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On the Temperatures in Forced-Ventilation Fires

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ABSTRACT

Full-scale burning tests were performed on wood, polymethylmethacrylate (PMMA), and methanol fires in a compartment with forced ventilation. The gas temperatures at seven positions were measured together with the transient mass loss rate of the fuel. Average temperatures of the hot gases were then compared with the values predicted by a simple model proposed by Deal and Beyler (1990). The heat loss coefficient of the compartment was found to be an important parameter, and an empirical parameter is fitted from these tests with small fires.

INTRODUCTION

The hot gas temperature in a building fire is an important parameter to be considered by fire engineers. Certainly, it can be predicted by empirical formula or fire models that are available in the literature. However, most of the models are derived from fires in buildings with natural ventilation, but most of the commercial buildings in Hong Kong are designed with forced ventilation. The ventilation rate depends on the use of the buildings and their floor area. Usually the smaller the floor area, the larger the ventilation rate. Large open-area offices might have a ventilation rate of about 10 air changes per hour; smaller compartments, such as a toilet, would have a higher ventilation rate, up to 50 air changes per hour.

A fire that occurs in a compartment with forced ventilation will be very different from one in a building with natural ventilation. For example, the stratified thermal layer induced by the fire would be unstable. Estimation of the fire environment inside such compartments using the theories based on natural-ventilation fires is not very good. The peak gas temperature has been found to be underestimated (e.g., Mitler 1984) if the effect of forced ventilation is neglected. Studying the effect of ventilation rate on the changes of enclosure fire properties is, therefore, important in providing data for fire services design.

The fire environment inside a compartment with forced ventilation can be simulated using a modified computer fire model, but modifying this kind of model would require some knowledge of fire science. A simple model for estimating the average hot gas temperature in the compartment is more desirable. This would enable engineers to estimate the fire temperature quickly.

Although not much work has been reported on forced-ventilation fires, fires in compartments with extraction ventilation in a test chamber were studied experimentally by Alvares et al. (1984), Foote et al. (1986), and Backovsky et al. (1988). The experimental data were later complied and modeled with a simple formula by Deal and Beyler (1990) and Beyler (1991). The objective of this paper is to evaluate the validity of this simple model through an experimental study in a compartment under forced-ventilation conditions. Materials investigated included wood, polymethylmethacrylate (PMMA), and methanol. A fire chamber (Chow and Chan 1993) at a university with extraction ventilation and of size similar to a typical office in a commercial building is used for the study. Interest lies on the smaller fires since most of the fire service systems are designed to operate at the preflashover stage. Fire of thermal power less than 40 kW (136,500 Btu/h) and hot gas temperatures lower than 50°C (122°F) are studied.

EXPERIMENTS

The fire chamber used for the burning tests is 4 m (13.1 ft) long, 3 m (9.8 ft) wide, and 2.8 m (9.2 ft) high. It is shown in

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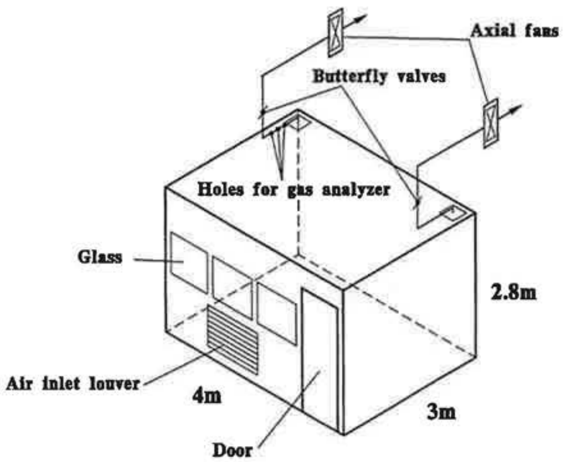

Figure 1 *Sketch of the fire chamber.*

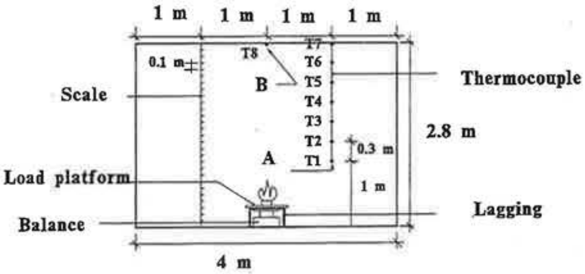

Figure 2 *Equipment setup.*

Figure 1. The air inlet louvre is 0.52 m (1.7 ft) wide and 0.41 m (1.3 ft) high, as shown in Figure 1. There are six slots in the louvre, each with an area of 0.017 m^2 (0.18 ft^2) and height of 0.034 m (0.11 ft). The ventilation factor (defined as $A\sqrt{H}$, where A is the area and H is the height of the opening) of each slot is 0.003 $m^{5/2}$ (0.06 $ft^{5/2}$), i.e., 0.017 m^2 × 0.034 $m^{1/2}$, and the total ventilation factor of the louvre is 0.18 $m^{5/2}$ (3.5 $ft^{5/2}$). Hot gases are extracted from the fire chamber by two axial fans through an air duct of 0.09 m^2 (0.97 ft^2) cross-sectional area. Butterfly valves are installed in the upstream of axial fans to control the gas extraction rate. Gas velocities are measured by vane-type aneometers inside the air duct in order to calculate the gas extraction rate. The experimental setup inside the fire chamber is shown in Figure 2. An electronic balance with a dura-steel load platform was placed at the center to measure the mass of the fuel. A thermocouple rake with seven thermocouples, labelled T1 to T7 and spaced at 300 mm (0.98 ft) intervals, was installed at position A at the right of the platform. Another thermocouple, T8, was located at position B above the load platform to measure the temperature of the hot gases. All thermocouples were connected to a data acquisition system outside the fire chamber.

TABLE 1
Summary of the 21 Fire Tests

Test Number	Fuel	Gas Extraction Rate: L/s (cfm)	Gas Extraction Rate: Air changes per hour
1	Wood	0 (0)	0
2	Wood	93 (197)	10
3	Wood	185 (392)	20
4	Wood	278 (589)	30
5	Wood	371 (786)	40
6	Wood	464 (983)	50
7	Wood	556 (1178)	60
8	PMMA	0 (0)	0
9	PMMA	93 (197)	10
10	PMMA	185 (392)	20
11	PMMA	278 (589)	30
12	PMMA	371 (786)	40
13	PMMA	464 (983)	50
14	PMMA	556 (1178)	60
15	Methanol	0 (0)	0
16	Methanol	93 (197)	10
17	Methanol	185 (392)	20
18	Methanol	278 (589)	30
19	Methanol	371 (786)	40
20	Methanol	464 (983)	50
21	Methanol	556 (1178)	60

The burning behavior of wood, polymethylmethacrylate (PMMA), and methanol was investigated at different fuel sizes and ventilation rates. Twenty-one tests were performed in the fire chamber with gas extraction rates up to 60 air changes per hour or 556 L/s (1180 cfm) as noted in Table 1. The fuel sizes were 310 mm × 310 mm × 100 mm (1.02 ft × 1.02 ft × 0.33 ft) for wood and 140 mm × 140 mm × 120 mm (0.46 ft × 0.46 ft × 0.39 ft) for PMMA. Both of them were arranged in cribs with four layers of sticks and soaked with 30 g (0.066 lb) of methanol for easier ignition (see Mizuno and Kawagoe 1984). Methanol was put in a pan with 150 mm (0.49 ft) diameter and 20 mm (0.066 ft) depth. The fuel sizes were fixed in the fire tests performed. Once the fuel was ignited, temperature and mass of fuel were recorded at intervals of one minute. The mass loss rate, $\dot{M}_f$, was calculated from the slope of the transient mass curve, with the heat-release rate Q_t estimated by (SFPE 1988/1995)

$$Q_t = \dot{M}_f \Delta H_c . \quad (1)$$

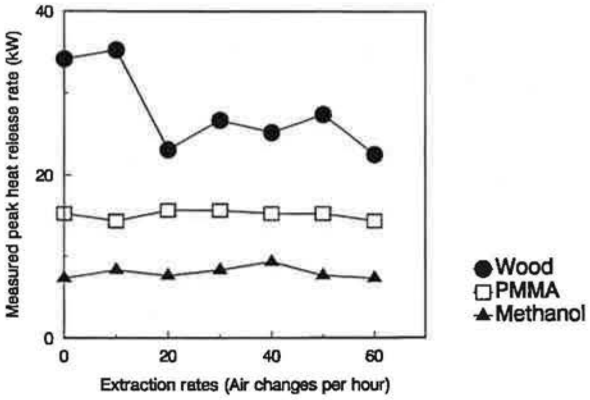

Figure 3 *Peak heat release rates vs. ventilation rates.*

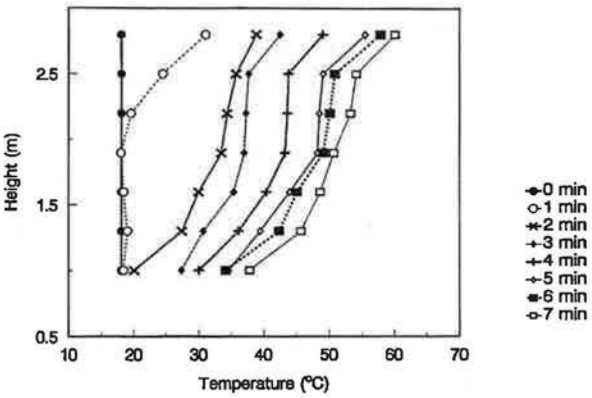

Figure 4 *Vertical temperature profile of wood fires at extraction rate of 30 air changes per hour.*

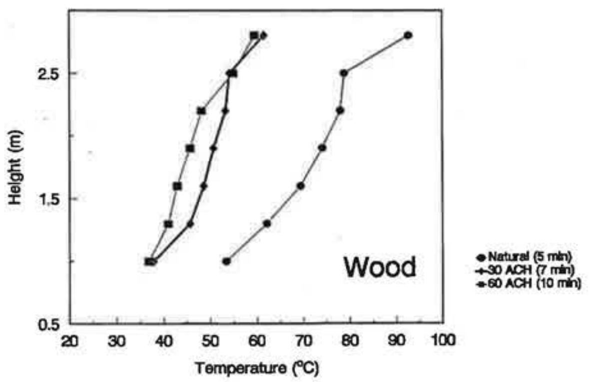

Figure 5 *Vertical temperature profiles at steady burning (wood).*

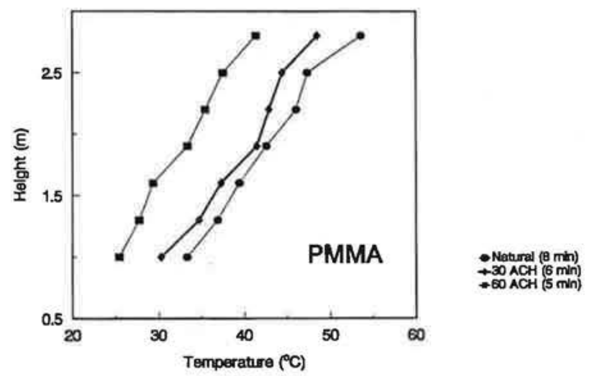

Figure 6 *Vertical temperature profiles at steady burning (PMMA).*

The heat of combustion ΔH_c for PMMA (Tewarson 1976) is 25.3 kJ/g (8.5×10^6 ft·lb$_f$/lb) and 19.83 kJ/g (6.6×10^6 ft·lb$_f$/lb) for methanol (Drysdale 1985). For wood cribs, ΔH_c is taken to be 12.83 kJ/g (4.3×10^6 ft·lb$_f$/lb) as suggested by Janssens (1991). The heat release rate of methanol is fairly constant, but there is a peak value for PMMA and wood fires. The peak heat release rates are calculated from the maximum mass loss rates and are plotted in Figure 3.

The vertical variation of the temperatures was measured by the seven thermocouples located at point A in Figure 2. A typical example of the transient temperature profiles for the wood fire with an extraction rate of 30 air changes per hour, test 4, is shown in Figure 4. The variation of temperature profiles for steady burning of wood fires at extraction rates of 0, 30, and 60 air changes per hour, tests 1, 4, and 7, are plotted in Figure 5. Results for PMMA and methanol fires at the same ventilation conditions, tests 8, 11, 14, 15, 18, and 21, are shown in Figures 6 and 7, respectively. As shown in Figure 5, the thermal stratified layer became unstable at large gas

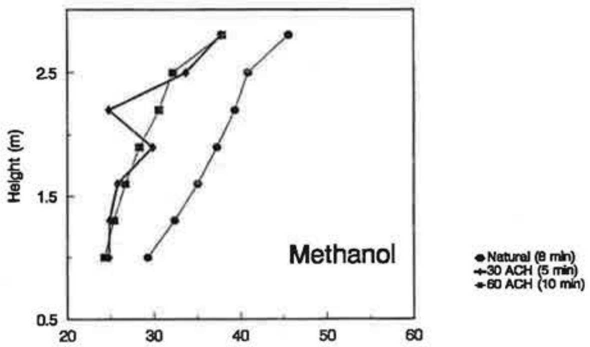

Figure 7 *Vertical temperature profiles at steady burning (methanol).*

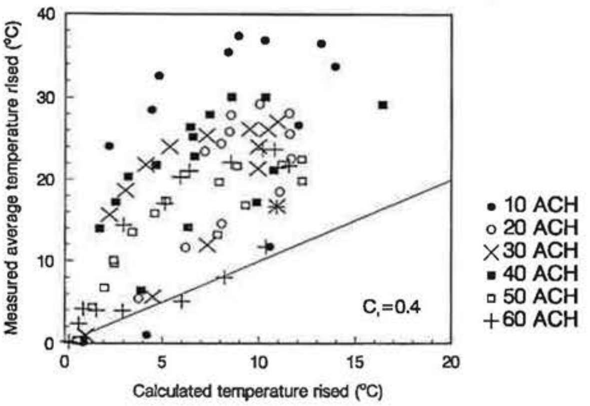

Figure 8 *Calculated temperatures for wood fire tests at different ventilation rates.*

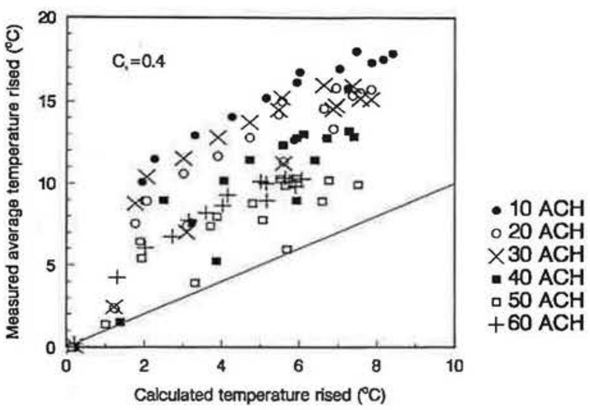

Figure 9 *Calculated temperatures for PMMA fire tests at different ventilation rates.*

extraction rates. There were no thermal stratified layers for PMMA and methanol fires under these ventilation conditions.

The measured temperature rise δT of the hot gas inside the compartment was calculated by a weighted average of the temperatures measured by the seven thermocouples at T_1 to T_7 minus the initial value T_0:

$$\delta T = \frac{1.15T_1 + 0.3(T_2 + T_3 + T_4 + T_5 + T_6) + 0.15T_7}{2.8} - T_0 . \quad (2)$$

PREDICTION OF THE HOT GAS TEMPERATURE

A forced-ventilation fire is very different from a natural-ventilation fire. For example, the thermally stratified layer was unstable at higher ventilation rates. A simple model for calculating the average temperature in a forced-ventilation compartment was proposed by Deal and Beyler (1990). There, the average gas temperature rise δT in the compartment above the ambient can be calculated from the volumetric flow rate, $\dot{V}$,

$$\delta T = \frac{Q_t}{\dot{V}\rho C_p + h_k A_T} \quad (3)$$

where ρ and Cp are density and specific heat capacity of air, Q_t is the heat release rate, h_k is the heat loss coefficient of the compartment, and A_T is the total surface area of the compartment. The heat loss coefficient h_k included both convective and radiative effects of the walls, ceilings, and floors. As proposed by McCaffrey et al. (1981) for natural-ventilation fires and later modified by Deal and Beyler (1990) and Beyler (1991), for fires h_k can be approximated by the expression below when the time t is smaller than the thermal penetration time t_p:

$$h_k = C_1 Max\left\{\sqrt{\frac{k_w \rho_w c_w}{t}}, \frac{k_w}{\delta_w}\right\} \quad (4)$$

where t_p is, in fact, the time taken for the thermal wave generated inside the room to reach the exterior surface of the wall and is given by

$$t_p = \frac{\rho_w C_w}{k_w}\left(\frac{\delta_w}{2}\right)^2 . \quad (5)$$

In the above expression, t is the time, k_w, ρ_w, C_w, and δ_w are the thermal conductivity, density, specific heat capacity, and thickness of the wall, respectively. The parameter C_1 is taken to be 0.4 (Deal and Beyler 1990) for having a better fit on the experimental data. For this fire chamber, k_w is 0.98 W/(m·K) (or 6.8 Btu·in./(h·ft^2·°F), ρ_w is 2300 kg/m^3 (144 lb/ft^3), C_w is 653 kJ/(kg·K) (0.156 Btu/(lb·°F)), $\sqrt{k_w \rho_w C_w}$ is about 1213 W/(m^2·K)·s$^{1/2}$). The total surface area of the chamber A_T is 63.2 m^2 (680 ft^2) and the thickness δ_w is about 5 cm (0.16 ft). The thermal penetration time t_p is calculated to be 958 s using Equation 5. The times taken for all the fire tests were smaller than this value and so Equation 4 is applied.

Transient values of δT are calculated from Equation 3. The measured values are plotted against the calculated values from Figures 8-10. The calculated temperatures were much smaller than the measured values. The problem lies on calculating the heat loss coefficient of the chamber. A possible reason is C_1 being too high. As suggested by Deal and Beyler (1990), this value can be fitted by measuring the transient temperature. This is possible in this experiment, and a better fit is achieved when C_1 is 0.165 (or $h_k = \frac{200}{\sqrt{t}}$ W/(m^2·K)). With this smaller value of C_1, δT is calculated. The predicted temperatures more closely match with the measured values for this case ($C_1 = 0.165$) when compared to the original case ($C_1 = 0.4$) with a typical example for a methanol fire shown in Figure 11.

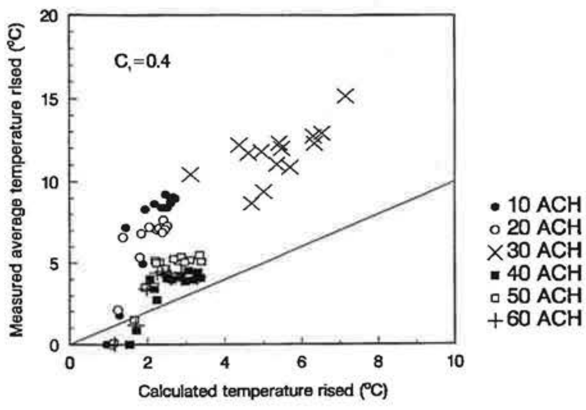

Figure 10 *Calculated temperatures for methanol fire tests at different ventilation rates.*

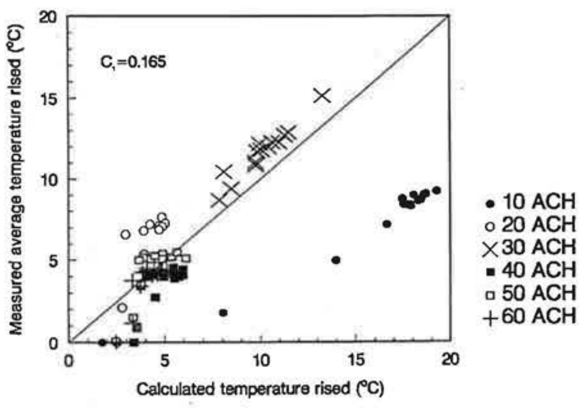

Figure 11 *Calculated temperatures for methanol fire tests at different ventilation rates and smaller heat loss coefficients of chamber.*

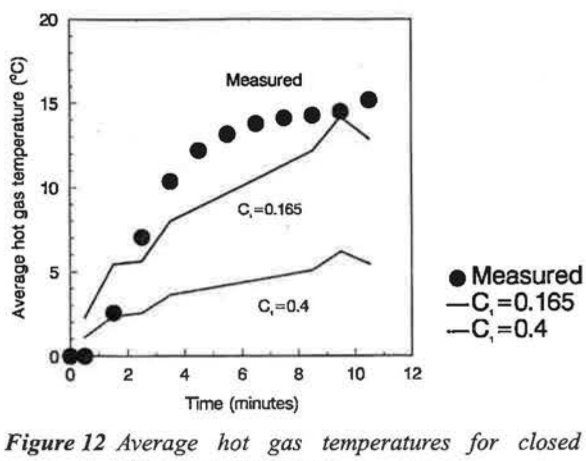

Figure 12 *Average hot gas temperatures for closed chamber methanol fire.*

CLOSED CHAMBER FIRES

When the extraction fans are switched off, the room can be treated as a closed chamber. The average gas temperature in the chamber is given by

$$mC_p\frac{dT}{dt} = Q_t - h_k A_T \delta T, \tag{6}$$

where m is the mass of air in the chamber. For constant heat release rate Q_t and using the expression for h_k given by Equation 4 with $C_1 = 0.4$, the temperature rise δT can be solved (Beyler 1991),

$$\delta T = \frac{2K_2}{K_1^2}\left\{K_1\sqrt{t} - 1 + e^{-K_1\sqrt{t}}\right\} \tag{7}$$

where

$$K_1 = \frac{2(0.4\sqrt{k_w \rho_w C_w})A_T}{mC_p}, \tag{8}$$

$$K_2 = \frac{Q_t}{mC_p}. \tag{9}$$

Putting in numerical data and taking m as 32.54 kg (71.7 lb),

$$K_1 = \frac{2(0.4 \times 1213) \times 63.2}{32.54 \times 1009} = 1.8679,$$

$$K_2 = \frac{Q_t}{32.54 \times 1009} = \frac{Q_t}{32832}.$$

In this experimental study, the heat release rate for the methanol fire in a closed chamber was fairly constant at about 7500 W (25,598 Btu/h); therefore, it was utilized to evaluate the above expressions. The calculated temperature time curve is plotted together with the measured values in Figure 12. Again, the predicted temperatures are much lower than the measured values for $C_1 = 0.4$, but when C_1 is changed to 0.165, the predicted results more closely track with the measured temperatures. Therefore, the choice of this value of C_1 is good for calculating the heat loss coefficient of the chamber.

CONCLUSIONS

Twenty-one forced-ventilation fire tests for wood, PMMA, and methanol were performed to evaluate the simple model proposed by Deal and Beyler (1990) for calculating the hot gas temperature before flashover. The fire sizes of the tests were smaller than 40 kW and the temperature of the hot gas rise was less than 50°C (122°F). In this small range of temperature rise, the measured data are compared with the values calculated by the simple model. Good agreement was found when the coefficient C_1 in the expression for calculating the heat loss coefficient of the chamber was 0.165. In addition, this coefficient was satisfactory for modeling the temperature rise in a closed chamber fire. Equations 3 and 4 are good for quick estimation of the hot gas temperature due to a small fire in a forced ventilation compartment. This will be very useful for engineers to determine the probable fire environment, especially for refurnishing works in small offices. For a more detailed investigation in a bigger compartment with forced ventilation, computational fluid dynamics (Chow and Mok 1995) might be suitable, but that would have to be supported by further experimental studies.

NOMENCLATURE

A_T	= total surface area of the chamber
C_p	= specific heat capacity of air
C_1	= coefficient in forced-ventilation equation
C_w	= specific heat capacity of wall
h_k	= heat transfer coefficient of wall
K_1, K_2	= coefficients in temperature equation
ΔH_c	= heat of combustion
k_w	= thermal conductivity of wall
$\dot{M}_f$	= mass burning rate
m	= mass of air in the chamber
Q_t	= average total heat-release rate within quasi-steady burning period
t	= time
t_p	= thermal penetration time of wall
$\dot{V}$	= gas extraction rate
ρ	= density of air
ρ_w	= density of wall
δT	= average gas temperature rise above ambient within quasi-steady burning period
δ_w	= thickness of wall

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