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CORRELATING PREFLASHOVER ROOM FIRE TEMPERATURES

by

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SUMMARY

In this investigation the existing correlational methods for predicting room fire temperatures are evaluated by comparison with a database of 559 data points derived from over 250 room fire experiments. In addition, several new methods based on a simple energy balance are proposed and evaluated.

For forced ventilation compartment fires, the energy equation method developed here is shown to provide better predictions than existing methods. For single opening naturally ventilated fires, the McCaffrey, Quintiere, Harkleroad (MQH) method provides better predictions than the layer driven method developed here. In many situations, there is no practical way to have prior knowledge of the ventilation rates or the ventilation openings. Under these conditions, it is prudent to make conservative assumptions concerning the ventilation rate. The stoichiometric model described in this paper embodies this assumption and has been shown to yield conservative results. This method may be very useful in conditions where the burning rate can be estimated, but the ventilation rate cannot be anticipated.

INTRODUCTION

While computer based compartment fire models are becoming increasingly available, it is often necessary to make estimates of preflashover room fire temperatures and assessments of the likelihood of flashover using simpler methods. Over the years, a number of correlational methods for estimating preflashover room fire temperatures and conditions required for flashover have been developed. In this paper, we review the available methods and propose several new methods for consideration. The existing and new methods are evaluated using an experimental database developed from literature. Based on these evaluations, recommendations for the best available methods are made.

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REVIEW OF EXISTING METHODS FOR EVALUATING FLASHOVER POTENTIAL

The existing correlational methods for predicting flashover can be categorized as those which use a temperature correlation and those which do not. Babrauskas[1] and Thomas[2] have developed methods which do not use a temperature correlation. In these methods, the variables considered are the rate of heat release and the ventilation factor, $A_o\sqrt{H_o}$. Wall heat losses are not explicitly considered.

McCaffrey *et al.*[3] , Mowrer and Williamson[4], and Foote *et al.*[5] have developed correlations for preflashover room fire temperatures based on the rate of heat release, the wall properties, and the ventilation. These correlations can be used to predict flashover using an assumed layer temperature required for flashover. While there is no fundamental basis for a critical layer temperature at flashover, many investigators have

observed that the onset of flashover is indicated by layer temperatures of 500-650°C[6] . Indeed, there is no single definition of flashover which is universally applicable.

Several common definitions of flashover used in the literature include 1) ignition of fuels throughout the room by hot layer radiation, 2) ventilation controlled burning in the room, 3) ignition and burning of the hot layer as indicated by the emergence of flaming hot layer gases from the room. In the remainder of this section the existing correlational methods for predicting flashover and preflashover room fire temperatures will be reviewed.

Babrauskas

Using a compartment energy balance and a simple energy loss model, Babrauskas[1] developed a relationship between the ventilation parameter and the rate of heat release required to cause flashover. Based on a database of 33 compartment fire tests, he found that the rate of heat release required to cause flashover, $\dot{Q}_{fo}$, is described by the relation

$$\dot{Q}_{fo} = 750\, A_o \sqrt{H_o}\,[\mathrm{kW,m}] \qquad (1)$$

This corresponds to half the heat release rate which would occur for stoichiometric burning utilizing all available air if the vent flow rate is fully choked, i.e.

$$\dot{m}_a = 0.5\, A_o \sqrt{H_o}\,[\mathrm{kg/s,m}]$$

Using 3 MJ/kg as the heat released per unit mass of air consumed, this yields a stoichiometric rate of heat release of

$$\dot{Q}_{stoich} = 1500\, A_o \sqrt{H_o}\,[\mathrm{kW,m}] \qquad (2)$$

Among the experiments in the Babrauskas database, the ratio of $\dot{Q}_{fo}$ *to* $\dot{Q}_{stoich}$ ranged from 0.23 to 0.86, with an average of 0.5.

Equation 1 is an extremely simple and easy to use relation, though it does not take into account the wall surface area or the thermal properties of the wall.

Thomas

Thomas[2] developed a simple correlational form for the heat release rate required for flashover based on a simple energy balance and assuming choked flow through the vent. His expression takes the form

$$\dot{Q}_{fo} = 7.8\, A_T + 378\, A_o \sqrt{H_o}\,[kW,m] \qquad (3)$$

where A_T is the internal bounding surface area of the compartment. While this expression does not include the thermal properties of the wall material, it does attempt to account for the effect of wall surface area. The Thomas and Babrauskas expressions match for $A_T/A_o\sqrt{H_o}$ = 47.7 $m^{-1/2}$, a value in the midrange of typical fire test compartments.

McCaffrey, Quintiere, Harkleroad (MQH)

Motivated by a simple energy balance for the compartment, McCaffrey *et al.*[3] correlated room fire temperature data using the form

$$\frac{\Delta T}{T_\infty} = C\left(\frac{\dot{Q}}{c_p\, \rho_\infty\, A_o \sqrt{g\, H_o}}\right)^m \left(\frac{h_k\, A_T}{c_p\, \rho_\infty\, A_o \sqrt{g\, H_o}}\right)^n \qquad (4)$$

Based on the correlation of 112 experiments, they found that C = 1.5, m = 0.650, and n = –0.387 gave the best fit if the bounding surface area, A_T, included the floor area. For convenience McCaffrey *et al.* suggest using the following representation of the fit.

$$\frac{\Delta T}{T_\infty} = 1.63\left(\frac{\dot{Q}}{c_p\, \rho_\infty\, A_o \sqrt{g\, H_o}}\right)^{2/3} \left(\frac{h_k\, A_T}{c_p\, \rho_\infty\, A_o \sqrt{g\, H_o}}\right)^{-1/3} \qquad (5)$$

This can be further simplified by substituting the values for ambient conditions.

$$\Delta T = 6.85\left(\frac{\dot{Q}^2}{h_k\, A_T\, A_o \sqrt{H_o}}\right)^{1/3} \qquad (6)$$

In Equations 4-6 the heat loss coefficient is given by

$$h_k = \sqrt{\frac{k\, \rho\, c_p}{t}} \text{ for } t < \frac{\alpha\, \delta^2}{4}$$
$$h_k = \frac{k}{\delta} \text{ for } t > \frac{\alpha\, \delta^2}{4} \qquad (7)$$

where δ is the wall thickness and α is the thermal diffusivity of the wall ($k/\rho\, c_p$). It is of interest to note that Equation 7 yields a discontinuity in h_k at $t = (\alpha\, \delta^2)/4$.

In the heat loss model, t is the time of exposure. For purposes of this paper the time of exposure is

the time after established burning. In general established burning is the state where the fire is of sufficient size that it is unlikely to self-terminate. For most fire experiments the initial ignition source is sufficiently large to make the difference between the ignition time and established burning negligible. Where this is not the case we have used 10 kW as the fire size at established burning.

The MQH model includes the effects of ventilation through the ventilation parameter, $A_o\sqrt{H_o}$, the effects of the internal bounding surface area, A_T, as well as the effect of the thermal properties of the wall and time dependent wall losses using Equation 7. The correlation includes data with compartment heights from 0.3 to 2.7 m and floor areas from 0.14 to 12 m^2. The data also includes a variety of door and window openings. Only single opening naturally ventilated fires are included in the experimental database.

Mowrer and Williamson

Mowrer and Williamson[4] developed corrections to the MQH method to account for the effect of the proximity of the fire to walls or corners. They found that for the wall configuration the temperature rise was 1.3 times the MQH method prediction, and for the corner configuration the temperature rise was 1.7 times the MQH method. Mowrer and Williamson reported data for only three heat release rates, one door opening size, and one compartment size.

Foote, Pagni, Alvares (FPA)

Foote, Pagni, and Alvares[5] conducted forced ventilation compartment fire experiments in which the gases were extracted from the top of the compartment. Recognizing that the power law correlations like the MQH are prone to non-physical behavior in limiting cases, Foote *et al.* used an exponential correlation which does behave reasonably in limiting cases. Based on 35 methane burner experiments and 15 data points from naturally vented compartment fires conducted at Center for Fire Research, National Bureau of Standards (with measured exhaust flow rates), Foote *et al.* found the correlation

$$\Delta T = \left(\frac{\dot{Q}}{\dot{m}_o\, c_p}\right) \exp\left(-0.53\left(\frac{h_k A_T}{\dot{m}_o\, c_p}\right)^{0.43}\right) \qquad (8)$$

where the exhaust rate is known and h_k is determined using Equation 7. The successful correlation of both natural and forced ventilation experiments suggests that there is no inherent difference between these two types of compartment fires.

Experimental Database

In order to better understand the accuracy and range of applicability of existing correlations and to assist in the evaluation of new correlations, an attempt was made to assemble a large database of compartment fire data, including a wide range of experimental conditions. The data sources for the database are summarized in Table 1.

The database contains the following pieces of information: the room height, the fire source location and fuel, the ambient conditions, the vent dimensions and sill height, the bounding surface(s) properties (thermal properties, thickness and area), the time, heat release rate, hot gas layer temperature and depth, and exhaust rate (where measured). In many cases the original investigator processed the data to determine all of the above information. In other cases it was necessary to convert the measured mass loss rate to a heat release rate by multiplying by the heat of combustion and the combustion efficiency. The combustion efficiencies used are shown in Table 2.

In addition it was often necessary to process the temperature profile measured in the room to determine the hot and cold layer depths and temperatures. Because the room height is the sum of the two layer depths, there are three unknowns to be determined. There are three integral averaging methods which have been used to determine these quantities: the hot layer temperature method[7], the layer interface method[8], and the cold layer temperature method[9]. These will be described below.

Each of these methods requires that one of the unknowns be determined *a priori* and the two remaining variables be determined by integral averaging methods. Conservation of mass requires that the following integral be conserved by the two layer representation.

$$B_1 = \int_{z_b}^{z_u} \frac{1}{T(z)}\, dz \qquad (9)$$

where z_b and z_u are the height of the bottom and top of the vent, respectively, and $T(z)$ is the temperature in the compartment at height z. The

Table 1. Database Sources

[A] Croce, P.A., Editor, "The Large Scale Bedroom Fire Test, July 11, 1973", Serial No. 21011.4, RC74-T-31, Factory Mutual Research Corp., 1974.

[B] Alpert, R.L., Modak, A.T., Newman, F.S., "Third Full-scale Bedroom Fire Test of the Home Fire Project", Volume 1, RC-B-48, Factory Mutual Research Corp., 1975.

[C] Alpert, R.L. *et al.*, "Influence of Enclosures on Fire Growth", RC-BT-8 through 13, 1977.

[D] Alvares, N.J., Foote,K.L., Pagni, P.J., Hasegawa, , H.K., "Fire Protection Research for Department of Energy Facilities-Fiscal Year End Report", UCRL-53179-84, Lawrence Livermore National Laboratories, 1985.

[E] Babrauskas, V., Tu, King-Mon, "Calibration of a Burn Room for Fire Test on Furnishings", NBS-TN 981, National Bureau of Standards, 1981.

[F] Steckler, K.D., Quintiere, J.G., Rinkinen, W.J., "Flow Induced by Fire in a Compartment", NBSIR 82-2520, National Bureau of Standards, 1982.

[G] Steckler, K.D., *et al.*, "Flow Induced Flows Through Room Openings- Flow Coefficients", NBSIR 82-2801, National Bureau of Standards, 1984.

[H] Quintiere, J.G., McCaffrey, B.J., Harkleroad, M., "The Burning of Wood and Plastics in an Enclosure: Volumes 1 & 2", NBSIR 80-2054, National Bureau of Standards, 1980.

[I] Fang, J.B., "Fire Performance of Selected Residential Floor Constructions Under Room Burnout Conditions", NBSIR 80-2134, National Bureau of Standards, 1980.

[J] Cooper, L.Y., Harkleroad, M., Quintiere, J.G., Rinkinen, W.J., "An Experimental Study of Upper Hot Layer Stratification in Full-scale Multi-room Fire Scenarios", *Journal of Heat Transfer*, **104**, 1982, p 741.

[K] Fang, J.B., Breese, J.N., "Fire Development in Residential Basement Rooms", NBSIR 80-2120, National Bureau of Standards, 1980.

[L] Quintiere, J.G., McCaffrey, B.J., Den Braven, K., "Experimental and Theoretical Analysis of Quasi-steady Small Scale Enclosure Fires", *Seventeenth Symposium (International) on Combustion*, The Combustion Institute, Pittsburg, PA, 1978, p. 1125.

[M] McCaffrey, B.J., Rockett, J.A., "Static Pressure Measurements of Enclosure Fires", *Journal of Research of the National Bureau of Standards*, **82** (2), 1977.

[N] Quintiere, J.G., Steckler, K.D., Corley, D., "An Assessment of Fire Induced Flows in Compartments", *Fire Science and Technology*, **4** (1), 1984.

[O] Tanaka, T., Nakaya, I., Yoshida, M., "Full Scale Experiments for Determining Conditions to be Applied to Toxicity Tests", *First International Symposium on Fire Safety Science*, Hemisphere Pub., 1986, p.129.

[P] Croce, P.A.(ed.), "A Study of Room Fire Development: The Second Full-scale Bedroom Fire Test of the Home Fire Project", Factory Mutual Research Corp., Serial No. 21011.4, 1975.

obvious second choice is conservation of energy. However, the ideal gas law guarantees that this is conserved. Another, but arbitrary, choice is

$$B_2 = \int_{z_b}^{z_u} T(z)\, dz \qquad (10)$$

For the two layer idealization B_1 and B_2 are given by

$$B_2 = T_h (z_u - z_d) + T_c (z_d - z_b) \qquad (11)$$

$$B_1 = \frac{z_u - z_d}{T_h} + \frac{z_d - z_b}{T_c} \qquad (12)$$

Once B_1 and B_2 are determined from the measured profiles, Equations 11 and 12 can be solved simultaneously, once T_h, T_c, or z_d has been determined *a priori*. In the hot layer temperature method T_h is determined *a priori*, in the layer interface method method z_d is determined *a priori*, and in the cold layer temperature method T_c is determined *a priori*. In this work the cold layer temperature method was used because estimating the cold layer temperature is generally easier and determining T_h or z_d *a priori* amounts to direct estimation of desired quantities. However, Table 2 shows that some of the data were reduced by the original authors using other methods.

Table 2. Layer Temperature and Combustion Efficiency Determinations

Data Set	Fuel Type	Combustion Efficiency	Temperature[6] Averaging
[A]	Polyurethane	0.65[1]	CL method
[B]	Polyurethane	0.65[1]	CL method
[C]	Polyurethane	0.65[1]	CL method
[D]	Methane	1.0[1]	CL method[1]
[E]	Methane	1.0[1]	CL method
[F]	Methane	1.0[1]	HL method[1]
[G]	—	—	—
[H]	Wood	0.80[1]	CL method
	Rigid polyurethane	0.82[1]	CL method
[I]	—	—	—
[J]	Methane	1.0[1]	Arithmetic Avg.[1,2]
[K]	Furniture	1.0[1]	Arithmetic Avg.[1,2]
[L]	PMMA	0.92[3]	Single TC[4]
[M]	—	—	—
[N]	Methane	1.0[1]	CL method[1]
[O]	Propane	1.0[1]	Unspecified[5]
[P]	Polyurethane	1.0[1]	Unspecified[5]

1. The data was obtained directly from the investigator's report without any manipulation.
2. Average temperature determined from all thermocouples.
3. Drysdale, D., *Introduction to Fire Dynamics*, John Wiley & Sons, 1985, p. 172.
4. A single thermocouple was placed immediately below the ceiling.
5. No methodology was given in the the source as to how the hot gas layer temperature was determined.
6. CL - cold layer temperature method, HL - hot layer temperature method

Before using the database for evaluating existing correlations, it was necessary to evaluate the quality of the data. Of the original 909 data points extracted from 279 experiments, 350 data were removed. Several criteria were used for eliminating data points. First, any data in the first 60 seconds of the start of the fire or any data collected when the heat release rate was less than 10 kW were eliminated due to the poor definition of the hot layer under these conditions. Second, data which exceeded the adiabatic limit temperature were eliminated, since such temperatures are physically impossible. This nonphysical result can be expressed as

$$\frac{\Delta T}{\left(\frac{\dot{Q}}{\dot{m}_o c_p}\right)} > 1.0$$

where $\dot{Q}$ is the heat release rate and $\dot{m}_o$ is the compartment exhaust rate. For this evaluation it is necessary to select a vent flow expression for naturally vented fires. The choked flow expression was not chosen because it would

have eliminated 'good' data by artificially lowering the calculated adiabatic temperature due to overestimation of the flow. Instead, the flow rate was estimated as $\dot{m} = 0.08\, A_o\sqrt{H_o}$. This is a less restrictive estimate which is in reasonable agreement with the naturally vented experiments in the database where vent flows were measured and is similar to the estimate used by Foote *et al.* Most of the data eliminated by the adiabatic limit test were in data sets [K] and [L]. In addition, any data with combustible wall linings or where the temperature rise exceeded 600 K were also excluded. As previously observed by McCaffrey *et al.*[3], at temperatures above 600 K correlations of the type discussed in this paper are generally not successful. It remains unclear why these methods fail to correlate data above 600 K temperature rise.

The size and diversity of the database is summarized in Table 3. The database includes 279 experiments from which 559 data points were extracted and used. The database includes transient and steady state fires and wood, plastic, and gas burner fires. The ventilation conditions were highly dominated by single door opening experiments as one would expect given the extensive use of this configuration by researchers.

Evaluation of Existing Temperature Correlations

To evaluate the temperature correlation methods discussed in the previous section, each method was used to predict the results of the appropriate experiments in the database. A plot of the measured vs. predicted temperatures was constructed and statistical evaluations of the performance of each method were carried out. Figures 1-4 show the performance of the existing methods. The straight line shown in these figures corresponds to equality between the predicted and measured temperatures.

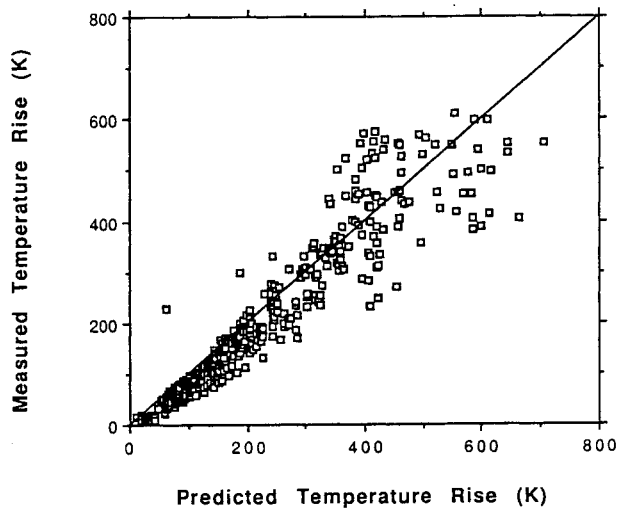

Figure 1. Comparison of measured and predicted temperature rises for the MQH correlation.

The performance of the MQH, shown in Figure 1, is somewhat less satisfactory than the original correlation, due largely to the increased diversity of the data. The standard error (standard deviation for the sampling distribution) for the MQH model is 52 K. There is a slight tendency for the predictions to exceed the measured values with the errors increasing as the temperature increases. Figure 2 shows the vari-

Table 3. Overview of the Database

Quantity	279 experiments 909 data points acquired 559 data points accepted and used
Fire Types (including only the accepted and used data)	188 transient fire data points 371 constant HRR fire data points fuels: wood, plastics, wood/plastic, methane, propane
Ventilation Conditions	460 door data points 56 window data points 7 door and window data points 36 forced ventilation data points
Compartment and Vent Sizes	Floor area- .168 to 24 m^2 Vent dimensions- 0.225 m × 0.015 m to 2.04 m × 0.78 m

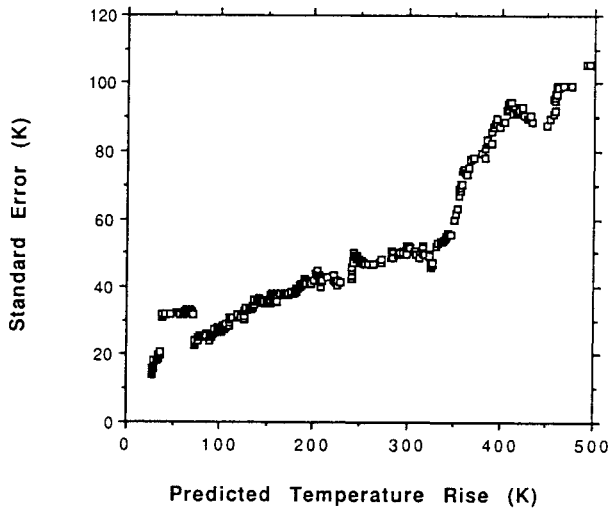

Figure 2. Variation of the standard error with predicted temperature rise for the MQH correlation. The standard error was calculated as a 51 point running standard error.

ation of the standard error with temperature. The figure was generated by calculating a running standard error including 51 data points.

The database contains 131 data points for fires against a wall or in a corner, with most of these data from experiments using line burners. The Mowrer correction for wall/corner effects actually reduced the performance of the MQH correlation, increasing the standard error by 12 K. The Mowrer and Williamson correction factors are based on point source entrainment models. As such the correction scheme would not be expected to work well with line fire sources. While the data used by Mowrer and Williamson included only 6 data points, it is difficult to understand why the correction actually deteriorates the performance of the MQH correlation.

The FPA temperature correlation, shown in Figure 3, has a standard error of 51 K. Because this method requires a known exhaust flow rate, only 181 data points could be used for the evaluation. These 181 data points include 145 naturally vented fire data points and 36 forced ventilation fire data points. Figure 3 shows some clear systematic deviations between measured and predicted temperature rises. At low temperatures, the FPA method systematically underpredicts the temperature, while at high temperatures the FPA method clearly overpredicts. It is significant that the highest measured temperature rise in a test with a measured exhaust rate is about 350°C. This shortcoming of the available data coupled with the systematic nature of the errors makes extrapolation to temperatures characteristic of flashover very risky.

Figure 4 shows the variation in the standard error with temperature rise, determined using a 25 point running standard error. The errors observed at a given predicted temperature are larger than those

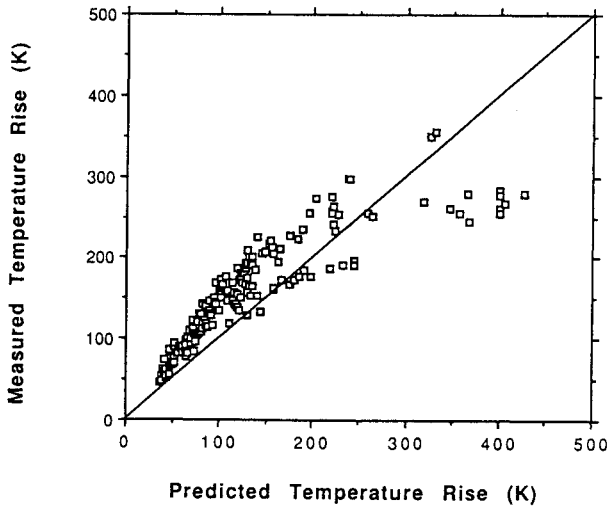

Figure 3. Comparison of measured and predicted temperature rises for the FPA correlation.

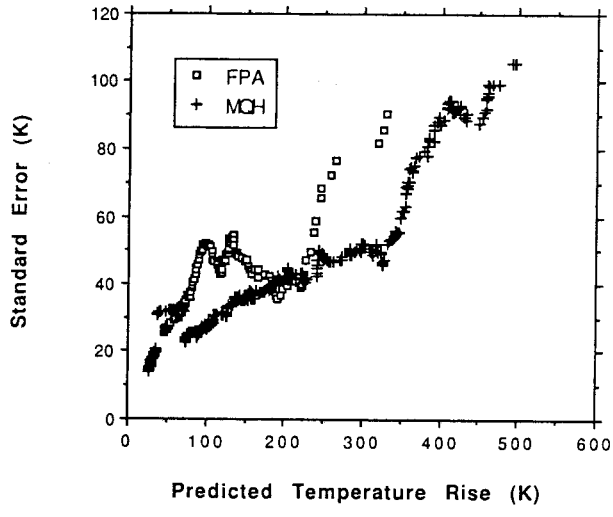

Figure 4. Variation of the standard error with predicted temperature rise for the FPA and MQH correlations. The FPA standard error was calculated as a 25 point running standard error.

observed with the MQH correlation despite the fact that these tests were better characterized in that the vent flow rate was measured.

Summary

The correlations of McCaffrey, Quintiere, Harkleroad (MQH) and of Foote, Pagni, Alvares (FPA) clearly have merit, though there are systematic errors in the predictions in the latter. The Mowrer and Williamson modification of the MQH correlation does not represent a viable method as judged by the comparisons with the database. The correction actually weakens the performance of the MQH correlation.

Formulation and Evaluation of New Models

While minor improvements in the MQH and FPA correlations could be made by adjusting the fitting constants to better fit the current database, an alternative is to develop a new correlational form more directly linked to the underlying theory. The major advantage of this approach is an increased confidence in the correlation due to its theoretical foundation. In addition, by using a theoretically based correlational form nonphysical behavior can be eliminated. For instance, in the MQH correlation as the vent area or the bounding surface area approaches zero, the temperature rise approaches infinity. This is clearly nonphysical since the estimated temperature could exceed the adiabatic flame temperature. Further, it is known that closed rooms do not give rise to the hottest fires. These nonphysical limits are not often likely to be encountered, but their existence reduces the confidence in using the correlations for rooms different from those included in the database on which the correlations were established. In order to avoid these difficulties, new candidate correlational forms will be developed in this section.

In developing new correlations it is important to remember that any model developed for correlational purposes must be highly simplified and any correlation of hot gas layer temperature should be based on an approximate energy balance for the hot layer. With this in mind, we begin with the simplified energy balance which motivated the correlation developed by McCaffrey *et al.*[3]

$$\dot{Q} = \dot{m}_o\, c_p\, (T_h - T_\infty) + h_k\, A_T\, (T_h - T_\infty) \qquad (13)$$

This simply states that the energy released in the compartment is either exhausted out the ventilation openings or is lost through the bounding surfaces of the room. The bounding surface losses are treated in a very approximate way, essentially assuming that conduction through the walls controls the energy losses. To this point the development follows McCaffrey *et al.* exactly.

Equation 13 can be rearranged to give

$$\frac{\Delta T\, \dot{m}_o\, c_p}{\dot{Q}} = \frac{1}{1 + \dfrac{h_k\, A_T}{\dot{m}_o\, c_p}} \qquad (14)$$

The following terms are defined for future use.

$$\Delta T^* \equiv \frac{\Delta T\, \dot{m}_o\, c_p}{\dot{Q}} \qquad (15)$$

$$Y^* \equiv 1 + \frac{h_k\, A_T}{\dot{m}_o\, c_p} \qquad (16)$$

Equation 14 expresses the nondimensional temperature rise as a function of the ratio of the bounding surface losses to the ventilation losses. The dimensionless groups which appear in Equation 14 were recognized by Foote *et al.*, but they did not make use of Equation 14 in their work. Equation 14 does not behave nonphysically in the limit of $A_T \rightarrow 0$ or $\dot{m}_o \rightarrow 0$. When the bounding surface losses approach zero via h_k or A_T approaching zero, the temperature approaches the adiabatic flame temperature. As the vent flow rate approaches zero, the temperature does not tend to infinity.

Measured Vent Flow Rate (Forced and Natural)

The simplest case to test Equation 14 is where the vent exhaust rate is known. In practice this occurs only in forced ventilation conditions. However, in some of the naturally vented compartment fires in the database, the exhaust rate was measured by making velocity and temperature measurements across the vent and integrating to find the exhaust rate. Figure 5 shows a log-log plot of ΔT^* as a function of Y^* for all data with known ventilation rates. The figure clearly shows that Equation 10 with heat losses modeled using Equation 7 will always underpredict the temperature rise.

Given that $\dot{Q}$ and $\dot{m}_o$ are measured, the weakest part of the model is the bounding surface heat loss model. In particular, the heat loss coefficient, h_k, is a very crude model of heat loss. Despite the crude nature of the heat loss model, the trend in nondimensional temperature rise with Y^* is well reproduced. If the heat losses were systematically decreased, the data would fit Equation 14 much better.

Recalling that the MQH wall loss model has a discontinuity in it, it might be possible to modify this model to remove the discontinuity and improve the performance of Equation 14 simultaneously. Motivated by this, the following wall heat loss model was formulated.

$$h_k = C_1 \max\left(\sqrt{\frac{k\,\rho\,c_p}{t}}, \frac{k}{\delta}\right) \qquad (17)$$

This expression switches from the transient to the steady state at a value of $t = \alpha\,\delta^2$ rather than at $t = (\alpha\,\delta^2)/4$. By using C_1=0.4, the fit of the data to Equation 14 is shown in Figure 6. The slope of the best fit to the data in Figure 6 is –1.01, nearly –1 as Equation 14 suggests. This result is most encouraging. The need for a value of C_1 less than one reflects the fact that the effective area for heat loss is less than the actual compartment area as well as the fact that the resistance to heat losses is not wholly in the solid phase of the walls.

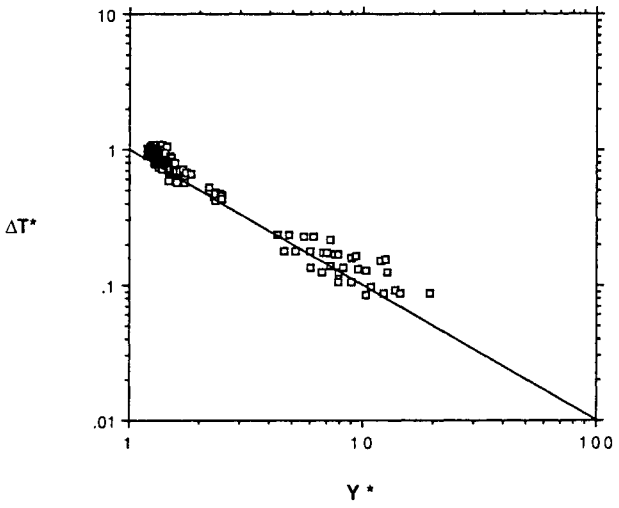

Figure 6. Correlation of the nondimensional temperature rise as a function of the nondimensional heat loss using the modified heat loss model (Equation 17) for data with measured vent flow rate. The solid line is Equation 14.

There are radiative and convective resistances to heat flow at both the inner and outer surfaces of the wall, in addition to the conductive resistance of the wall itself. It is important to note that all the data used to find the constant C_1 were for steady state heat release rate experiments. Further transient experiments with measured vent flow rates would improve the determination of the constant.

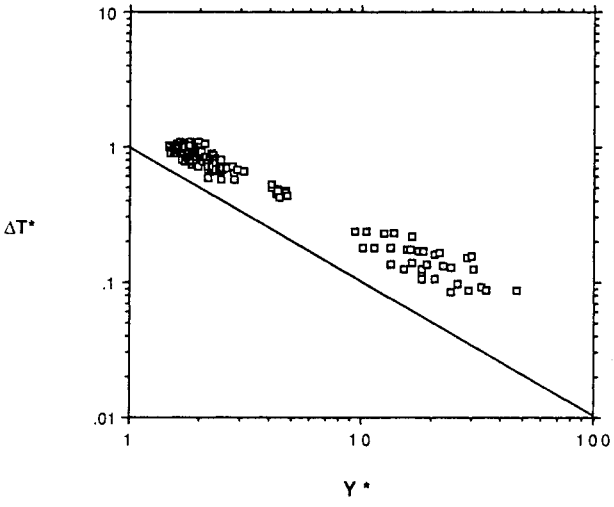

Figure 5. Correlation of the nondimensional temperature rise as a function of the nondimensional heat loss using the MQH heat loss model (Equation 7) for data with measured vent flow rates. The solid line is Equation 14.

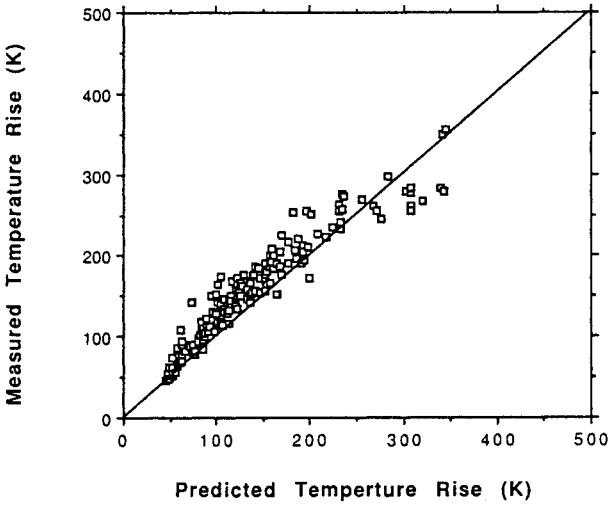

Figure 7. Comparison of measured and predicted temperature rises using Equations 14 and 17.

Figure 7 shows a comparison of predicted and measured temperature using Equations 14 and 17 with $C_1 = 0.4$. The overall performance of the model is excellent. The standard error is 29 K, much less than the 51 K standard error for the FPA method. Figure 8 shows a comparison of the 25 point moving standard error for the FPA and the energy equation method (Equations 14 and 17) for all experiments with measured vent flow rates. It should be remembered that the energy equation method has only one fitting constant, while the FPA has two fitting constants. The energy equation method better represents the data than the FPA method despite the reduced number of fitting constants.

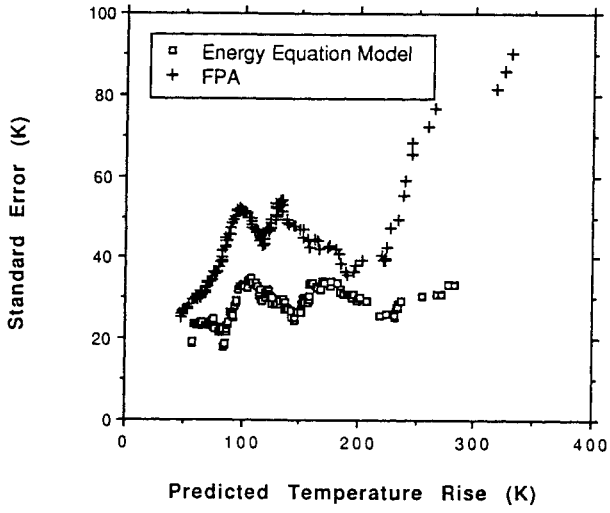

Figure 8. Comparison of the 25 point moving standard error for the FPA and the energy equation method (Equations 14 and 17) for experiments with measured vent flow rates.

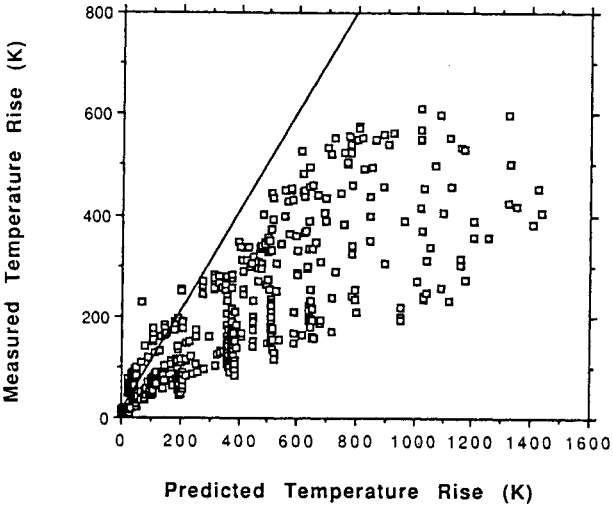

Figure 9. Comparison of the measured and predicted temperature rises using the stoichiometric model (Equations 17 and 20). The stoichiometric model evaluates the worst case ventilation condition and thereby yields a conservative estimate of the hot layer temperature.

Unspecified Ventilation Rate

The above section shows that the energy equation model, Equations 14 and 17, can be used directly to predict the hot gas layer if the vent flow rate is known. However, most often the ventilation rate is not known *a priori*. If Equation 14 and 17 are to be used, it is necessary to develop a simple model for the vent flow rate. Two flow rate models will be presented and tested: stoichiometric flow, and a layer depth driven flow model. A third model assuming choked flow was attempted, but was found to be unsuitable[9].

Stoichiometric Flow

Not only is the exhaust rate from a room most often not known, it is often unclear what the expected vent height and area will be. Will the door be open or closed, will each window break and when? These are uncertainties which arise in engineering practice that are easily avoided in fire research. Because in general the vent characteristics cannot be known, it may be prudent in practice to choose the worst case vent size. This is analogous in philosophy to ventilation pessimization used by Babrauskas for post-flashover fires[10]. Here we assume that the heat release rate from the fire is known and the ventilation is varied to maintain stoichiometric conditions which leads to the maximum hot layer temperature.

Since the heat of reaction of air, ΔH_r, is essentially constant[11] at 3 MJ/kg, the air flow rate for stoichiometric conditions is

$$\dot{m}_a = \frac{\dot{Q}}{\Delta H_r} \tag{18}$$

and the exhaust rate is

$$\dot{m}_o = \left(1 + \frac{1}{r}\right)\frac{\dot{Q}}{\Delta H_r} \tag{19}$$

where r is the stoichiometric air to fuel ratio.

Because r is generally on the order of 10 and may not be known for the fuels being burned, $1/r$ will be assumed small compared to one and will be ignored. Using $\dot{m}_o = \dot{Q}/\Delta H_r$ in Equation 14 yields

$$\frac{\Delta T\, c_p}{\Delta H_r} = \frac{1}{1 + \dfrac{h_k A_T \Delta H_r}{\dot{Q}\, c_p}} \tag{20}$$

Figure 9 shows a plot of the measured temperature rise as a function of the temperature rise predicted by the stoichiometric flow model, using Equations 20 and 17. The solid line corresponds to equality between the measured and predicted values. Clearly, the stoichiometric flow model is conservative as expected, inasmuch as the stoichiometric method evaluates the worst case ventilation, not the actual ventilation. Some of the predictions surpass the maximum temperatures observed in postflashover fires (ΔT~1200 K).

In the absence of definite knowledge of the compartment ventilation rate, the stoichiometric method gives a conservative estimate of the expected compartment fire temperature and as such may be very suitable for design situations, where uncertainties in compartment ventilation are generally unavoidable.

Layer Driven Method

When no forced ventilation is involved and the compartment openings are known it is possible to determine the vent flow rate from a knowledge of vent flow dynamics and plume entrainment. Such calculations are routinely performed in zone fire models[12] as implemented in computer models. The key here is to build on such solutions, but to develop sufficiently simple expressions that computer calculations are not required. This section will develop such a model which will be referred to as the layer driven method.

We begin our development with the vent flow equation for a two layer compartment fire as expressed by Rockett[13].

$$\dot{m}_{out} = \frac{2\sqrt{2g}}{3} C_D\, \rho_\infty A_o \sqrt{H_o} \times \sqrt{\left(\frac{T_\infty}{T_h}\left[1 - \frac{T_\infty}{T_h}\right]\right)}\,(1-N)^{3/2} \tag{21}$$

where

$\dot{m}_{out}$ = the vent discharge rate
C_D = the discharge coefficient (0.68)
ρ_∞ = the ambient density (1.2 kg/m3)
A_o = the vent area
H_o = the vent height
T_∞ = the ambient temperature
T_h = the hot layer temperature
N = (height of the neutral plane above the base of the vent)/H_o

Fortunately, for values of T_h greater than 475 K

$$\sqrt{\left(\frac{T_\infty}{T_h}\left[1 - \frac{T_\infty}{T_h}\right]\right)} \equiv 0.48 \tag{22}$$

Using this approximation we need only know the ventilation factor, $A_o\sqrt{H_o}$, and N. The value of N depends on the hot gas layer temperature and the location of the layer interface in the compartment. Rockett[13] has worked out an exact expression for N. Unfortunately, the solution is an implicit one and as such it is not convenient for use.

For purposes of this work, we desire a simple explicit approximation for N which can be directly substituted into Equation 21. The expression

$$N = 0.4 + 0.6\, D^2 \tag{23}$$

is within 10% of the exact solution as given by Rockett[13] for hot layer temperature rises less than 600 K. D is the height of the interface above the base of the vent normalized by the vent height. $D = 0$, if the interface is below the vent base, and $D = 1$, if the interface is above the top of the vent. Substituting Equations 22, 23, and the normal values for C and ρ_∞ given above, this into Equation 21 yields

$$\dot{m}_{out} = 0.54\, A_o \sqrt{H_o}\,(1 - D^2)^{3/2}\ [\mathrm{kg/s}] \tag{24}$$

Equation 24 is extremely simple, if the interface location is known. Equation 24 gives no flow if the layer is above the vent, and the well known choked flow expression usually assumed in postflashover fires if the interface is below the base of the vent.

In order to find D it is necessary to use a plume flow equation and equate the vent flow out with the air entrained by the plume. Plume entrainment has been investigated quite intensively[14], yet in the near field (within the flame itself) no

consensus has yet emerged. We will begin with the most widely used plume entrainment model.[14]

$$\dot{m}_{entr} = 0.076\,\dot{Q}^{1/3}\,Z^{5/3} \tag{25}$$

$$= 0.076\,\dot{Q}^{1/3}\,H_0^{5/3}\left(D - \frac{z_F}{H_o}\right)^{5/3} [\text{kg/s, kW,m}]$$

where

Z = the height of the hot layer interface above the fuel source

z_F = the height of the fuel source above the base of the vent

$\dot{Q}$ = the heat release rate

H_o = the vent height

$D = (z_d - z_b)/H_o$

z_d = the height of the layer interface above the floor

z_b = the height of the base of the vent above the floor

Equating the vent flow rate (Equation 24) and the entrainment rate (Equation 25) yields

$$\frac{1-D^2}{\left(D - \frac{z_F}{H_o}\right)^{10/9}} = \left(\frac{0.076\,\dot{Q}^{1/3}\,H_o^{5/3}}{0.54\,A_o\sqrt{H_o}}\right)^{2/3} \equiv \kappa \tag{26}$$

If we approximate 10/9 as 1, then Equation 22 becomes a simple quadratic and can be solved directly for D.

$$D = 0.5\left(-\kappa + \sqrt{\kappa^2 + 4\left(\kappa\frac{z_F}{H_o} + 1\right)}\right) \tag{27}$$

Recall that D is limited to the range of 0 to 1, though Equation 27 can generate values outside this range which have no physical significance. This approximate D can be used in Equation 24 to estimate the vent flow rate so that Equation 14 may be used to predict the hot layer temperature. We will refer to this overall model as the layer driven method.

Where exhaust rates have been measured and the fire is in the center of the room so that the plume model is expected to work well, we can compare the predicted and measured exhaust rates. Such a comparison is shown in Figure 10. While there are few data points for such a comparison, the results shown in Figure 10 are reasonable. In these experiments, all from data set [F], the smaller vent flow rates correspond to small vent openings in which D was nearly equal to zero. At low values of D, the vent flow rate is relatively

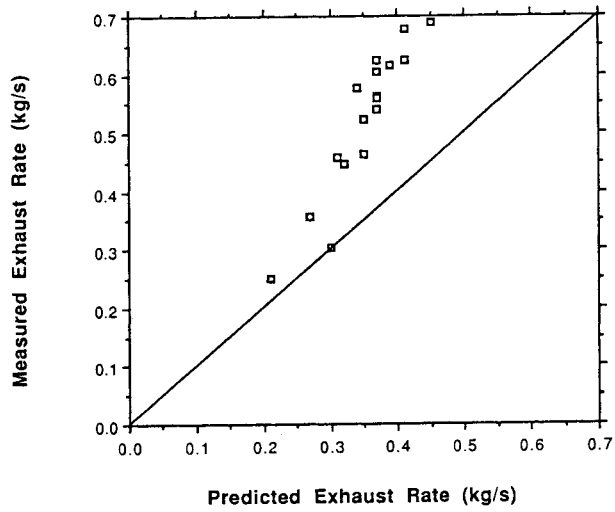

Figure 10. Comparison of the measured exhaust rates with the predictions of Equations 24 and 27 for fires not near a wall or corner.

insensitive to D and the vent flow would be well predicted even if the entrainment law was not very accurate. At the higher vent flow rates, the value of D is higher and as such the resulting vent flow rate is more sensitive to the accuracy of the entrainment law. The results shown in Figure 10 indicate that the entrainment rate has been underestimated in most cases.

The layer driven model is compared with the database in Figure 11. The standard error is

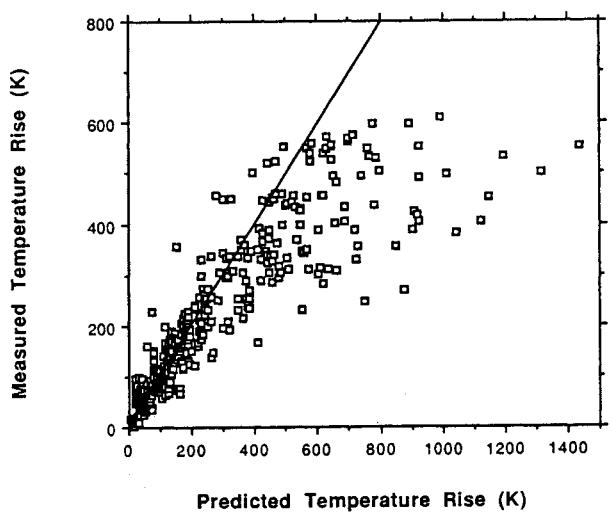

Figure 11. Comparison of measured and predicted temperatures for the layer driven model.

135 K. The largest errors for the layer driven model are the rapidly growing fires of data set [C]. This is largely the result of the heat loss modeling. The heat loss model constant C_1 was determined from experiments with measured vent flow rates, which in all cases were steady state heat release rate fire experiments. It would be possible to add a second fitting constant to improve the fit of the data in Figure 11, but this would amount to brute force fitting and will not be pursued at this time.

Figure 12 shows a comparison of the temperature dependent standard error of the layer driven and the MQH models. Over the range 0 to 500 K, the two models have very similar standard errors. It is important to note that many of the experiments had fire sources for which the plume law is not expected to work well, such as wall/corner configurations and line burners. Despite this and the crude nature of the plume entrainment model, the layer driven model has an accuracy comparable to that of the MQH model in this temperature range. However, Figure 11 shows a significant tendency of the layer driven method to overpredict the layer temperatures. Though this is conservative, the systematic nature of the deviations suggest that some problems exist with the methodology at high temperatures.

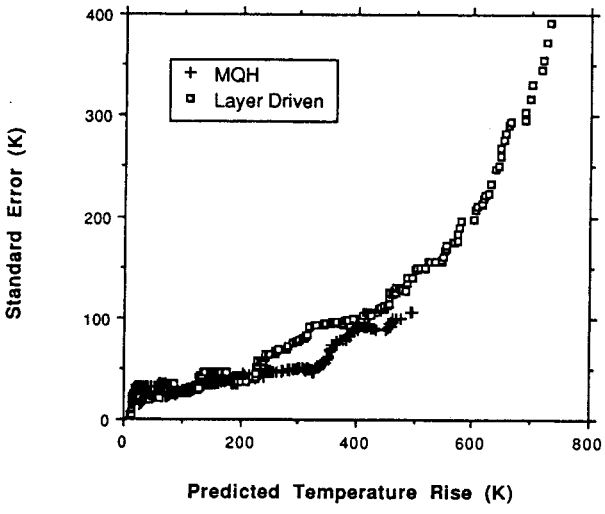

Figure 12. Comparison of the variation of the standard error with predicted temperature for the layer driven model and the MQH model. The standard error is calculated as a 51 point running standard error.

Discussion of Models

Forced Ventilation

The two methods for predicting hot gas layer temperatures of known ventilation rates are the FPA method and the energy equation method (Equations 14 and 17). Using the same set of 181 data points with measured ventilation rates, the energy equation method is a better fit to the data. The overall standard errors are shown in Table 4 and the temperature dependent standard errors are shown in Figure 8.

The energy equation method can be recommended for use when the ventilation rate is known. In practice this will involve forced ventilation fires. However, it is important to note that neither method will work well when the forced ventilation is delivered from above[15]. This type of air supply severely disrupts the two layer nature of the fire environment. Further work is required for this situation. As such, the current methods are useful for forced exhaust from the space or for forced supply of air to the lower portions of the space.

Unknown Ventilation Rate

For cases where neither the ventilation rate nor the size of the ventilation openings can be determined, the stoichiometric model provides a conservative estimate of the upper gas layer temperature. The case of unknown ventilation is a common one in design situations since the condition of windows, doors, and ventilation systems very often cannot be anticipated. As such, the conservative estimates of the stoichiometric method will be widely applicable.

Natural Ventilation

Where doors and/or windows provide the air for the fire, the MQH or layer driven models are applicable. Both methods have comparable standard errors at predicted temperature rises below 500K as shown in Figure 12. The overall standard error for the layer driven method is definitely higher than the MQH model, though the layer driven model has two less arbitrary fitting constants than the MQH model.

Both models are quite simple, though the MQH model is definitely the easiest to use and yields

Table 4. Comparison of Methods

Model	Fitting Constants	Data Points	Std. Error (K)
Known Ventilation			
Foote, Pagni, Alvares (FPA)	2	181	51
Energy Equation Method	1	181	29
Natural Ventilation			
McCaffrey, Quintiere, Harkleroad(MQH)	3	523	52
Layer Driven Method	1	523	135

more accurate results. However, the MQH model is less well grounded because it simply uses dimensionless numbers which arise from Equation 13, whereas the layer driven method uses the energy balance (Equation 13) directly. As such, the MQH model is likely to be somewhat less robust over a wide range of conditions. As noted previously, the MQH behaves inappropriately in a number of limiting cases. As our understanding of plume entrainment improves, we will be able to specialize the layer driven model to different entrainment conditions, such as against a wall, in a corner, different fire source shapes, etc. Because the entrainment law is an integral part of the layer driven method, fewer experiments should be required to validate its use in these different conditions. While the layer driven model is not yet highly refined, it has great promise for further development as our understanding of entrainment and heat losses improves. Transient experiments in which the vent flow rate is measured would contribute greatly to the improvement of the layer driven model.

CONCLUSIONS

In this investigation the existing correlational methods for predicting room fire temperatures have been evaluated by comparison with a database of 559 data points derived from a wide range of room fire experiments. In addition several new methods based on a simple energy balance have been proposed and evaluated.

For forced ventilation compartment fires, the energy equation method is shown to provide better predictions than the FPA method. However, neither method is has been tested for situations in which forced ventilation is provided in the upper portions of the room. As noted in Reference 15, ventilation from above will severely disrupt the two layer nature of the fire environment.

For single vent naturally ventilated fires, the MQH method provides better predictions that the layer driven method described here. The layer driven method is grounded more firmly in the fundamentals and as such has two fewer fitting constants. Nonetheless, it appears that our knowledge of entrainment and compartment heat losses is not sufficient to provide the layer driven method with accuracy comparable to the MQH method. Future developments in our understanding of entrainment and heat losses may allow for further development of the layer driven method. However, at present the MQH method is to be preferred for use.

In many situations there is no practical way to have prior knowledge of the ventilation rates or the ventilation openings. Under these conditions it is prudent to make conservative assumptions concerning the ventilation rate, such as assuming that the ventilation rate is exactly that required for combustion. This will lead to the highest possible compartment temperature for a given heat release rate. The stoichiometric model described in this paper embodies this assumption and has been shown to yield conservative results. This method may be very useful in conditions where the burning rate can be estimated, but the ventilation rate cannot be anticipated.

While computer fire modeling is becoming increasingly accessible, the need for simpler methods of estimating compartment fire temperatures is likely to continue for some time. Ongoing research to improve computer based models will also allow us to improve simpler methods as well.

ACKNOWLEDGEMENT

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NOMENCLATURE

A - area
B - defined by Equations 9 and 10
c_p - specific heat
C_1 - correlation constant in Equation 17
D - nondimensional layer interface height, $(z_d - z_b)/H_o$
g - acceleration due to gravity
h_k - overall heat transfer coefficient
k - thermal conductivity
$\dot{m}$ - mass flow rate
N - nondimensional neutral plane height
$\dot{Q}$ - heat release rate
r - stoichiometric air to fuel mass ratio
t - time
T - temperature
Y^* - nondimensional compartment heat loss defined by Equation 16
z - height
Z - height of the hot layer interface above the fuel source
α - thermal diffusivity
δ - wall thickness
ΔT^* - nondimensional temperature rise defined by Equation 15
κ - defined by Equation 26
ρ - density

Subscripts

a - air
b - bottom of vent
c - cold (lower) layer
d - thermal discontinuity height (layer interface height)
entr - entrainment
fo - flashover
h - hot (upper) layer
o - opening or vent, outflow
stoich - stoichiometric
T - total internal compartment surface area
u - top of vent
∞ - ambient

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