CALCULATING COMBINED BUOYANCY - AND PRESSURE-DRIVEN FLOW THROUGH A SHALLOW, HORIZONTAL, CIRCULAR VENT; APPLICATION TO PROBLEM OF STEADY BURNING IN A CEILING-VENTED ENCLOSURE

by

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ABSTRACT

A model was developed previously for calculating combined buoyancy- and pressure-driven (i.e., forced) flow through a shallow, circular, horizontal vent where the vent-connected spaces are filled with fluids of different density in an unstable configuration (density of the top fluid is larger than that of the bottom). In this paper the model equations are summarized and then applied to the problem of steady burning in a ceiling-vented enclosure where normal atmospheric conditions characterize the upper space environment. Such fire scenarios are seen to involve a zero-to-relatively-moderate cross-vent pressure difference and bi-directional exchange flow between the enclosure and the upper space. A general solution to the problem is obtained. This relates the rate of energy release of the fire to the area of the vent and to the temperature and oxygen concentration of the upper portion of the enclosure environment. The solution is seen to be consistent with previously-published data involving ceiling-vented fire scenarios.

INTRODUCTION

Consider the flow through a horizontal vent (i.e., a vent in a horizontal partition) where the fluids which fill the vent-connected spaces near the elevation of the vent are of different density and in an unstable configuration, a relatively dense fluid in the upper space, ρ_{TOP}, overlaying a relatively less dense fluid in the lower space, ρ_{BOT},

$$\Delta\rho = \rho_{TOP} - \rho_{BOT} > 0 \tag{1}$$

The focus of this work is on problems where the fluids in the upper and lower spaces can both be accurately described as the same perfect gas in the sense that use of identical gas-property models in the two spaces would everywhere lead to accurate estimates of thermodynamic and transport properties. Relative to the technology of fire safety, a prototype example of such problems involves exchange flows through a vent where uncontaminated ambient-temperature air in an upper space overlays elevated-temperature combustion-product-contaminated or "smokey" air in a lower space.

Assume that in each space, away from the vent, the environment is relatively quiescent with pressure well-approximated by the hydrostatic pressure field. As in Figure 1, designate the spaces connected by the vent as top and bottom. Subscripts TOP and BOT, respectively, refer to conditions in these spaces near the vent elevation, but removed laterally from the vent. Similarly, subscripts HIGH and LOW refer to the conditions on the high- and low-pressure sides of the vent, respectively.

$\dot{V}_{TOP}$ and $\dot{V}_{BOT}$ are the volume flow rates through the vent from top to the bottom side of the vent and from the bottom to the top side of the vent, respectively. Similarly, $\dot{V}_{HIGH}$ and $\dot{V}_{LOW}$ are the volume flow rates from the high- to the low-pressure side and from the low- to the high-pressure side of the vent. Flow through the vent is determined by: the design of the vent, its shape and its depth, L; ρ_{TOP} and ρ_{BOT}; and the pressures, p_{TOP} and p_{BOT}.

The cross-vent pressure difference is

$$\Delta p = p_{HIGH} - p_{LOW} \geq 0$$
$$p_{HIGH} = \max(p_{TOP}, p_{BOT}); \tag{2}$$
$$p_{LOW} = \min(p_{TOP}, p_{BOT})$$

Δp and the characteristic elevation interval of the flow problem, H, are assumed to be so small that

$$\Delta\rho g|H|/\bar{p} << 1;$$
$$\Delta p/\bar{p} << 1;\ \bar{p} = (p_{HIGH} + p_{LOW})/2 \tag{3}$$

where g is the acceleration of gravity. Then, for the

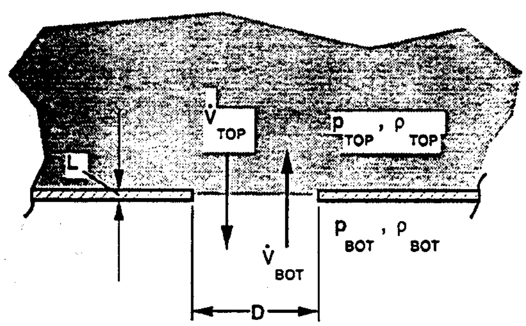

Figure 1. The Horizontal Vent Configuration.

purpose of establishing the interdependence throughout the flow region of density, ρ, and temperature, T, the equation of state for the gas can be well approximated by

$$\rho T = \text{constant} = \rho_{TOP}T_{TOP} = \rho_{BOT}T_{BOT} = \bar{p}/R \quad (4)$$

where R is the gas constant and temperatures T_{TOP} and T_{BOT} correspond to specified ρ_{TOP} and ρ_{BOT} though Eq. (4).

For any unstable arrangement of densities across a vent there will always be a value $\Delta p = \Delta p_{FLOOD}$, denoted as the critical or flooding value of Δp, which separates a uni-directional or "flooding" flow regime ($\Delta p \geq \Delta p_{FLOOD}$) where $\dot{V}_{LOW} = 0$, from a "mixed" flow regime ($0 \leq \Delta p < \Delta p_{FLOOD}$) where $\dot{V}_{LOW} > 0$. Associated with any particular Δp_{FLOOD} value is a corresponding volumetric flooding flow rate, $\dot{V}_{FLOOD} = \dot{V}_{HIGH}(\Delta p = \Delta p_{FLOOD})$. When $\Delta p = 0$, $\dot{V}_{HIGH} = \dot{V}_{LOW}$ and the HIGH/LOW designations are arbitrary.

$\dot{V}_{NET}$ is the net volume flow rate from the high to the low-pressure side of the vent.

$$\dot{V}_{NET} = \dot{V}_{HIGH} - \dot{V}_{LOW} \geq 0 \quad (5)$$

This is the forced or pressure-driven part of the vent flow. At the two extremes of the mixed flow regime, $\dot{V}_{NET} = \dot{V}_{FLOOD}$ at $\Delta p = \Delta p_{FLOOD}$ and $\dot{V}_{NET} = 0$ at $\Delta p = 0$. Similarly, $\dot{V}_{LOW}$ is the buoyancy-driven part of the flow, a non-zero value for which corresponds to non-zero "exchange flow." $\dot{V}_{LOW} = 0$ when $\Delta p = \Delta p_{FLOOD}$ and $\dot{V}_{LOW}$ reaches its maximum value, $\dot{V}_{EX,MAX}$, as Δp and the forced part of the flow go to zero.

THE ALGORITHM VENTCL2 FOR DETERMINING $\dot{V}_{HIGH}$ AND $\dot{V}_{LOW}$ THROUGH A SHALLOW CIRCULAR VENT FOR ARBITRARY VALUES OF p_{TOP}, p_{BOT}, AND $\rho_{TOP} > \rho_{BOT}$

For arbitrary specified values of p_{TOP}, p_{BOT}, $\rho_{TOP} > \rho_{BOT}$, a mathematical model and concise algorithm, **VENTCL2**, to calculate $\dot{V}_{HIGH}$ and $\dot{V}_{LOW}$ was developed and presented in Reference 1. Use of the model and algorithm is limited to the prediction of turbulent, large-Grashof number flows[1] though a shallow ($L/D << 1$), circular vents. **VENTCL2** is an advanced version of the algorithm **VENTCL**[2,3].

APPLICATIONS OF VENTCL2: STEADY BURNING IN A CEILING-VENTED ROOM

The Problem

Consider a room with a fire, fully-enclosed except for a shallow circular ceiling vent. Refer to Figure 2. The air above the vent has ambient density, absolute temperature, and oxygen (O_2) mass concentration, T_{AMB}, ρ_{AMB}, and ψ_{AMB}, respectively. Assume steady conditions where the room environment immediately below the vent has density, temperature, and O_2 mass concentration ρ, $T > T_{AMB}$, and $\psi < \psi_{AMB}$, respectively. ψ_{LOW}, the O_2 mass concentration in the lower part of the room at the elevation of the fire, must exceed the minimum, extinction value, ψ_{EXT}, associated with the particular fuel. For example, for the combustion of CH_4 diffusion flames from round burners with diameters D in the range $0.50\ m \leq D \leq 0.089\ m$, ψ_{EXT} was measured by Morehart, Zukoski, and Kubota[5,6] as ranging from 0.140 ($D = 0.50m$) to 0.161 ($D = 0.089\ m$). Note that under the conjectured steady state condition, the O_2 that supplies the lower part of the room and maintains it at a $\psi_{LOW} > \psi$ comes from the cool and relatively O_2-rich ambient air that enters the ceiling vent and drops toward the floor in a negatively buoyant plume.

In this section the **VENTCL2** algorithm will be used to estimate the exchange flow through the vent and the burning rate that can be supported by the net rate of oxygen inflow.

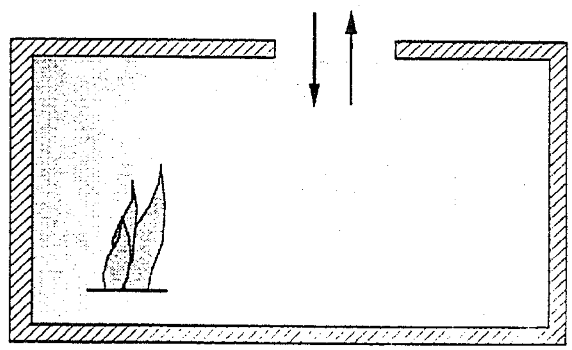

Figure 2. Configuration of a ceiling-vented room with a fire.

The Relationship Between Δp and T

Assume: the mass-flow-rate of fuel introduced by the fire is negligible compared to the mass-flow-rate of the exchange-flow; the environment inside and outside the room can be modeled as a perfect-gas approximation to air; and there is no mixing in the vent, i.e., all inflow is at the ambient condition and all outflow is at the upper room environment. Using the approximation of Eq. (3), it follows from Eq. (4) that

$$T/T_{AMB} = \rho_{AMB}/\rho = \rho_{TOP}/\rho_{BOT} > 1 \quad (6)$$

From Eqs. (1) and (6) it is evident that the present problem involves an unstable configuration, and that the **VENTCL2** flow algorithm is applicable.

Conservation of mass across the vent requires

$$\rho \dot{V}_{BOT} = \rho_{AMB} \dot{V}_{TOP} \quad (7)$$

Using Eq. (6), it follows from Eq. (7) that the high and low pressure sides of the vent are at the bottom and top, respectively, and that the problem the mixed-flow regime.

$$\dot{V}_{HIGH} = \dot{V}_{BOT};\ \dot{V}_{LOW} = \dot{V}_{TOP};\ \rho_{HIGH} = \rho;\ \rho_{LOW} = \rho_{AMB} \quad (8)$$

Following **VENTCL2**, define

$$\varepsilon' = -\Delta\rho/\bar{\rho} = -2\Delta\rho/(\rho + \rho_{AMB}) = -2(T^* - 1)/(T^* + 1) < 0;\ T^* \equiv T/T_{AMB} = (2 - \varepsilon')/(2 + \varepsilon')$$
$$\delta p^* = \Delta p/\Delta p_{FLOOD};\ \Delta p_{FLOOD} = 0.2427(4g\Delta\rho D)(1 + \varepsilon'/2)\exp(1.1072\varepsilon') \quad (9)$$

Then **VENTCL2** is found to require the following dependence of δp^* on ε'

$$\phi(\delta p^*) = \lambda(\varepsilon') \quad (10)$$

where

$$\lambda(\varepsilon') = -2(0.282)\varepsilon'\exp(-0.5536\varepsilon')/(2 + \varepsilon');\ M = 9.400;$$
$$\phi(\delta p^*) = \{M - [1 + (M^2 - 1)(1 - \delta p^*)]^{1/2}\}/\{(M - 1)[0.6465(1 - \delta p^*)^2 - 1.6465(1 - \delta p^*)]^2\} \quad (11)$$

Using the numerical root-finder **RTSAFE**[6], the solution of Eq. (10) for δp^* as a function of ε' or T/T_{AMB} was found for a wide range of $\varepsilon' < 0$ ($T > T_{AMB}$). This is plotted in Figure 3.

The Energy Release Rate of the Fire as a Function of T and Its Maximum Possible Value

The energy-release rate, $\dot{Q}$, of the fire is related to the net rate of O_2 inflow which is consumed by the combustion. Eqs. (7) and (8) lead to

$$\text{net rate of } O_2 \text{ consumption} = \psi_{AMB}\rho_{AMB}\dot{V}_{LOW} - \psi\rho_{HIGH}\dot{V}_{HIGH}$$
$$= 0.055D^{5/2}g^{1/2}|\varepsilon'|^{1/2}\psi_{AMB}\rho_{AMB}(1 - \psi/\psi_{AMB})\dot{V}_{LOW}/\dot{V}_{EX,MAX} \quad (12)$$

where, $\dot{V}_{LOW}/\dot{V}_{EX,MAX}$, a function of δp^*, is obtained from Eqs. (12).

From Reference 7

$$C_{O2} \equiv \dot{Q}/(\text{net rate of } O_2 \text{ consumed}) = 13.2(10^3)\text{kW}/(\text{kg}_{O2}/\text{s}) \quad (13)$$

Using Eq. (12) in Eq. (13) and defining a dimensionless $\dot{Q}$

$$\dot{Q}^* \equiv \dot{Q}/[(1 - \psi/\psi_{AMB})\rho_{AMB}\psi_{AMB}C_{O2}A_V^{5/4}g^{1/2}] \quad (14)$$

leads to

$$\dot{Q}^* = 0.074|\varepsilon'|^{1/2}\dot{V}_{LOW}^{(1)}/\dot{V}_{EX,MAX} \quad (15)$$

where **VENTCL2**[1] includes the estimate

$$\dot{V}_{LOW}/\dot{V}_{EX,MAX} = [0.6465(1 - \delta p^*)^2 - 1.6465(1 - \delta p^*)]^2;$$
$$\dot{V}_{EX,MAX} = 0.055(4/\pi)A_V(gD|\varepsilon|)^{1/2} \quad (16)$$

The previously determined δp^* *vs* ε solutions were used in Eqs. (9), (15), and (16) to obtain Q^* *vs* T/T_{AMB} and this is plotted in Figure 3. From this it is seen that Q^* is predicted to rise rapidly from 0, at $T/T_{AMB} = 1$, to a maximum value, $Q^*_{MAX} = 0.037$, at $T/T_{AMB} = 1.65$, and to monotonically decrease with further increases of T/T_{AMB}. Associated with Q^*_{MAX}, let Q_{MAX} be the maximum possible Q for a given ψ. Taking $T_{AMB} = 300$ K, $\rho_{AMB} = 1.2\ kg/m^3$, $\psi_{AMB} = (0.23\ kg\ O_2)/kg$, and $g = 9.8\ m/s^2$, Eqs. (13) and (14) lead to

$$Q_{MAX} = 0.41(10^3)\{1 - \psi/[0.23(kg\ O_2)/kg]\}(A_V/m^2)^{5/4}\ kW \tag{17}$$

The scenario, leading to the largest value of Q_{MAX}, is one where ψ is negligible. This would likely be associated with $\psi_{LOW} \approx \psi_{EXT}$. Thus, from Eq. (17)

$$Q_{MAX} < 0.41(10^3)(A_V/m^2)^{5/4}\ kW = 0.41(10^3)\ kW,\ 1.3\ kW,\ \text{and}\ 0.23\ kW$$
$$\text{for}\ A_V = 1.0\ m^2,\ 1.0(10^{-2})\ m^2,\ \text{and}\ 25.0(10^{-4})\ m^2,\ \text{respectively} \tag{18}$$

Note that replacing 0.41 by 0.6 in the inequality of Eq. (18) leads to a result equivalent to Eq. (13) of Reference 10.

The results of Figure 3 are now related to data acquired in "full-scale" experiments reported in References 8 and 9. In this it is assumed that the present circular-vent results can be used to provide estimates for the square- and rectangular-vented enclosures used in the experiments.

Experimental Validation of the Figure 3 Solution

Fire in Ceiling-Vented Ship Quarters. Reference 8 reports on a fire in a mock-up fully-furnished three-person ship accommodation quarter (3.84 m x 2.82 m x 2.38 m high), fully enclosed except for a single square vent, $A_V = 1\ m^2$, in a corner of the ceiling, away from the furnishings. The fire involved an initial interval of intense burning which rapidly decayed to smoldering (10 minutes); an interval of smoldering (20 minutes), and a final interval of intense burning (30 minutes). The final interval involved a 19-20 minute sub-interval in which the heat release rate was relatively constant at $Q = (0.25 \pm 0.05)10^3$ kW. It is reasonable to expect that the latter sub-interval was a time of steady state during which the present example analysis of ventilation conditions is relevant. Indeed, the measured burn rate does satisfy the criterion of Eq. (18), i.e., $Q = (0.25 \pm 0.05)10^3\ kW < Q_{MAX} = 0.41(10^3)$ kW. Also, $\psi = 0.09\ (kg\ O_2)/kg$ was estimated from Eq. (17). There is no reported measured value to validate the latter result. However, the result is plausible since, as required, it is clearly less than the likely value of $\psi_{LOW} \approx \psi_{EXT}$.

Note that because of the original assumption of no mixing in the vent, the $Q_{MAX} = 0.41(10^3)$ kW and $\psi = 0.09\ (kg\ O_2)/kg$ estimates must be regarded as upper and lower bounds, respectively, to the actual expected values. Thus, the actual rate of O_2 inflow would be less than $\psi_{AMB}\rho_{AMB}V_{TOP}$ and the rate of O_2 outflow would be greater than $\psi\rho V_{BOT}$.

Wood Fires in a Ceiling-Vented 27 m^3 Cubic Enclosure. Reference 9 reports on five experiments involving wood fires located at the center of the floor of a cubic room (6 m x 6 m x 6 m), fully enclosed except for a single, centrally-located, ceiling vent. Three vents were used: $A_V = 4\ m^2$ (2 m x 2 m), $2\ m^2$ (1 m x 2 m), and $1\ m^2$ (1 m x 1 m). The burn times were 30 min. Measured and reported variables included: dM/dt, where M is the mass of the fuel; $T_{UP,AVE}$, the average of the upper-enclosure temperatures; and $c_{LOW,O2}$, $c_{LOW,CO2}$, the molal fractions of O_2 and CO_2, respectively, in the lower part of the enclosure, 1 m from the floor and 1 m from the combustion zone. The data were studied to identify intervals that could be reasonably construed to represent quasi-steady-state conditions for which the present example calculation would be relevant. The selected criterion for this was that all measured variables reported in Reference 9 were relatively constant over an interval of at least 5 min.

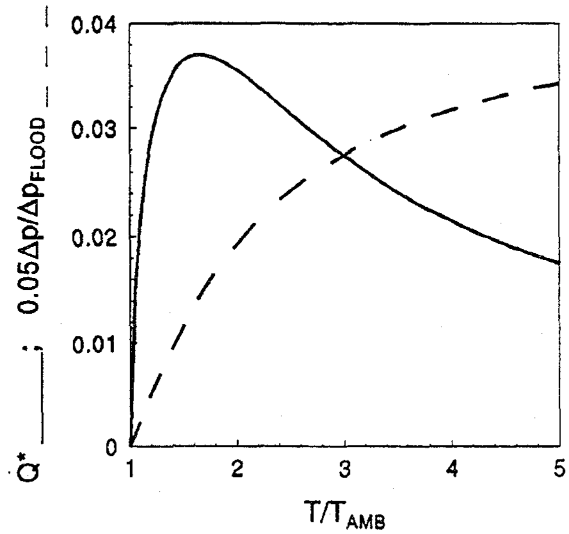

Figure 3. Plots of $\Delta p/\Delta p_{FLOOD}$ and $Q^* = Q/[(1 - \psi/\psi_{AMB})\rho_{AMB}\psi_{AMB}C_{O2}A_V^{5/2}g^{1/2}]$ as functions of T/T_{AMB} for the configuration of Figure 2.

The "best" steady state interval was found and analyzed for experiment 2, 3, and 4. No steady state intervals were identified in experiments 1 and 5.

For the experiments, the heat of combustion of the wood fuel was taken to be[11] 19.5 kJ/g and it was assumed that the smoke yield was negligible. Then, for the intervals of steady state burning, the Figure 3 results and Eqs. (13) and (14) were used to estimate ψ from Q_{MEAS} (the measured values of $\dot{Q}$ as deduced from the measured values of dM/dt) and from T (estimated to be identical to $T_{UP,AVE}$).

The results of the analyses are summarized in Table 1. In the table, ψ_{LOW} was estimated from $c_{LOW,O2}$ according to $\psi_{LOW} \approx 0.23(c_{LOW,O2}/0.21)$.

Note that the low ψ_{LOW} values of experiments 2 and 3, approximately 0.15, indicate that the fire in both cases was close to extinction. The measured values of $c_{LOW,O2}$ in these two cases were 0.137 $\pm$ 0.004 and 0.141 $\pm$ 0.002 for experiments 2 and 3, respectively; these are the lowest O_2 concentrations measured throughout the entire Reference-9 test series.

As in the analysis of Reference 9, there are no reported measured values of ψ to directly confirm the calculated results of Table 1. However, once again the calculated results are plausible, since, as required, they are always less than ψ_{LOW}. As with the previous example, the results are also consistent with the original assumption of no mixing in the vent in that it is reasonable to anticipate that actual values of ψ, expected to be greater than the predicted values of Table 1, would, as required, also be less than the corresponding values of ψ_{LOW}. Thus, in experiment 2, for example, it is expected that the experimental value of ψ was somewhat greater than 0.08, while still being less than $\psi_{LOW} \approx 0.15$.

Exp't. no., interval, initial mass	A_V [m^2]	$T_{UP,AVE}$ [K]	Q_{MEAS} [kW]	ψ [(kg O_2)/kg]	ψ_{LOW} [(kg O_2)/kg]
2, 15-20 min, 100 kg	2	440 +/- 6	620 +/- 60	0.08	0.15 +/- 0.004
3, 15-20 min, 100 kg	1	386 +/- 1	250 +/- 10	0.07	0.15 +/- 0.002
4, 5-10 min, 25 kg	4	373 +/- 5	550 +/- 100	0.16	0.21 +/- 0.003

Table 1. Data on Ceiling-Vented Wood Fire Scenarios[9] and Application of Figure 3 and Eqs. (13) and (14).

SUMMARY AND CONCLUSIONS

A previously developed model[1] for calculating combined buoyancy- and pressure-driven flow through a shallow, circular, horizontal vent was used to solve the problem of steady burning in a ceiling-vented enclosure. The phenomenon involves an exchange flow at the vent driven by the unstable configuration of relatively cool and dense gas above the vent (the outside air) over elevated-temperature low-density gas below the vent (the heated, product-of-combustion- contaminated air in the enclosure). A general solution to the problem, presented in Figure 3, provides the functional dependence between the energy release rate of the burning fuel, the mass concentration of oxygen in the enclosure, the diameter of the vent, and the ratio of inside-to-outside temperature. From this it is seen that, in general, for specified values of oxygen mass concentration and vent diameter, the energy release rate from combustion in the enclosure is
a maximum for T/T_{AMB} = 1.65. Also presented in Figure 3 is the solution for cross-vent pressure difference as a function of inside-to-outside temperature ratio. The Figure-3 solution was found to be consistent with previously-published data involving full-scale ceiling-vented fire scenarios.

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NOMENCLATURE

A_V	vent area
C_{O2}	Eq. (13)
$c_{LOW,O2}$, $c_{LOW,CO2}$	molal fractions of O_2, CO_2 in lower part of enclosure

D	diameter of vent or burner
g	acceleration of gravity
H	characteristic elevation interval of the flow problem
L	depth of vent
M	9.400 in Eqs. (11), or mass of fuel
p; p_{HIGH}, p_{LOW}	pressure; far-field p on high-, low-pressure side of vent, near the vent elevation;
$\bar{p}$	$(p_{HIGH} + p_{LOW})/2$
Q; Q_{MEAS}; Q_{MAX}	burning rate; Q measured in Ref. 9; maximum of Q
Q^*	dimensionless Q, Eq. (14)
R	gas constant
T; T_{TOP}, T_{BOT}; T_{AMB}; T^*	absolute temperature; far field T in top, bottom space, near the vent elevation; T of ambient; T/T_{AMB}
$T_{UP,AVE}$	average of upper-enclosure T measured in Ref. 9
$\bar{T}$	$(T_{TOP} + T_{BOT})/2$

$\dot{V}_{EX,MAX}$	maximum exchange flow rate, at $\Delta p = 0$, i.e., maximum possible $\dot{V}_{LOW}$
$\dot{V}_{FLOOD}$	$\dot{V}_{HIGH}$ at onset of flooding
$\dot{V}_{HIGH}$, $\dot{V}_{LOW}$	volumetric flow rates from high- to low-pressure and from low- to high-pressure side of vent
$\dot{V}_{TOP}$, $\dot{V}_{BOT}$	volumetric flow rates from top- to bottom-side and from bottom- to top-side of vent
$\dot{V}_{NET}$	$\dot{V}_{HIGH} - \dot{V}_{LOW}$
Δp, Δp_{FLOOD};	$p_{HIGH} - p_{LOW}$; Δp at onset of flooding
$\Delta\rho$	$\rho_{TOP} - \rho_{BOT}$
δp^*	$\Delta p/\Delta p_{FLOOD}$
ε'	dimensionless $\Delta\rho$, Eq. (9)
λ	Eq. (11)
ρ; ρ_{TOP}, ρ_{BOT}	density; far-field ρ in top, bottom space, near the vent elevation
ϕ	Eqs. (10) and (11)
ψ, ψ_{AMB}, ψ_{LOW}	mass concentration of O_2 in enclosure, of ambient, in lower part of enclosure

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