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Estimation of air temperature in a compartmental fire with forced ventilation using ventilation theory

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Abstract

A simple model is proposed to calculate the air temperature in a compartmental fire with forced ventilation. The model is based on the mixing process in displacement ventilation theory. This is reasonable as the air intake rate in a forced ventilation chamber can be fixed at a certain value. Expressions for the hot air temperature are derived by considering the cases with and without a plume region. Experimental studies in a small chamber with forced ventilation were performed to verify the predicted results.

1. Introduction

Forced ventilation systems are installed in most of the commercial buildings in Hong Kong [1],[2]. The ventilation rate depends on the floor area and usually, smaller the floor area, larger is the ventilation rate per floor area. The design ventilation rate depends also on the use of the area, ranging from 10 air changes per hour in a large office to 50 air changes per hour for a small toilet. A fire occurring in such a forced-ventilation compartment will be very different from that a building with natural ventilation. Firstly, a thermal stratified hot smoke layer might not be formed due to the interaction between the air flow induced by the fire and the ventilation system. Locations of air intake at a higher level would be an obvious example. In this way, assumptions made in traditional zone models would not apply. Secondly, even if a hot smoke layer is formed, say for the case with air intake at low level, air exhaust at high level, the air entrainment rate towards the plume might be neglected by plume expressions available in literature for natural ventilation fire. Estimation of the fire temperature inside those compartments using the theories for a natural ventilated fire is then not good enough.

Experimental studies were performed [3] with the data compiled and modelled [4],[5], leading to the derivation of a simple model for estimating the average hot gas temperature in the compartment. The use of computational fluid dynamics (CFD) had also been reported [2],[6]. For a fire occurred in a forced ventilation chamber, the amount of air entrained into the fire plume depends on the ventilation rate. It would be limited to a fraction of the air intake rate and is different from the case in a natural ventilation chamber. Suppose the air intake is at a lower position, and the air extraction is at the upper location as shown in Fig. 1, a stable hot layer can be formed. The proportion of the air which is not entrained into the fire plume would move to the upper layer. Also part of the air in the upper part would move down to the lower part of the chamber due to the interactions of the intake air and the fire induced air motions. The mixing theory proposed by Sandberg [7] can be applied to study the air flow. In fact this is similar to the mixing mechanism used in displacement ventilation [8]. A simple model is presented here using this concept of mixing parameter in ventilation theory [7].

2. The Model

A simple mathematical model is developed for forced ventilation fire. Consider a room of volume V_{RM} with a fire of heat release rate Q. There are two ventilation fans with the intake position at a lower level and the exhaust position at a higher level but on the opposite wall as in Fig. 1. Using the technique of Computational Fluid Dynamics [e.g. 2], a rather stable thermal stratified layer would be found under this situation and so there are two zones in the room. The upper hot zone is at temperature T_H and the lower cool zone is at temperature T_L. Although there are two distinct zones, both experimental observation and simulation by computational fluid dynamics indicated that there would be mixing between the two zones. Suppose the air intake rate is $\dot{m}_{in}$ and the exhaust rate is $\dot{m}_{out}$. In the lower zone, a fraction of air α is used for combustion. By mass balancing, the oxygen consumed would generate a mass flow rate of combustion production $\dot{m}_p$ which is then transferred to the upper zone through the plume. The plume is assumed to be at a constant temperature T_p. Therefore, the outward mass flow rate $\dot{m}_{out}$ has two components: the mass flow rate of unreacted air (1 - α) $\dot{m}_{in}$ and the mass flow rate of product $\dot{m}_p$. By conservation of mass:

$$\dot{m}_{out} = (1 - \alpha)\ \dot{m}_{in} + \dot{m}_p \approx \dot{m}_{in} \qquad \text{... (1)}$$

or $$\dot{m}_p \approx \alpha\ \dot{m}_{in} \qquad \text{... (2)}$$

(a) Without a plume region:

The plume is taken as an imaginary region for transferring heat to the upper zone only. There is no mass transfer to and out of the plume. A flow diagram can be drawn for the mass transfer between the two zones as shown in Fig. 2. A mixing factor β [7] is defined to describe the exchange flow rate between the two zones. The conservation of enthaply at the steady-state would give:

$$\frac{Q}{C_p} + \dot{m}_{in}T_{in} + \beta(1-\alpha)\dot{m}_{in}T_H = (1-\alpha)(1+\beta)\dot{m}_{in}T_L + \alpha\dot{m}_{in}T_p \qquad \text{... (3)}$$

$$\alpha\ \dot{m}_{in}\ T_p + (1-\alpha)\ (1+\beta)\ \dot{m}_{in} T_L = \beta(1-\alpha)\dot{m}_{in} T_H + \dot{m}_{in} T_H \qquad \text{... (4)}$$

There are two equations with three unknowns T_L, T_H and T_p. Taking T_L to be same as T_{in}, this gives:

$$T_H = \frac{Q}{\dot{m}_{in} Cp} + T_L \qquad \text{... (5)}$$

$$T_p = \frac{Q}{\dot{m}_{in} Cp\alpha}\left[1 + (1-\alpha)\beta\right] + T_L \qquad \text{... (6)}$$

(b) With a plume region:

Secondly, the plume is considered as a sepearate region with heat and mass transfer. A flow diagram shown in Fig. 3 is obtained. By using the conservation law of enthaply at the steady-state, three equations on the lower zone, plume and upper zone can be derived:

Lower zone:

$$\dot{m}_{in} T_{in} + (1-\alpha)\beta\dot{m}_{in} T_H = (1-\alpha)(1+\beta)\dot{m}_{in} T_L + \alpha\dot{m}_{in} T_L \qquad \text{... (7)}$$

Plume:

$$Q = \alpha\dot{m}_{in} C_p\ T_p - \alpha\dot{m}_{in}\ C_p\ T_L \qquad \text{... (8)}$$

Upper zone:

$$\alpha\dot{m}_{in} T_p + (1-\alpha)(1+\beta)\dot{m}_{in} T_L = \dot{m}_{in} T_H + (1-\alpha)\beta\dot{m}_{in} T_H \qquad \text{... (9)}$$

Simplying expressions (7) to (9) gives:

$$T_{in} + (1-\alpha)\beta T_H - [(1-\alpha)(1+\beta) + \alpha] T_L = 0 \qquad \text{... (10)}$$

$$T_p = \frac{Q}{\alpha \dot{m}_{in} Cp} + T_L \qquad \text{... (11)}$$

$$\alpha T_p + (1-\alpha)(1+\beta)T_L - [1+(1-\alpha)\beta]T_H = 0 \qquad \text{... (12)}$$

Further simplication would give:

$$T_L = \frac{\frac{Q}{\dot{m}_{in} Cp}(1-\alpha)\beta + [1+(1-\alpha)\beta]T_{in}}{[\alpha + (1-\alpha)(1+\beta)]} \qquad \text{... (13)}$$

$$T_p = \frac{Q}{\dot{m}_{in} Cp\alpha} + \frac{\frac{Q}{\dot{m}_{in} Cp}(1-\alpha)\beta + [1+(1-\alpha)\beta]T_{in}}{[\alpha + (1-\alpha)(1+\beta)]} \qquad \text{... (14)}$$

$$T_H = \frac{Q}{\dot{m}_{in} Cp} + T_{in} \qquad \text{... (15)}$$

3. No Mixing

Relatively simpler expressions can be derived when there is no mixing between the two zones.

(a) Without a plume region:

Assuming value of T_{in} is same as T_L and for no mixing between the two zones, β is zero. Putting into equation (5), the plume temperature is now:

$$T_p = \frac{Q}{\dot{m}_{in} Cp\alpha} + T_L \qquad \text{... (16)}$$

(b) With a plume region:

Putting $\beta = 0$ would give $[\alpha + (1-\alpha)(1+\beta)] = 1$, and putting into equations (13) to (15) would give:

$$T_L = T_{in} \qquad \text{... (17)}$$

$$T_p = \frac{Q}{\dot{m} Cp\alpha} + T_{in} \qquad \text{... (18)}$$

$$T_H = \frac{Q}{\dot{m}_{in} Cp} + T_L \qquad \text{... (19)}$$

These expressions are the same as the case without a plume region when there is no mixing as in equation (16). And they are similar to the ones derived by using the well-mixed theory with heat loss taken into account [5].

4. Experimental Verification

Experiments were conducted in a small test chamber [9] of length 1 m, width 0.65 m and height 1 m as shown in Fig. 4. The inner and outer surfaces of the test chamber were made of steel with thickness $_s$ of 0.002 m, thermal conductivity k_s of 0.0458 kW m^{-1} C^{-1}, density $_s$ of 7,850 kgm^{-3} and specific heat c_{ps} of 0.46 kJ kg^{-1} C^{-1}. Fiberglass insulation inserted in between the steel wall has thickness $_f$ of 0.026 m, thermal conductivity k_f of 3.6 x 10^{-5} kW m^{-1} C^{-1}, density $_f$ of 140 kgm^{-3} and specific heat c_{pf} of 0.96 kJ kg^{-1} C^{-1}. There were two axial fans [10] installed at two square vents, each of size 0.24 m by 0.24 m opposite to each other, with one at high level of 0.12 m below the ceiling, and the other at low level of 0.52 m above the floor. The air inlet and outlet positions were altered so that they can be either at high level or at low level. The incoming air temperature was about 23°C.

Four experiments (labelled as from A to D) with air supplied at low level and exhausted at high level was conducted. The ventilation rates $\dot{m}_{in}$ (kgs^{-1} or ℓs^{-1} or air changes per hour ACH) in the experiments were set to 0.1292 kgs^{-1} (100 ℓs^{-1} or 636 ACH), 0.0969 kgs^{-1} (75 ℓs^{-1} or 486 ACH), 0.0646 kgs^{-1} (50 ℓs^{-1} or 318 ACH) and 0.0303 (25 ℓs^{-1} or 162 ACH) by varying the face area of the opening. The vertical temperature profiles were measured by type K thermocouples of chromel-alumel wires. All the thermocouples were connected to a data logging system for recording the transient variation of temperature. The fuel used for the tests was ethanol of mass 0.2 kg placed in a pan of 0.21 m diameter as shown in Fig. 4. The rate of fuel loss was measured by an insulated electronic balance at the bottom of the pan. The whole set of experiment was repeated using a smaller pan of diameter 0.16 m in order to check for any deviation of results. Results on the burning time t_B, together with the maximum and average temperature, and the smoke layer thickness are summarized in Table 1.

There were two methods to measure the heat release rate for the chamber at different ventilation rate. The first method is to heat release rate by multiplying the measured mass loss rate $\dot{m}_1$ to the heat of combustion of the ethanol $\dot{H}$ and is labelled as Q_m:

$$Q_m = \dot{m}_1 \Delta H \qquad \text{... (20)}$$

The second method is to calculate the heat release rate using the oxygen consumption method [11],[12] and is labelled as Q_o. By measuring the percentage concentration of oxygen $_{02}$ at the exhaust duct and the volumetric air flow V, taking the density of oxygen $_{02}$ to be 0.2993 kgm^{-3} and the heat of combustion of fuel in terms of oxygen $H_{c,ox}$ to be 12880 kJ kg^{-1}, Q_o can be found by:

$$Q_o = (0.21 - \eta_{02}) V 10^3 \rho_{02} \Delta H_{c,0x} \qquad \text{... (21)}$$

Maximum and averaged values of Q_m and Q_o are shown in Table 1 as well. A steady-state flow pattern was found within the steady burning period t_s and so the average heat release rate was used to calculate the temperature T_L, T_p and T_H given by the expressions derived in the above section.

5. Results

Results for the temperature calculated by the model without a plume region given by equation (5) are shown in Fig. 5. Note that both sets of heat release rate Q_m and Q_o were used and similar results were derived. The average temperature is taken to be the average values of the hot and cold layer temperature calculated by measuring the smoke layer thickness z (in m) and taking the ceiling height to be 1 m:

$$T_{av} = T_H z + T_L (1 - z) \qquad \text{... (22)}$$

The plume is taken to be an imaginary surface having a very small volume in comparing with the volume of the room. It can be seen that fairly good agreement is found between the experimental results and the predicted results. In fact, the average temperature predicted should be the same for the case with a plume region.

Values of the average temperature calculated from the model with a plume region using equations (13) and (15) and the heat release rate Q_o for different mixing factors β are shown in Fig. 6. Good agreement can be fitted by selecting suitable values of the mixing factor β. However, the mixing factor cannot be determined easily. Computational Fluid Dynamics can be used to calculate a set of mixing factors under different ventilation rates and conditions (e.g. [13-17]).

6. Conclusions

A simple model based on the mixing process in ventilation theory is proposed to estimate the hot gas temperature in a forced ventilation chamber with air intake at a lower position and air exhaust at higher position. The justification is that the air entrainment in a fire plume would be limited by the air intake rate and is kept at a factor α of it. The proportion of the unreacted air would be transferred into the upper layer. A proportion of the air at the upper layer would move back to the lower level and mix with it. This is different from the case in a natural ventilation chamber although the ventilation factor there is important in determining the air entrainment rate. A mixing factor as proposed by Sandberg [7] can be defined. Based on this concept, expressions for the hot layer temperature, cool layer temperature and the plume temperature are derived for cases with and without considering heat and mass transfer in the plume region. The choice of the mixing factor is important and this can be computed using Computational Fluid Dynamics [13-17]. The mixing factor depends on the ventilation rates, locations of the air intake and exhaust, and the heat release rate of the fire. Finally, experimental results for a model chamber with forced ventilation [9] are used to verify the predicted results. This supports the argument that the choice of mixing factor is important.

7. Additional Comments

The paper was published as a journal paper [18]. With the practice of publisher, preprint is allowed to upload at website. There are further studies on this topic [19-36].

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Table 1: Results

Case	Ventilation rates $\dot{m}_{in}$			t_b	Maximum value Q_{max}		Average value Q_{ave}		$T_{max}(t_1)$	$T_{av}(t_2)/t_s$	z_{max}	α
	$/kgs^{-1}$	$/\ell s^{-1}$	ACH	/s	Q_m/kW	Q_o/kW	Q_m/kW	Q_o/kW	/C	/C (s)	/m	
A	Bigger fire: Both outlet and inlet with fans											
	0.1292	100	636	300	22.54	-	19.37	-	122.9	98.8(180)	0.45	-
	0.0969	75	486	360	19.12	-	17.09	-	124.1	94.7(240)	0.55	-
	0.0646	50	318	360	23.21	-	19.77	-	264.9	167(180)	0.35	-
	0.0323	25	162	360	26.48	-	21.01	-	242.5	191(240)	0.45	-
B	Smaller fire: Both outlet and inlet with fans											
	0.1292	100	636	540	12.57	12.98	11.74	11.02	50.4	46(270)	0.65	0.136
	0.0969	75	486	570	12.35	10.98	10.97	9.73	85.3	77.2(330)	0.65	0.16
	0.0646	50	318	570	12.65	8.99	10.77	8.85	123.9	113(300)	0.65	0.218
	0.0323	25	162	570	12.2	6.99	11.26	6.42	195.7	167(270)	0.65	0.317
C	Bigger fire: Only outlet with fan											
	0.1292	100	636	420	14.58	19.64	13.97	18.67	100.6	79.5(240)	0.35	0.231
	0.0969	75	486	420	15.84	22.22	14.53	17.33	125.5	107.9(300)	0.35	0.285
	0.0646	50	318	420	20.68	18.97	15.81	17.97	209.4	173.5(240)	0.45	0.444
	0.0323	25	162	420	23.21	19.89	16.42	14.24	361.4	282(240)	0.35	0.704
D	Smaller fire: Only outlet with fan											
	0.1292	100	636	570	12.2	14.98	10.49	14.19	65.7	59.5(420)	0.55	0.175
	0.0969	75	486	690	10.49	12.23	9.74	11.32	96.1	84.6(360)	0.65	0.186
	0.0646	50	318	690	13.09	12.23	9.21	8.99	98.3	85.6(450)	0.55	0.222
	0.0323	25	162	660	10.49	6.99	9.35	6.68	148.2	138.4(480)	0.55	0.33

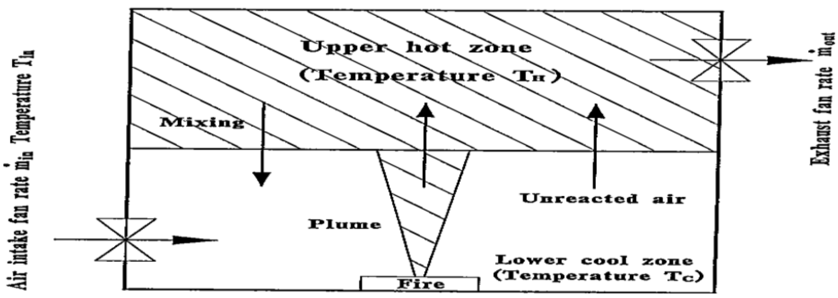
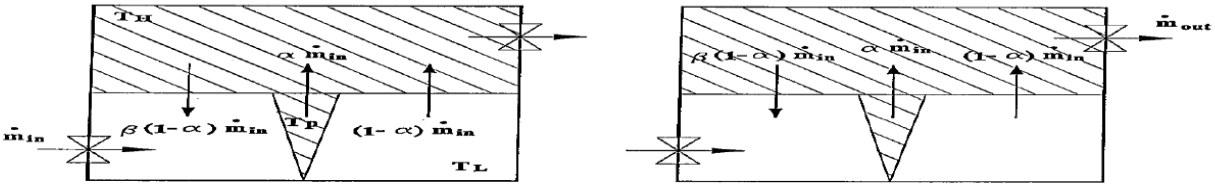
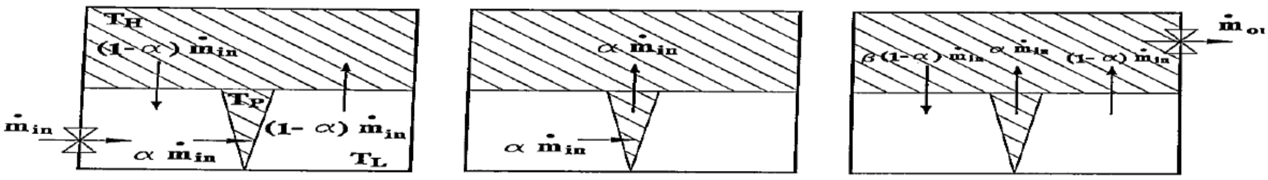


Fig. 1: Geometry of the problem



(a) Lower zone (b) Upper zone

Fig. 2: Flow path without considering the plume region



(a) Lower zone (b) Plume (c) Upper zone

Fig. 3: Flow path with a plume region

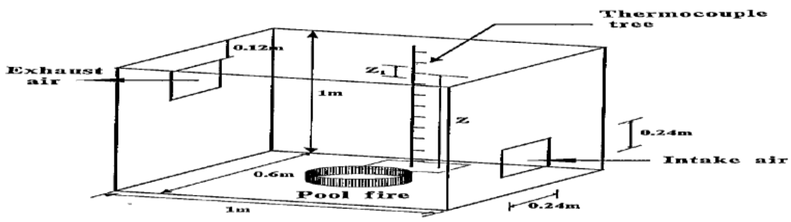


Fig. 4: Geometry of the scale model chamber

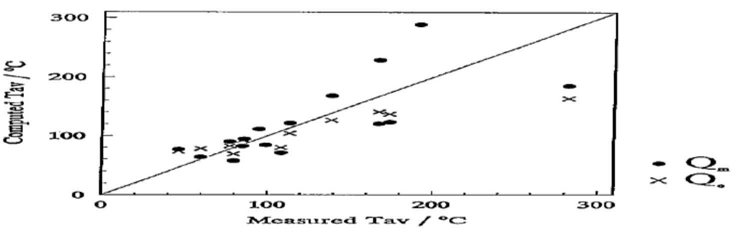


Fig. 5: Predicted temperature without a plume region

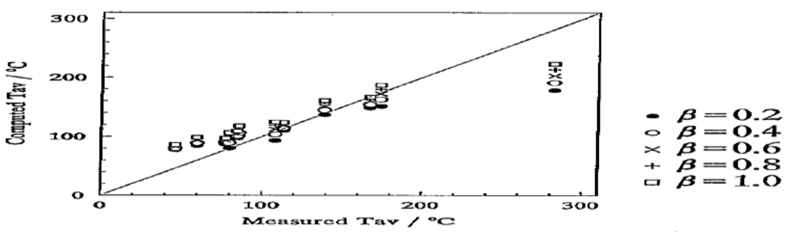


Fig. 6: Predicted temperature with a plume region