dot{m}_w \ll \dot{m}_p \). Thus when \( \dot{m}_w / \dot{m}_p \ll 1 \) the wall effect can be neglected, and when this is not the case it must be taken into account. This will be referred to below as the Wall Effect Criterion.

It is noteworthy that the described wall flow phenomenon would not necessarily occur only in rooms of fire origin. Indeed, it can also play an important role in the redistribution of the products of combustion in spaces which are adjacent to, or communicate with rooms-of-fire-origin. Here, the fire generated plume in the above room-of-fire-origin scenario would be replaced, for example, by an analogous upper doorway plume which was generated by inflowing combustion products. "Smoke layering" in the adjacent space would be initiated in the usual way. Also, at least until relatively late into the fire when radiative heating of adjacent space wall surfaces could become substantial, these surfaces, being only weakly heated by the conjectured buoyancy driven wall flows, would likely remain close to their original ambient temperature.

The above discussion is from the perspective of the early growth stage of the fire when a definitive \((T_u - T_w)\) temperature differential exists, when no significant differential between the lower wall temperature and the lower gas temperature, \( T_L \), (both of which are initially at \( T_w \)) has developed, and when the overall wall flow activity, as described earlier, is conjectured to occur. As time moves on, however, a substantial difference between \( T_L \) and the
(probably) relatively greater lower wall temperature (heated primarily by radiation) may begin to develop. At least near the floor of the enclosure this would (probably) tend to lead to an upward-directed, lower wall flow. It is also possible that such a lower wall flow would become relatively strong at a time when the downward-directed, upper wall flow starts to decrease in strength by virtue of a reduction in \((T_u - T_w)\). Under such a circumstance, the nature of the interaction of such combined, counter-current wall flows could be important in an adequate description of the overall enclosure environment; the early growth stage description of a dominant upper wall flow would have to be revised.

An analysis of the combined wall flow effect, which may come into play at later stages of the fire, is outside the intended scope of the present paper. The combined flow effect will not be discussed further except to note that a study has been recently initiated on the significance of the phenomenon in ventilated enclosure fire scenarios during steady state conditions.\(^3\)

3. SOME PROPERTIES OF THE WALL FLOW

Refering to the wall flow depicted in figure 1, the total mass flux, \(\dot{m}(x)\), at a distance \(x = H - Z\) below the ceiling (\(Z\) above the fire) can be estimated, for Prandtl Number \((\text{Pr}) = 0.72\) and according to whether the boundary layer is laminar or mostly turbulent, from

\[
\dot{m}(x) = \left\{ \begin{array}{ll}
1.70 \mu \infty Gr_x^{1/4} & \text{Laminar: } Gr_x < 0.5(10^9) \\
0.102 \mu \infty Gr_x^{2/5} & \text{Turbulent: } Gr_x > 1.5(10^{10})
\end{array} \right.
\]
where \( P \) is the total length of the perimeter of the enclosure, \( \text{Gr}_x \) is the Grashoff number

\[
\text{Gr}_x = \frac{(T_u - T_w)g x^3}{T^2 v^2} = (\phi_u - \phi_w)(1 - \zeta)^3 \frac{gH^3}{v^2}
\]

and where dimensionless values of \( T_u, T_w, \) and \( Z \) have been defined as

\[
\phi_u = \frac{T_u}{T_\infty}; \quad \phi_w = \frac{T_w}{T_\infty}; \quad \zeta = \frac{Z}{H}
\]

The above estimates for \( \dot{m}(x) \) were established from results presented in references 4-6. The \( \text{Gr}_x \) bounds for a laminar or mostly turbulent boundary layer are from reference 7. The Bousinesq approximation was used in developing eq. (1), and will be used throughout the analysis to follow. All gas properties will be taken as those of air at standard ambient conditions. The wall temperature is taken to be uniform.

From the origin of the boundary layer to some \( x \) above the interface \((0 < x < H - Z_i)\), the average rate of heat transfer per unit area, \( \overline{q}'' \), to the wall can be estimated from

\[
\overline{q}'' = \bar{h} (T_u - T_w)
\]

where

Average Nusselt Number =

\[
\overline{\text{Nu}}_x = \frac{\bar{h} x}{k} = \begin{cases} 0.48 \text{ Gr}_x^{1/4}, \text{ Laminar}^5 \\ 0.0184 \text{ Gr}_x^{2/5}, \text{ Turbulent}^6 \end{cases}
\]
and where $k$ is the thermal conductivity of the gas mixture. Below the interface the heat transfer rate will be less than $q''$.

The mixing cup temperature, $T_m$, of the wall flow at position $x$ is defined by

$$T_u - T_m = \frac{q'' \rho x}{\dot{m} C_p} \tag{6}$$

where $C_p$ is the specific heat.

Using $Pr = 0.72$, the results of eqs. (1), (4) and (5) in eq. (6) lead to the following estimate for $T_m$:

$$\frac{T_u - T_m}{T_u - T_w} = \begin{cases} 0.39, \text{ Laminar} \\ 0.25, \text{ Turbulent} \end{cases} \tag{7}$$

This last result indicates that $T_m$ is independent of $x$ and somewhat closer to $T_u$ than to $T_w$. In particular, eq. (7) is valid at the interface position $x = x_i = H - Z_i$.

The kinematic momentum flux, $K$, of the wall flow

$$K = \int_0^\infty u^2 \, dy \tag{8}$$

where $u$ is the downward velocity and $y$ is the distance from the wall surface, can be estimated from results of integral boundary layer analyses in references 6 and 7. Thus, for $Pr = 0.72$
Finally, the order of magnitude of the downward velocity will be of interest in later discussion. From results in references 4 and 6 it is possible to obtain the result that for Pr = 0.72 the maximum velocity in the boundary layer, \( u_{\text{max}} \), for both laminar and mostly turbulent flow can be estimated from

\[
u_{\text{max}} = 0.55 v_\infty \frac{\text{Gr}}{x}^{1/2}
\]  

(10)

4. IMPLICATIONS OF THE WALL EFFECT CRITERION

The mass flux of the plume at the upper layer - lower layer interface can be estimated from

\[
\dot{m}_p = 0.21 \rho_\infty g^{1/2} H^{5/2} (1 - \lambda_r) \frac{1}{3} Q_0^{*1/3} \frac{\xi_1 5/3}{1/3} \psi^{1/3}
\]  

(11)

where

\[
\xi_1 = \frac{Z_1}{H}, \psi = \frac{Q(t)}{Q_0}, Q_0^* = \frac{Q_0}{\rho_\infty C_p T_0 g^{1/2} H^{5/2}}
\]  

(12)

and where \( Q_0 \) is a characteristic energy release rate, and \((1 - \lambda_r)\) is the fraction of \( Q \) which effectively acts to heat the plume gases and ultimately drive the plume's upward momentum. \( \lambda_r \) is approximately the fraction of \( Q \) lost by radiation from the combustion zone and plume. For hazardous flaming fires, \( \lambda_r \) is typically in the range 0.3 - 0.4.8 In real fires, the simple
description of Eq. (11) has been shown to yield excellent results above the combustion zone when the "point" location is taken to be at the lowest elevation of the combustion zone or plume above it where "free unrestricted", lateral entrainment of ambient air is possible.\textsuperscript{8,10,11}

In order to invoke the Wall Effect Criterion, eqs. (1) and (11) are used to estimate $\dot{m}_w/\dot{m}_p$ with the result

$$
\frac{\dot{m}_w}{\dot{m}_p} = \frac{\alpha \text{Gr}_n(P/H)}{0.21(1-\lambda_r)^{1/3} Q_o^{1/3} \zeta_1^{5/3} (gH^3/v_\infty^2)^{1/2} \psi^{1/3}}
$$

where

$$Gr = Gr_x(x = x_1 = H-Z_1) = (\phi_u - 1)(1-\delta)(1-\zeta_1)^3 gH^3/v_\infty^2$$

$$\delta = (\phi_w - 1)/(\phi_u - 1)$$

$$\alpha = 1.70, n = 1/4 \text{ if } Gr < 0.5(10^9), \text{ Laminar}$$

$$\alpha = 0.102, n = 2/5 \text{ if } Gr > 1.5(10^{10}), \text{ Turbulent}$$

To apply eq. (13) to the dynamic fire scenario discussed in the last section it is now necessary to estimate the values of $\zeta_1(t)$, $\phi_u(t)$ and $\phi_w(t)$.

In the absence of any wall effect, and from ignition ($\zeta_1 = 1$) up to the time that the interface elevation starts to approach the elevation of the fire (i.e., $\zeta_1$ becomes small), the solutions for $\zeta_1(t)$ and $\phi_u(t)$ have been previously obtained\textsuperscript{10} for enclosure fires where $Q(t) \sim t^m$ (arbitrary $m \geq 0$),
and where any pressure relieving leakage from the enclosure is primarily near the lower portions of the enclosure boundary. Under such circumstances

\[ \phi_u = \left[ 1 - (m + 3) \varepsilon \tau^{3(m+1)/(m+3)} / [3(m+1)(1 - \xi)] \right]^{-1} \]

where

\[ \varepsilon = (1 - \lambda_c) \left[ \left( \frac{m+3}{3} \right)^{m}\left( \frac{1}{Q_0} \right) \right]^{2/(m+3)} / (1 - \lambda_r)^{(m+1)/(m+3)} \]

\[ \tau = 3 \left[ (1 - \lambda_r)Q_0 \right]^{1/3} \left( \frac{tH^{3/2} g^{1/2}}{A} \right)^{(m+3)/3} / (m+3) \]

and where \( Q_0 \) is explicitly defined as

\[ Q_0 = (Q/t^m) \left( \frac{H^{3/2} g^{1/2}}{A} \right)^{-m} \]

Also, \( \lambda_c \) is defined as the fraction of \( Q \) which is instantaneously transferred to the internal bounding surfaces of the enclosure. \( \lambda_cQ \) would include all heat transfer to surfaces by both convection and radiation, and \( \lambda_c \) is typically in the range \( 0.6 - 0.9 \). For magnitudes of energy release rates and enclosure sizes of practical interest, \( \varepsilon \ll 1 \) and \( \xi \) can be estimated from

\[ \lim_{\varepsilon \to 0} \xi = \left( 1 + 0.140\tau \right)^{-3/2} \left[ 1 + 0(\varepsilon) \right] \]

for \( \xi \) and \( m \) which include the ranges \( 0.5 \leq \xi \leq 1 \) and \( 0 \leq m \leq 2 \). (For a more definite idea of the utility of the eq. (21) approximation refer to figure 2 of reference 12.) Unless noted otherwise all further estimates in this paper will only be applicable when the above approximation yields an acceptable estimate for \( \xi(\tau) \).
Using eqs. (17) and (21), and eliminating the explicit dependence of $\phi_u$ on time leads to

$$\lim_{\varepsilon \to 0} (\phi_u - 1) = \frac{\varepsilon(m+3)}{3(m+1)(1-\zeta_i)} \left( \frac{\varepsilon_i^{-2/3} - 1}{0.140} \right) 3(m+1)/(m+3) + O(\varepsilon)^2$$

(22)

Using eq. (22) in eq. (14) and rewriting eq. (13) eventually leads to

$$\frac{\dot{m}_w}{\dot{m}_p} = \frac{\alpha Gr^n X_2}{0.21 x_1 1/2 \zeta_i^{5/3}} \left( \frac{0.140}{\zeta_i^{-2/3} - 1} \right)^{\frac{m}{(m+3)}}$$

(23)

$$Gr = Gr(\zeta_i; X_1, m, \delta)$$

$$= \frac{(1-\delta)(m+3)}{3(m+1)} X_1(1-\zeta_i)^2 \left( \frac{\zeta_i^{-2/3} - 1}{0.140} \right) \frac{3(m+1)}{(m+3)}$$

(24)

where $\alpha$ and $n$ are given in eq. (16), and where $X_1$ and $X_2$ are defined according to

$$X_1 = \frac{g e H^3}{v_f^2} ; X_2 = \left( \frac{1 - \lambda_c}{1 - \lambda_f} \right)^{1/2} \left( \frac{P}{H} \right)$$

(25)

### 4.1 Applying Results When Wall Temperatures Are Close to Ambient

During the early growth stage of the fire the increase of $T_w$ above its initial value, $T_{\infty}$, will often develop at a much slower rate than will the increase of $T_u$. Thus, for the purpose of evaluating the significance of the wall effect, it is reasonable to investigate the implications of eqs. (23) and (24) under the condition

$$\delta \ll 1$$

(26)
According to these equations, and for \( m = 0, 1.0, \) and 2.0, figure 2 presents plots of \( \zeta_1 \) as a function of \( X_1 \) with \( \left( \frac{\dot{m}_w}{\dot{m}_p} \right)/X_2 \) as a parameter, under the condition \( \delta = 0 \). Thus, for \( 0 \leq m < 2.0 \), and for a specified \( X_1 \) fire scenario, the plots of figure 2 can be used to determine the value of \( \zeta_1 \) below which the wall effect starts to become significant (i.e., \( \frac{\dot{m}_w}{\dot{m}_p} \) becomes larger than, say, several tenths).

The results of figure 2 will be used to evaluate the significance of the wall effect during test runs of two full-scale enclosure fire test programs reported in references 1 and 2. As mentioned earlier, the wall effect seems to have played a role in those experiments.

The first evaluation refers to a constant 16.2 kW enclosure fire reported in reference 2. The fire was an acetylene diffusion flame generated from a gas burner source 2.06 m below the ceiling and in the center of a square, "fully" enclosed space (designed to leak from below) with sides of length 3.64 m. Tests on the burner indicated a 50% radiation loss from the sooty flame \( (\lambda_r = 0.5) \). Total loss to the ceiling and walls was estimated to be 84\% \( (\lambda_c = 0.84) \). In terms of previous definitions, and using the values \( T_\infty = 2940^\circ K, \rho_\infty = 1.2 \) kg/m\(^3\), \( v_\infty = 1.5 \left(10^{-5}\right) \) m\(^2\)/s, \( C_p = 240 \) cal/(kg\(^\circ\)K), and \( g = 9.8 \) m/s\(^2\) here and in all later calculations, the test run is found to be characterized by

\[
\begin{align*}
m &= 0, \quad \varepsilon = 0.0036, \quad X_1 = 1.4(10^9), \quad X_2 = 4.0
\end{align*}
\]
Now assume the wall effect to be significant when $\dot{m}_w/\dot{m}_p > 0.3$, i.e.,
$(\dot{m}_w/\dot{m}_p)/X_2 > 0.075$. Then figure 2 for $m=0$ leads to the conclusion that the
wall effect must be taken account of once $z_4$ drops to 0.62, i.e., once the
upper layer thickness exceeds 0.78 m.

The remaining evaluations will refer to test runs reported in reference 1
in a 2.36 m high, fully enclosed two to three room space (designed to leak
from below), where one of the rooms, having an area of 14.0 m$^2$ and designated
as the burn room, contained a centrally located methane burner 2.12 m below
the ceiling. The one to two rooms adjacent to the burn room were corridor-
like in configuration, and the total multi-room test space area ranged from
40.6 - 89.6 m$^2$. For test runs with all rooms freely connected (i.e., full
open doorways), data indicated that after an initial time interval a single
room, two layer mod had some applicability in analyzing the dynamic conditions
within the overall space. It is therefore reasonable to utilize the present
room-of-fire-origin results in an evaluation of the significance of the wall
effect under consideration. As mentioned earlier, the existence of the wall
effect was observed during the course of both the reference 1 and reference 2
test programs.

The first evaluation of the above test series will involve the smallest
constant fire, 25 kW, and the largest test space area, 89.6 m$^2$ of the test
program. This configuration had a total wall perimeter of approximately
81 m. Previous tests on the burner had indicated $\lambda_r = 0.19$, and data from
vertical arrays of thermocouples indicated an average value $\lambda_c = 0.87$. Using
all of the above data the test run is found to be characterized by
Again, assuming the wall effect to be significant when \( \frac{m}{m_p} > 0.3 \), i.e., 
\( \frac{m_w}{m_p} \cdot X_2 > 0.020 \), figure 2 leads to the conclusion that the wall effect
must be taken account of once \( \xi_1 \) drops to 0.85, i.e., once the upper layer
thickness exceeds 0.32 m.

The second evaluation of the test series will involve the largest
constant fire, 225 kW, and the smallest test space area, 40.6 m², of the test
program. This configuration had a total wall perimeter of approximately
43 m. Tests on the burner indicated \( \lambda = 0.24 \) for burner fires in the range
50 to several hundred kW, and test run data indicate an average value
\( \lambda_c = 0.75 \). All of this leads to

\[
m = 0, \, \varepsilon = 0.027, \, X_1 = 1.1(10^{10}), \, X_2 = 12.\]

Using the previous criterion and figure 2, the wall effect is predicted to be
significant here once \( \xi_1 < 0.76 \).

The final two test run evaluations will refer to the largest and smallest
area configurations where the burner was controlled to generate a fire growing
linearly with time according to

\[
Q(t) = (t/2) \, \text{kW/s}
\]  \hspace{1cm} (27)

Thus, \( m = 1 \), and according to eq. (20)
\[ Q_0 = \frac{A}{(2H^{3/2}g^{1/2})} \text{ kW/s} \]  \hfill (28)

First the largest space \((A = 89.6 \text{ m}^2, P = 81 \text{ m})\): Here \(\lambda_F = 0.24\), and test data indicate an average value \(\lambda_c = 0.81\). This leads to

\[ m = 1, \varepsilon = 0.0064, X_1 = 2.6(10^9), X_2 = 39. \]

Using the previous criterion and the \(m = 1.0\) plots of figure 2, the wall effect is predicted to be significant once \(\zeta_1 < 0.93\).

Finally, the smallest space \((A = 40.6 \text{ m}^2, P = 43 \text{ m})\): Here \(\lambda = 0.24\), and test data indicate an average value \(\lambda_c = 0.71\). This leads to

\[ m = 1, \varepsilon = 0.0065, X_1 = 2.7(10^9), X_2 = 27. \]

From figure 2, the wall effect is predicted to be significant once \(\zeta_1 < 0.90\).

5. ESTIMATING THE WALL TEMPERATURE

Figure 2 and the above example calculations were based on the assumption \(\delta \ll 1\) of eq. (26). In this section estimates of \(\delta\) will be developed so that the validity of this assumption can be evaluated, and so that, when required, estimates of \(m_w/m_p\) under somewhat more general nonambient wall temperature conditions can be obtained from eqs. (23) and (24).

In order to predict \(\delta\), estimates for \(\phi_w - 1\) under each of two different wall heating conditions will be established; increases of \(\phi_w - 1\) due to
possible radiation from the fire's combustion zone, and increases of $\phi_w - 1$ at the upper wall surfaces due to convective heating from the wall layer flows. For the purpose of this investigation it is assumed that for the fire's growth stage of present interest, and far enough away from significant ceiling jet-wall flow interaction, one of these two mechanisms will dominate the increase of $\phi_w$.

5.1 Wall Heating Due to Radiation

Increases in $\phi_w$ due to radiation may be the more significant of the heating mechanisms for enclosure spaces which are configured in such a manner that a large fraction of the total wall surface is illuminated by the fire's combustion zone. Under such circumstances and for the purpose of estimating $\phi_w - 1$ and comparing it to $\phi_u - 1$, assume the fire to be a characteristic distance $(A/\pi)^{1/2}$ from all wall surfaces. Then the average rate of wall heat transfer due to radiation from the combustion zone, $q_R^\prime$, can be estimated by

$$q_R^\prime = \frac{\lambda_r Q(t)}{4A}$$

(29)

For the purpose of the present estimate, first consider wall materials which are thermally thick up to some time, $t_I$, of interest. To be definite, it is reasonable to consider a wall material as thermally thick if its thickness, $L_{THICK}$, and thermal diffusivity, $\kappa_w$, satisfy

$$L_{THICK} > 2\sqrt{\kappa_w t_I}$$

(30)
For \( t_1 \) of 3 minutes, and using material properties from table 1, the criterion would be satisfied, for example, with gypsum board walls thicker than 0.011 m or concrete walls thicker than 0.055 m.

Using the thick wall assumption and eq. (29) in Carslaw and Jaeger's solution\(^1\) to the appropriate, specified flux, heat conduction problem for the wall leads to the result

\[
\phi_w - 1 = \frac{\lambda_T k_w^{1/2} q_0}{4T_w \pi^{1/2} A k_w} \int_0^\infty \frac{\psi(t - \eta)}{\eta^{1/2}} d\eta
\]

where \( k_w \) is the thermal conductivity of the wall material.

For the \( t^m \) fire growth studied earlier, eq. (31) can be integrated to yield

Radiation to Thermally Thick Wall:

\[
\phi_w - 1 = 0.14 X_3^{1/4} \frac{\epsilon (m+3)^{1/2} \Gamma(m+1)}{\Gamma(m+1/2)} \left( \frac{-2/3 - 1}{0.140} \right)^{3(2m+1)} \]

(32)

\[
\delta = 0.43 X_3^{1/4} \left( \frac{m+1}{\Gamma(m+1/2)} \right) (1-\xi_1) \left( \frac{0.140}{\xi_1^{-2/3} - 1} \right)^{3/2(m+3)}
\]

(33)

where

\[
X_3 = \frac{\lambda_T}{(1-\lambda_T)} \left( \frac{1-\lambda_T}{1-\lambda_c} \right)^{5/4} \left( \frac{\kappa_w}{\kappa} \right)^{1/2} \left( \frac{k}{\kappa_w} \right)^{Pr^{1/2} H/A^{1/2}}
\]

(34)
Γ(x) is the Gamma function, \(X_1\) is defined in eq. (25), and \(κ\) and \(κ\) are the thermal diffusivity and conductivity of air at ambient conditions.

The above result can now be applied to the previously discussed 16.2 kW enclosure fire (gypsum board wall) of reference 2. The geometry of fire scenario is one where all walls of the enclosure are illuminated by the fire, and where radiation can be anticipated to dominate the wall surface heating. Using the previously identified parameters of that fire, and the \(κ\), \(κ\) values of table 1 in eq. (33) leads to

\[
δ = (\phi_w - 1)/(\phi_u - 1) = 0.29 \text{ at } \tau_i = 0.62
\]

Note that this ratio is small enough to allow the \(δ = 0\) estimate of figure 2 to be useful. Indeed, if the approximation of eq. (26) is supplanted by the newly revised estimate

\[
\phi_u - \phi_w = (\phi_u - 1)(1 - δ) = 0.71(\phi_u - 1)
\]

then eqs. (23) and (24) would predict \(\dot{m}_w/\dot{m}_p = 0.28\) at \(\tau_i = 0.62\), instead of the earlier estimate of \(\dot{m}_w/\dot{m}_p = 0.30\).

If the wall of the reference 2 enclosure had been concrete instead of gypsum board (and if \(λ_c\) would not have been significantly different from the measured value of 0.84), then eq. (33) leads to the estimate

\[
δ = (\phi_w - 1)/(\phi_u - 1) = 0.09 \text{ at } \tau_i = 0.62
\]
Besides thermally thick walls, wall temperatures for thermally thin walls are also of interest. Again, for definiteness, it is reasonable to consider a wall material as thermally thin if its thickness $L_{\text{THIN}}$ satisfies

$$L_{\text{THIN}} < 0.6\sqrt{\frac{k}{w} t}$$  \hspace{1cm} (35)

Using material properties from reference 11 and a $t_I$ of 3 minutes, this criterion would be satisfied, for example, with mild steel walls thinner than 0.028 m, glass walls (windows) thinner than 0.006 m, or wood panel walls thinner than 0.004 m.

Using the radiant heat transfer rate of eq. (29) and the $t^m$ fire growth leads to

Radiation to Thermally Thin Rear-Insulated Wall:

$$\phi_w - 1 = \frac{e}{12} X_4 \left( \frac{m+3}{m+1} \right) \left( \frac{0.140}{\zeta_1^{-2/3} - 1} \right)^{3(m+1)/(m+3)}$$  \hspace{1cm} (36)

$$\delta = 0.25 \times 4 (1 - \zeta_1)$$  \hspace{1cm} (37)

where

$$X_4 = \frac{\lambda_r}{(1 - \lambda_c)} \left( \frac{H}{L_{\text{THIN}}} \right) \left( \frac{k_w}{k} \right)$$  \hspace{1cm} (38)

If the wall of the reference 2 enclosure had been sheet metal construction of thickness 0.0015 m instead of gypsum board (and if $\lambda_c$ was not significantly different than the measured value of 0.84), then eq. (37) leads to the estimate
\( \delta = 0.13 \) at \( \zeta_1 = 0.62 \)

and the original small \( \delta \) determination of the significance of the wall effect continues to be relevant.

### 5.2 Wall Heating Due to Convection

In some enclosure fire configurations, radiant heating of enclosure wall surfaces from the fire's combustion zone may play a relatively minor role compared to convection. This would be the case, for example, if direct, line-of-sight, combustion zone illumination of most wall surfaces was blocked by furniture or segmented partitions, or on account of an alcove-like fire location. Also, in a multi-room enclosure configuration, which is an extension of the alcove-like single room configuration, radiant wall heating in rooms other than the room-of-fire-origin could play a minor role well into the "smoke-filling" process.

An estimate for the convection driven \( \phi_w(t) \) increase, to be obtained here, will be based on an eq. (4) heat flux to the wall at every instant of time, \( t^* \), from \( t^* = 0 \) to \( t^* = t \). \( \bar{H}(t^*) \) in eq. (4) will be computed from eq. (5) with \( x = H - Z_2(t^*) \) and with \( Gr_x = Gr(t^*) \) according to eq. (14) or (24). In the estimate, \( \delta \) will be neglected compared to 1 and a laminar wall layer will be assumed through most of the heating history up to the time, \( t \), of interest.
All of the above considerations lead to a convective wall flux, \( \overline{q}_c \), given by

\[
\overline{q}_c(t^*) = \frac{0.48}{3/4} \frac{T_\infty}{H} ke \left( \frac{m+3}{m+1} \right)^{5/4} \frac{\chi^{1/4}_1}{[1-\xi_1^{1/2}]^{3/2}} \frac{15(m+1)}{4(m+3)}
\]  

(39)

where \( \tau \) is evaluated from eq. (19) with \( t = t^* \). Using this last result in the reference 13 solution to the specified flux heat conduction problem for a thermally thick wall leads to the result

Convection to Thermally Thick Wall:

\[
\phi_w^{-1} = \frac{0.17eX_5}{(1-\xi_1)^{1/2}} \left( \frac{\xi_1^{2/3-1}}{0.14} \right)^{11m+9} \frac{11m+9}{4(m+3)} (m+3)^{7/4} \frac{\Gamma(3m+1)/4}{\Gamma(3m+3)/4} \frac{I_1(1-\xi_1;m)}{I_1(1;0;m)}
\]  

(40)

\[
\delta = \frac{0.31X_5(1-\xi_1)^{1/2}}{(\xi_1^{2/3-1})^{1/4}} \frac{(m+3)^{3/4}}{(m+1)^{1/4}} \frac{\Gamma(3m+1)/4}{\Gamma(3m+3)/4} \frac{I_1(1-\xi_1;m)}{I_1(1;0;m)}
\]  

(41)

where

\[
X_5 = \left( \frac{1 - \lambda_c}{1 - \lambda_r} \right)^{1/4} A^{1/2} \left( \frac{k}{\kappa_w} \right)^{1/2} \left( \frac{k}{\kappa} \right)
\]  

(42)

\[
I_1(x;m) = \int_0^1 \left[ \frac{(1-x)^{-2/3-1}}{(1-x)^{-2/3-1}} \right]^{11m+15} \frac{4(m+3)}{11m+9} \left\{ 1 - \left[ \frac{(1-xn)^{-2/3-1}}{(1-x)^{-2/3-1}} \right]^{3} \right\}^{1/2} \frac{dn}{(1-xn)^{5/3} n^{3/2}}
\]  

(43)

\[
I_1(1;0;m) = \frac{(m+3) \Gamma(3/2)\Gamma(3m+1)/4}{3 \Gamma(3m+3)/4}
\]  

(44)
and where \( I_1(x;m)/I_1(0;m) \) has been evaluated and plotted in figure 3 for \( m = 0, 1.0, 2.0, \) and 5.0.

The convective wall flux of eq. (39) has also been used to obtain \( \phi_w - 1 \) and \( \delta \) for thermally thin wall constructions, viz

Convection to Thermally Thin Rear-Insulated Wall:

\[
\phi_w - 1 = \frac{0.26e X_6 x_1^{-1/4}}{(1-x_1)^{1/2}} \left( \frac{\zeta_1}{0.14} \right)^{11m+15} \left( \frac{m+3}{m+1} \right) \frac{I_2(1-\zeta_1;m)}{I_2(0;m)} \tag{45}
\]

\[
\delta = 0.77x_6 x_1^{-1/4} (1-x_1)^{1/2} \left( \frac{\zeta_1}{0.14} \right)^{2/3} \frac{I_2(1-\zeta_1;m)}{4(m+3)} \left( \frac{m+3}{m+1} \right)^{5/3} \frac{I_2(1-\zeta_1;m)}{I_2(0;m)} \tag{46}
\]

where

\[
X_6 = \left( \frac{1 - \lambda_C}{1 - \lambda_r} \right)^{1/2} \frac{A}{I_{THIN} HPr} \left( \frac{\kappa_w}{\kappa} \right) \left( \frac{k_w}{\kappa_w} \right) \tag{47}
\]

\[
I_2(x;m) = \int_0^1 \left( \frac{(1-x\eta)^{-2/3}-1}{(1-x)^{-2/3}-1} \right) \frac{11m+15}{4(m+3)} \frac{d\eta}{(1-x\eta)^{5/3} \eta^{3/2}} \tag{48}
\]

\[
I_2(0;m) = \frac{4}{9} \left( \frac{m+3}{m+1} \right) \tag{49}
\]

and where \( I_2(x;m)/I_2(0;m) \) has been evaluated and plotted in figure 3 for \( m = 0, 1.0, 2.0 \) and 5.0.
The above results can now be applied to the previously discussed multi-room enclosure fires of reference 1. In those enclosure configurations, radiant heating of most wall surfaces (i.e., outside the burn room) was likely to have played a minor role in smoke movement phenomena during the time frames of the experiments. Accordingly, values of δ under conditions of convective wall layer heating have been computed from eqs. (41) and (46). These are presented in table 2 along with the results of all previous example calculations. As can be seen, for the cases considered, the values of δ for the convection driven wall heating examples are never greater than 0.011, and the original, small δ determination of the significance of the wall effect continues to be relevant.

6. QUASISTEADINESS OF THE WALL FLOWS AND A MEASURE OF THE LIKELIHOOD OF THEIR PENETRATION

The validity of two tacitly assumed aspects of the wall flow will be studied in this section. The first has to do with the question of the time required to reach quasisteadiness relative to an assumed steady quiescent upper layer. Related to this, the second has to do with the expectation that the velocities of the wall flow are large enough to actually penetrate the dropping upper layer – lower layer interface.

6.1 The Quasisteady Flow Assumption

The time, $t_s$, for a wall flow to establish itself as a steady flow (subsequent to a step change in wall/ambient temperature, and from the leading edge of the boundary layer to a station, $x$) has been studied extensively in the literature. Reference 7 presents a review of this work, and, in terms of the present nomenclature, recommends the following estimate for $t_s$
\[ t_s = 3(1 - \xi_1^2)h^2/(Gr^{1/2}v_\infty) \]  

(50)

When the time required for the upper layer to grow to a given thickness and temperature is at least as large as the above \( t_s \) estimate, then there is good reason to expect that the quasisteady wall flow assumption is valid. This observation together with the previous results for the \( t^m \) enclosure fire environment finally leads to the following criterion:

With regard to the development of the wall flow in the upper layer during \( t^m \) enclosure fires, the assumption of quasisteadiness is justified when layer interface elevations, \( \xi_1 \), satisfy

\[
\frac{1}{(1-\xi_1)^3/2} \left( \frac{\xi_1 - 2/3}{0.140} \right)^{3/2} \geq 3 \left( \frac{m+1}{1-\delta} \right)^{1/2} \left( \frac{3}{m+3} \right)^{3/2} \frac{HP}{AX_2}
\]  

(51)

The criterion is illustrated in the upper curve of Figure 4, and has been successfully used to test the validity of the quasisteady assumption in all previously described fire scenarios.

6.2 Wall Layer Penetration

During a fire scenario it is reasonable to expect that the wall flow will penetrate the interface when the characteristic velocity of the former, say \( u_{\text{max}} \) of eq. (10), at elevation \( Z_1 \) is much larger than that of the latter, i.e., when

\[ u_{\text{max}}(Z = Z_1) > dZ_1/dt \]  

(52)
To be definite, the following criterion is adopted:

At the upper layer – lower layer interface, penetration of the wall flow into the lower layer is expected if

\[ u_{\text{max}}(Z = Z_1) \geq 5dZ_1/dt \]  \hspace{1cm} (53)

Using the present nomenclature and the previous results for the \( t^m \) enclosure fire environment, finally leads to the following version of eq. (53)

\[ \frac{1}{\xi_1^{5/3}} \left( \frac{\xi_i^{2/3} - 1}{0.140} \right)^{1/2} \geq 1.9 \left( \frac{m+1}{1-\delta} \right)^{1/2} \left( \frac{3}{m+3} \right)^{1/2} \frac{\text{HP}}{\text{AX}_2} \]  \hspace{1cm} (54)

The criterion is illustrated in the lower curve of Figure 4, and has been successfully used to test the validity of the penetration assumption in all previously described fire scenarios.

7. SUMMARY AND CONCLUSIONS

The results of this work point to the significance of buoyancy driven wall flows on the development of hazardous environments within enclosures containing practical growing fires. Such flows were seen to be important, for example, in fire scenarios involving relatively weak fires and/or plan views with relatively long peripheral dimensions. It would clearly be appropriate to formulate and carry out an experimental program for quantitative verification of these results. In any event it appears that the wall flow phenomenon must be accounted for in mathematical enclosure fire models that hope to
estimate dynamic environments generally for the purpose of predicting the response of fire detectors and fire intervention hardware and the time of onset of conditions detrimental to life safety. In order to carry out such an accounting, further analytic and/or experimental research on various aspects of the basic buoyancy driven wall flow may be required, and should be pursued. However, until the results of such studies are available, it may be prudent to add wall flow algorithms to existing models on a more timely basis. Such algorithms would be based on the kinds of calculation, estimates and results which have been developed in the present work. For engineering purposes, such algorithms may prove to be adequate, indeed, optimum, even in the long term.

8. ACKNOWLEDGMENTS

This work was supported by the U.S. Department of Health and Human Services, and the Bureau of Mines and National Park Service of the U.S. Department of Interior.

9. REFERENCES


10. NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>area of enclosure</td>
</tr>
<tr>
<td>$C_i$</td>
<td>concentration of product $i$</td>
</tr>
<tr>
<td>$C_p$</td>
<td>specific heat of air</td>
</tr>
<tr>
<td>$Gr_x$</td>
<td>Grashoff number, eq. (2)</td>
</tr>
<tr>
<td>$Gr$</td>
<td>$Gr_x$ at $x = H - Z_i$</td>
</tr>
<tr>
<td>$g$</td>
<td>acceleration of gravity</td>
</tr>
<tr>
<td>$H$</td>
<td>ceiling-to-fire distance</td>
</tr>
<tr>
<td>$\bar{h}$</td>
<td>average heat transfer coefficient</td>
</tr>
<tr>
<td>$I_1$, $I_2$</td>
<td>integrals, eqs. (43) and (48)</td>
</tr>
<tr>
<td>$K$</td>
<td>kinematic momentum flux, eq. (8)</td>
</tr>
<tr>
<td>$k$</td>
<td>thermal conductivity of air</td>
</tr>
<tr>
<td>$k_w$</td>
<td>thermal conductivity of wall material</td>
</tr>
<tr>
<td>$L_{THICK}$</td>
<td>thermally thick wall thickness</td>
</tr>
<tr>
<td>$L_{THIN}$</td>
<td>thermally thin wall thickness</td>
</tr>
<tr>
<td>$m$</td>
<td>an exponent</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>mass flux</td>
</tr>
<tr>
<td>$n$</td>
<td>an exponent, eqs. (13) and (16)</td>
</tr>
<tr>
<td>$\bar{Nu}_x$</td>
<td>average Nusselt number, eq. (4)</td>
</tr>
<tr>
<td>$P$</td>
<td>perimeter</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$Q$</td>
<td>energy release rate of fire</td>
</tr>
<tr>
<td>$Q_o$</td>
<td>a characteristic value of $Q$</td>
</tr>
<tr>
<td>$Q_o^*$</td>
<td>dimensionless $Q_o$, eq. (12)</td>
</tr>
<tr>
<td>$\bar{q}$</td>
<td>rate of heat transfer per unit area</td>
</tr>
<tr>
<td>$T$</td>
<td>absolute temperature</td>
</tr>
<tr>
<td>$t$, $t^*$</td>
<td>time from ignition</td>
</tr>
</tbody>
</table>
a time of interest
downward velocity in wall flow
dimensionless parameters, eqs. (25), (34), (38), (42), and (47)
distance below ceiling
distance from wall
elevation above fire
a constant, eq. (16)
gamma function
dimensionless temperature difference, eq. (15)
dimensionless parameter, eq. (18)
Z/H
dummy variable
thermal diffusivity of air
thermal diffusivity of wall material
fraction of Q lost by radiation
fraction of Q transferred to enclosure surface
viscosity
kinematic viscosity, \( \mu/\rho \)
dimensionless time, eq. (19)
dimensionless temperature, \( T/T_\infty \)
\( Q/Q_0 \)

Subscripts
C  convective
i  interface
L  lower layer
max  maximum
p
R
u
w
∞

plume
radiant
upper layer
wall flow
ambient
Figure 1. Sketch of the significant features of the wall effect.
Figure 2. Plots of \( \zeta_i \) as a function of \( X_1 \) per eqs. (23) and (24) for different values of \( (\dot{m}_w/\dot{m}_p)X_2 \) and for \( m = 0, 1.0 \) and 2.0 under the assumption \( \delta \ll 1 \).
Figure 3. Plots of $I_1(x;m)/I_1(0;m)$ and $I_2(x;m)/I_2(0;m)$ per eqs. (43)–(44) and (48)–(49), respectively, for $m = 0$, $1.0$, $2.0$ and $5.0$.\[\text{\[appropriate_mathematical_expression\]}\]
Figure 4. Plots of eq. (51) and (54) criteria for wall layer quasisteadiness and penetration, as functions of interface elevation and for t^m enclosure fires.
<table>
<thead>
<tr>
<th>Material</th>
<th>Conductivity $k\left(\frac{W}{m^0 K}\right)$</th>
<th>Diffusivity $k\left(\frac{m^2}{s}\right)$</th>
</tr>
</thead>
<tbody>
<tr>
<td>air$^{13}$</td>
<td>0.024</td>
<td>1.9($10^{-5}$)</td>
</tr>
<tr>
<td>gypsum board$^{14}$</td>
<td>0.17</td>
<td>1.6($10^{-7}$)</td>
</tr>
<tr>
<td>concrete (1:2:4)$^{11}$</td>
<td>0.92</td>
<td>4.2($10^{-7}$)</td>
</tr>
<tr>
<td>steel (0.1% C)$^{11}$</td>
<td>46.0</td>
<td>1.2($10^{-5}$)</td>
</tr>
</tbody>
</table>
Table 2. Results of Calculations on the Significance of the Wall Effect in Full-Scale Experiments of References 1 and 2

<table>
<thead>
<tr>
<th></th>
<th>$\lambda_c$</th>
<th>$\lambda_r$</th>
<th>$\epsilon$</th>
<th>$X_1$</th>
<th>$X_2$</th>
<th>$\frac{\dot{m}_w}{\dot{m}_p}$</th>
<th>Wall Material</th>
<th>Wall Heating Condition</th>
<th>$\delta$</th>
<th>$\frac{\dot{m}_w}{\dot{m}_p}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mulholland et al.</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>16.2 kW, m = 0, acetylene</td>
<td>0.84</td>
<td>0.50</td>
<td>0.0036</td>
<td>1.4(10^9)</td>
<td>4.0</td>
<td>0.62</td>
<td>gyp. bd.</td>
<td>rad-thick:$X_3 = 0.026$</td>
<td>0.29</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>concrete</td>
<td>rad-thick:$X_3 = 0.0078$</td>
<td>0.086</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.5 mm steel</td>
<td>rad-thin:$X_4 = 1.4$</td>
<td>0.13</td>
<td></td>
</tr>
<tr>
<td>Cooper et al.</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>25 kW, m = 0, 89.6 m²</td>
<td>0.87</td>
<td>0.19</td>
<td>0.0032</td>
<td>1.3(10^9)</td>
<td>15.</td>
<td>0.85</td>
<td>gyp. bd.</td>
<td>conv-thick:$X_5 = 0.043$</td>
<td>0.0093</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>concrete</td>
<td>conv-thick:$X_5 = 0.013$</td>
<td>0.0028</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.5 mm steel</td>
<td>conv-thin:$X_6 = 5.1$</td>
<td>0.0082</td>
<td></td>
</tr>
<tr>
<td>225 kW, m = 0, 40.6 m²</td>
<td>0.75</td>
<td>0.24</td>
<td>0.027</td>
<td>1.1(10^10)</td>
<td>12.</td>
<td>0.76</td>
<td>gyp. bd.</td>
<td>conv-thick:$X_5 = 0.034$</td>
<td>0.0081</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>concrete</td>
<td>conv-thick:$X_5 = 0.010$</td>
<td>0.0024</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.5 mm steel</td>
<td>conv-thin:$X_6 = 3.3$</td>
<td>0.0043</td>
<td></td>
</tr>
<tr>
<td>ramp, m = 1, 40.6 m²</td>
<td>0.81</td>
<td>0.24</td>
<td>0.0064</td>
<td>2.6(10^9)</td>
<td>39.</td>
<td>0.93</td>
<td>gyp. bd.</td>
<td>conv-thick:$X_5 = 0.032$</td>
<td>0.0059</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>concrete</td>
<td>conv-thick:$X_5 = 0.010$</td>
<td>0.0018</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.5 mm steel</td>
<td>conv-thin:$X_6 = 2.9$</td>
<td>0.0024</td>
<td></td>
</tr>
<tr>
<td>ramp, m = 1, 89.6 m²</td>
<td>0.71</td>
<td>0.24</td>
<td>0.0065</td>
<td>2.7(10^9)</td>
<td>27.</td>
<td>0.90</td>
<td>gyp. bd.</td>
<td>conv-thick:$X_5 = 0.053$</td>
<td>0.0110</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>concrete</td>
<td>conv-thick:$X_5 = 0.016$</td>
<td>0.0032</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.5 mm steel</td>
<td>conv-thin:$X_6 = 7.9$</td>
<td>0.0081</td>
<td></td>
</tr>
</tbody>
</table>
On the Significance of A Wall Effect in Enclosures with Growing Fires

Leonard Y. Cooper

This paper studies the significance of a wall effect that has been observed during the growth stage of enclosure fire experiments. Relative to the two-layer phenomenon which tends to develop during such experiments, the effect has to do with the near-wall downward injection of hot upper layer gases into the relatively cool uncontaminated lower layer. It is conjectured that these observed wall flows are buoyancy driven, and that they develop because of the relatively cool temperatures of the upper wall whose surfaces are in contact with the hot upper layer gases. For a growing fire (growth proportional to $t^m$; $t$ being time and $m \geq 0$) in an enclosed compartment, the conjectured mechanism for the wall flow leads to a time-dependent solution for the ratio of wall layer mass ejection rate from the upper layer, $\dot{m}_w$, to the fire plume mass injection rate to the upper layer, $\dot{m}$, as $\dot{m}_w / \dot{m}$. The solution indicates that in practical fire scenarios $\dot{m}_w / \dot{m}$ can be of the order of "several tenths" even prior to the time that the upper layer interface has dropped to an elevation midway between the ceiling and fire. In other words, the results of the analysis indicate the importance of taking the wall effect into account in two-layer zonal analyses of enclosure fire phenomena.

compartment fires, enclosure fires, fire growth, growing fires, mathematical modeling, smoke movement, two-layer phenomenon, wall flows.