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# Thermal Actuation of Extinguishing Systems

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**U.S. DEPARTMENT OF COMMERCE, Malcolm Baldrige, *Secretary***
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# THERMAL ACTUATION OF EXTINGUISHING SYSTEMS

David D. Evans

Abstract

A brief review of the Response Time Index (RTI) method of characterizing the thermal response of commercial sprinklers and heat detectors is presented. Measured ceiling layer flow temperature and velocity histories from a bedroom fire test are used to illustrate the use of RTI in calculating sprinkler operation times. In small enclosure fires, a quiescent warm gas layer confined by the room walls may accumulate below the ceiling before sprinkler operation. The effects of this warm gas layer on the fire plume and ceiling-jet flows are accounted for by substitution of an equivalent point source fire. Relationships are given for the location and strength of the substitute source relative to a point source representation of the actual fire. Encouraging agreement was found between measured ceiling-jet temperatures from steady fires in a laboratory scale cylindrical enclosure put into dimensionless form, based on parameters of the substitute fire source, and existing empirical correlations from fire tests in large enclosures in which a quiescent warm upper gas layer does not accumulate.

Key Words: Fire models; fire plumes; fire protection; heat detectors; sprinkler systems; zone models.

## 1. INTRODUCTION

Over the past decade a great deal of research has been directed at making quantitative predictions of fire growth in enclosures. Complementary efforts to quantify and model fire suppression have also been initiated. Contributions to fire suppression and detection modeling have generally been made under conditions where results are not influenced by any surrounding enclosure. Application of this research is generally limited to large spaces or short times after fire ignition in which enclosure effects are negligible.

In modeling the suppressive action of extinguishing systems, a primary factor is the time delay between fire ignition and the start of fire suppression. In the case of typical automatic fire protection sprinkler systems, suppression begins after actuation of thermal sensing elements heated to a predetermined temperature level predominately by convection from the fire driven gas flow. These sensing elements are incorporated into the sprinkler heads that will distribute the extinguishing agent. It is common to find these sprinkler heads installed close to the ceiling of an enclosure.

Prediction of the time delay required for ceiling mounted thermal sensing elements, such as those contained in sprinkler heads or heat detectors, to respond to a growing fire has been studied by Heskestad and Delichatsios [1][1] for the special case of fires having heat release rates that increase proportionally with the second power of time from ignition. Experimental correlations of the fire driven ceiling-jet flow temperatures and velocities for steady fires have been studied by Alpert [2], Heskestad [3], and Heskestad and Delichatsios [4]. Alpert [5] also has performed a theoretical study of ceiling-jet flow.

In all of these studies, the ceiling layer flow is considered unconfined by enclosure walls, and the enclosure ambient outside the ceiling-jet flow is assumed to be uniform in temperature. This can be a useful approximation for fires in large enclosures, such as warehouse or industrial facilities, where the time to heat detector alarm or sprinkler operation may be small compared to the time to fill the upper portion of the enclosure with recirculating combustion products.

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[1] Numbers in brackets refer to the literature references listed at the end of this report.

However, for fires in enclosure sizes characteristic of residential dwellings, the accumulation of a warm upper layer of gas confined by the room walls occurs rapidly after ignition, and is a primary feature that must be accounted for in fire models. Popular two-zone room fire models, such as the Harvard Computer Fire Code [6], predict enclosure fire conditions based on a division of the enclosure into a warm upper zone and a cool lower zone. To provide useful methods to predict the time delay before operation of ceiling mounted sprinklers and heat detectors in enclosures where an accumulated warm upper layer of gas is present, existing methods must be extended to consider a two-layer ambient room environment.

This paper gives a brief description of a laboratory test method, the Plunge Test [7], used to characterize the thermal response of commercial sprinklers and heat detectors. An illustrative example is given of the use of thermal characterizations from this test to estimate sprinkler operation times in response to a specific room fire. Correlations of steady ceiling-jet flow temperatures and velocities developed by Alpert [2] and Heskestad [3] are discussed. A technique is described that reduces the fire driven plume and ceiling-jet flows in the upper portion of a two-layer ambient to the equivalent single layer flow. Temperature measurements collected in the upper portion of a laboratory scale cylindrical enclosure under steady fire conditions are used to test the technique.

## 2. CHARACTERIZING THERMAL RESPONSE OF DETECTORS

Either measurement or calculation of the fire driven gas flow temperature and velocity history in the ceiling-jet provides information on the flow field in which a sprinkler or detector is submerged and heated to operating temperature. In order to determine the time needed to raise the temperature of the sensing element from the initial temperature at fire ignition to the operating temperature, the thermal response characteristics of the hardware must be known in such a way that they may be generalized for any given gas flow history.

Heskestad and Smith [7] developed an approximate model for heating of thermal sensing elements by fire driven flows. They found that for a wide range of commercial sprinklers, the sensing elements may be considered uniform in temperature when heated by the convective flow from a fire. Net radiative effects during early fire growth were calculated to be less than 20 percent of convective heating. Based on convective heating only, the equation for changing sensing element temperature is

$$d(\Delta T_L)/dt = \tau^{-1} \ (\Delta T_g - \Delta T_L) \qquad (1)$$

where $\Delta T_L$ is the excess temperature of the sensing element above its initial temperature, and $\Delta T_g$ is the excess temperature of the gas above the initial sensing element temperature. The time constant, $\tau$, may be related to the properties of the sensing element and the convective heating of the gas flow it is exposed to according to:

$$\tau = m_L c_L / h_c A_L \qquad (2)$$

where $m_L$ is the mass, $c_L$ is the heat capacity per unit mass, $A_L$ is the area of the sensing element, and $h_c$ is the average convective heat transfer coefficient over area $A_L$. The sensing element time constant, $\tau$, is not characteristic of the hardware alone because the value of $h_c$ depends on the velocity of the gas flow.

Typical fusible sprinkler sensing elements, links, in use today contain solder material which must undergo phase transition before the sprinkler can operate. Strictly speaking, equation 1 governs the temperature response of the sensing element up to the temperature at which phase transition of the solder material begins. However, in practice, equation 1 is used to determine an effective value of the time constant for a sensing element that includes a fusion process for the solder. In application, differences between the use of an effective time constant compared to explicit accounting of the phase transition process in a more complicated model have been studied by Evans [8] and were found to be minor even for elements with large solder contents.

Effective values of the time constant, $\tau$, for commercial devices are obtained from measurement of the time to operation for a sprinkler of known initial temperature after suddenly immersed in a heated gas flow of known uniform temperature and velocity. Operation is assumed to occur at the manufacturer's indicated operating temperature. Based on determination of the link temperature history by equation 1, the value of the time constant is calculated from the solution for the known constant $\Delta T_g$, and the measured time to operation, $t_r$, as

$$\tau = -t_r/\ln\,(1 - \Delta T_{Lr}/\Delta T_g) \tag{3}$$

where $\Delta T_{Lr}$ is the manufacturer's indicated operation temperature minus the initial temperature of the sensing element at the time of immersion. A test apparatus designed to measure sprinkler time constants is described by Heskestad and Smith [7].

The measured values of the time constant, $\tau$, are dependent on gas velocity in the test apparatus. The greater the gas velocity, the larger the value of convective heat transfer coefficient and smaller the value of $\tau$, from equation 2. For blunt bodies in crossflows, over the range of interest for application to sprinkler operation, variations of the convective heat transfer coefficient are proportional to the square root of gas velocity and independent of the gas temperature [7,9]. Therefore, the product of $\tau$ with gas velocity to the half power is a quantity which compensates for variations in gas velocity used in tests to measure the time constant and is a characteristic of the hardware alone. Heskestad [10] names this product the response time index, RTI = $\tau_o U_o^{1/2}$, where the subscript, o, identifies reference test measurement conditions. The time constant at any gas velocity is calculated from the RTI using

$$\tau = RTI/U^{1/2} \tag{4}$$

Customary units for the RTI are $m^{1/2}s^{1/2}$. Values for the response time index for ordinary sprinklers range from a low of 22 $m^{1/2}s^{1/2}$ for quick operating residential sprinklers to 375 $m^{1/2}s^{1/2}$ for liquid filled bulb sprinklers [10].

Application of the response time index to the prediction of sprinkler operation times can be examined using published test data from a large scale bedroom fire test. This fire test [11] was conducted on July 11, 1973 as part of a broad fire research program conducted in part under the direction of Professor Howard Emmons. In this test a bed mattress, in a fully furnished room, was ignited by a small fire source. Among the many measurements, the gas velocity and temperature 0.08 m below the ceiling, and 1.0 m from the axis of the initial fire plume was measured in 20 second intervals after ignition. Figure 1 shows the temperature and velocity histories. Velocity measurements terminated with the failure of the sensitive fan anemometers at 9 minutes. Temperature measurements continue throughout the test.

The measured velocity and temperature histories provide the necessary information to determine the temperature history of the sensing element of a sprinkler or heat detector that would be installed just below the ceiling at the position where the measurement were taken. As an illustration, three calculated curves produced by solving equation 1 numerically using the measured gas temperature and velocity histories are shown in figure 1 for sprinklers of different sensitivity: RTI = 25, 100, 400. As discussed an RTI of 25 is characteristic of current state-of-the-art quick operating sprinklers. The curves for RTI = 100 and 400 include the range of RTI values for most industrial sprinklers [10]. Selecting a typical operating temperature of 74°C, the three sensor temperature curves may be used to estimate the time from ignition of the mattress to sprinkler operation. In this case, the time to reach 74°C for the RTI values of 25, 100 and 400 are 3.47 min, 4.27 min, and 5.77 min, respectively.

In practice, it is unusual to have measured near ceiling temperature and velocity histories available when making estimates of sprinkler or heat detector response times. It is more common to be given fire heat release rate histories and fire locations. Therefore, ceiling-jet temperature and velocities must be calculated from this information. In the remaining portion of this paper methods to perform these calculations will be examined.

## 3. CALCULATING CEILING-JET TEMPERATURES AND VELOCITIES

### 3.1. Uniform Ambient

Studies of fire driven ceiling layer flow were first performed to characterize steady conditions by Alpert [2] and Heskestad [3]. Even though steady fires are generally of limited practical interest, results of these studies may be used to provide estimates of gas flow conditions in the ceiling layer for slowly varying fires. For such quasi-steady fires, changes in heat release rate are small over the time needed to adjust the flow field to new conditions. Later, Heskestad and Delichatsios [4] studied a special class of transient fires where the convective fraction of the heat release rate increases proportionally with the time from ignition raised to a power (power-law fires). Results from selected experiments with fires that increased as the square of time from ignition have been the basis of current engineering design calculations for fire detector spacing under large area ceilings [12].

Attention in this paper is limited to discussion and use of a series of steady ceiling layer flow results to predict sprinkler and heat detector operation for slowly growing fires. From experiments and use of dimensional arguments, Heskestad [3] has provided a correlation of maximum steady temperatures and velocities for unconfined ceiling layer flow as a function of radial distance from the axis of the fire plume impinging on an unobstructed ceiling. These correlations of dimensionless maximum temperature rise,

$$\Delta T_m^* = \Delta T_m \left(c_{p,\infty}^2 \rho_\infty^2 g H^5 / T_\infty \dot{Q}_c^2\right)^{1/3} \tag{5}$$

and dimensionless maximum velocity,

$$U_m^* = U_m \left(c_{p,\infty} T_\infty \rho_\infty H / g\, \dot{Q}_c\right)^{1/3} \tag{6}$$

are shown as solid line curves in figure 2, adapted from the work of Heskestad and Delichatsios [4]. All symbols are defined in the Notation section of this paper. The correlation of dimensionless maximum velocity is presented, combined with the dimensionless maximum temperature rise. The equation for the solid line curve shown in figure 2 is:

$$U_m^*/(\Delta T_m^*)^{1/2} = 0.68\ (r/H)^{-0.63} \text{ for } (r/H > 0.3) \qquad (7)$$

Both solid line curves in figure 2 are faired fits of data obtained from fire tests with steady heat release rates in the range of 5 KW to 375 KW and ceiling heights from 1.2 m to 2.4 m [3].

Other empirical correlations for maximum temperature rise and velocity in the ceiling-jet layer have been reported by Alpert [2]. Data for these correlations was obtained from tests of a larger scale than those used by Heskestad [3] ranging in heat release rate from 670 kW to 100 MW with ceiling heights from 4.6 m to 13.7 m. The correlations obtained by Alpert [2] have been recast into dimensionless variables using an ambient air temperature of 293°K and pressure of 1.0 atmosphere. They are:

$$\Delta T_m^* = 6.18 \qquad \text{for} \quad r/H < 0.18 \qquad (8)$$

$$\Delta T_m^* = 1.97\ (r/H)^{-2/3} \qquad \text{for} \quad r/H > 0.18 \qquad (9)$$

$$U_m^* = 3.16 \qquad \text{for} \quad r/H < 0.15 \qquad (10)$$

$$U_m^* = 0.65\ (r/H)^{-5/6} \qquad \text{for} \quad r/H > 0.15 \qquad (11)$$

These values are shown as the broken-line curves in figure 2, in the same form used by Heskestad and Delichatsios [4]. For the graph of velocity correlation, equation 11 was extended to calculate $U_m^*/(\Delta T_m^*)^{1/2}$ over the range $0.133 < r/H < 0.18$.

The correlations, equations 8-11 determined from large scale tests were found to be in good agreement with theoretical predictions also performed by Alpert [5] for weakly buoyant flows. Using both top-hat and half-Gaussian distributions of temperature and velocity for steady flow in the ceiling layer, Alpert calculated temperatures, velocities, and layer depths as a function of radial position from the axis of the fire plume impinging on the ceiling.

Results from either of the steady-state correlations presented above may be used to determine gas flow temperatures and velocities in the ceiling layer where sprinklers and heat detectors are typically located. For cases of slowly growing fires where quasi-steady assumptions may be reasonable, the results of these studies may be used to estimate transient conditions and provide a basis for calculation of response time. In enclosures where fires may produce an accumulated warm layer of gas above a cool lower layer before fire detector response, results from the above studies do not apply directly. However, methods have been developed to make use of these correlations to predict conditions in the two layer ambient characteristic of small enclosure fires.

### 3.2. Two-Layer Ambient

Soon after fire starts in small enclosures a quiescent layer of warm gas accumulates in the upper portion of the room. Zukoski [13] has developed methods to estimate the filling rate of this upper layer for closed rooms. To take advantage of the research described in the previous section where the accumulation of this quiescent upper warm layer of gas is unimportant, a method needs to be found to determine the plume flow in the warm layer from conditions given for the fire in the cool lower layer.

Plume flow above fires is often characterized by the weakly buoyant flow analysis from a virtual point source as described by Morton et al. [14] and Zukoski et al. [15]. The former has been used by Alpert [5] as part of his analysis of ceiling-jet flows in a uniform ambient. In a room fire in which warm gas has accumulated above cooler gas forming a two layer ambient, weakly buoyant point source plume flow calculations are just as applicable in the lower layer containing the fire as in a totally uniform ambient. However, adjustments are needed to describe the plume flow after penetration into the upper layer.

In a study of heat transfer to ceilings, Cooper [16] was the first to propose approximations to calculate steady plume flow in the upper warm layer by formulating a substitute virtual point source fire. The location of the second source relative to the one used to calculate flow in the lower layer, depends on the upper and lower layer temperatures, fire heat release rate, and distance of the interface between the cool and warm layers above the source point for the lower layer plume flow. Other methods to approximate plume flow in the upper layer have been proposed by Sargent [17], Evans [18] and Cooper [19]. Although several of these methods have shown good agreement with presently available two-layer ambient plume flow temperature data, a recently developed method intended specifically for use in models to estimate detector and sprinkler response to growing fires will be described.

Starting with plume flow equations developed by Zukoski et al. [15] a substitute virtual point source strength and location for plume flow in the upper layer can be determined by conserving axial temperature and excess enthalpy flux for the entire plume flow across the interface between the quiescent cool lower layer and warm upper layer. Using this substitute point source for plume flow in the upper layer, the problem of flow in the two layer ambient is reduced to flow in a uniform temperature ambient with temperature equal to the upper layer. For cases where the ceiling-jet flow, generated by impingement of the fire plume on the ceiling, remains in the warm layer, the correlations (equations 8-11) when used with the substitute source apply to the prediction of sprinkler or heat detector activation in the upper layer.

Zukoski et al. [15] proposed the following equations to describe fire plume temperature and velocity distributions, which will be taken as the starting point for this analysis.

$$\Delta T/\Delta T_m = \exp\,(-\beta^2 r^2/b^2) \qquad (12)$$

$$W/W_m = \exp\,(-r^2/b^2) \qquad (13)$$

$$b = C_\ell Z \qquad (14)$$

$$\Delta T_m/T_\infty = C_T Q^{*2/3} \qquad (15)$$

$$W_m = C_w\, g^{1/2} Z^{1/2}\, Q^{*1/3} \qquad (16)$$

$$Q^* = \dot{Q}_c/(\rho_\infty c_{p,\infty} T_\infty g^{1/2} Z^{5/2}) \qquad (17)$$

Zukoski et al. [15] recommended the use of the following values for constants determined directly or indirectly from experiments:

$$C_T = 9.115, \; C_w = 3.87, \; C_\ell = 0.131, \; \beta^2 = 0.913$$

Equations 12-17 can be used directly to calculate the plume flow in the lower cool gas layer containing a fire that can be characterized using a point source of heat at some distance below the interface. At the interface adjustments must be made.

In a previous study of plume flow across abrupt changes in ambient temperature by Evans [20], it was found that sharp decreases in calculated plume temperatures could occur when substitute plume sources are determined by matching at the interface only quantities that are integrated over the entire plume flow; such as mass flux, excess enthalpy flux, and momentum flux. These abrupt changes have not been observed in experiments. For calculation of ceiling mounted sprinkler and heat detector operation times, this abrupt decrease in calculated temperature is undesirable. In applying the substitute plume technique only two properties of the plume flow from the cool lower layer arriving at the interface can be preserved in calculating the new source location and strength representing plume flow above the interface. To calculate heat detector and sprinkler response, the best choice of quantities to preserve may be maximum temperature, and excess enthalpy flux. Preserving the maximum temperature prevents dramatic changes in the temperature profile. Conserving excess enthalpy flux is equivalent to energy conservation. Preserving velocity and closely associated plume mass and momentum fluxes are not considered as important as preserving temperature elevations and excess energy flux in calculating sprinkler operation times, because thermal detectors primarily respond to temperature rise and not gas velocity. The velocity of the gas flow does affect heat transfer rates, but only proportionally with the half-power of velocity.

At the interface between the lower cool layer, (1), and upper warm layer, (2), the calculated flow from the source in the lower layer is matched to plume flow from an imagined source point totally within a warm layer such that;

$$T_{m,1} = T_{m,2} \tag{18}$$

and

$$\int_0^\infty 2\pi c_p \rho_{\infty,1} W_1 \Delta T_1 \, r dr = \int_0^\infty 2\pi c_p \rho_{\infty,2} W_2 \left[T_{\infty,2} - T_{\infty,1} + \Delta T_2\right] r dr \tag{19}$$

Equation 18 requires that the maximum plume temperatures be equal. Equation 19 requires that the excess enthalpy flux with respect to the reference temperature $T_{\infty,1}$ be equal in both layers. Specific heat capacity is assumed to be constant. Radial temperature and velocity profiles in both layers are assumed to be Gaussian. Using equations 12-17 to represent the plume flow, substitutions are made in equations 18 and 19. The resulting two equations are solved for the relationships between the strength and location of the source points relative to the interface for flow in the lower and upper layers. They are:

$$Q^*_{I,2} = \left[(1 + C_T Q^{*\,2/3}_{I,1})/\xi C_T - 1/C_T\right]^{3/2} \tag{20}$$

$$Z_{I,2} = \left[\frac{\xi\, Q^*_{I,1}\, C_T}{Q^{*\,1/3}_{I,2}\left[(\xi-1)(\beta^2+1) + \xi\, C_T\, Q^{*\,2/3}_{I,2}\right]}\right]^{2/5} Z_{I,1} \tag{21}$$

where $\xi = T_{\infty,2}/T_{\infty,1}$ and subscript I represents conditions at the interface between cool and warm quiescent layers. A schematic representation of the calculated plume flow in the two layer environment divided into its two components is shown in figure 3. In the lower layer, flow is calculated from source $Q^*_1$ in an ambient with uniform temperature $T_{\infty,1}$ for $Z_1 < Z_{I,1}$. In the upper layer, plume flow is calculated from source $Q^*_2$ in an ambient with uniform temperature $T_{\infty,2}$ for $Z_2 > Z_{I,2}$ which corresponds to positions measured from the elevation of source $Q^*_1$ as $Z_1 = Z_2 - Z_{I,2} + Z_{I,1}$.

After a substitute source for plume flow in the warm upper layer has been calculated, the correlations developed for ceiling layer flow by Alpert (equations 8-11) and Heskestad (figure 2) may be applied to calculate maximum temperatures and velocities in the ceiling-jet flow using upper layer ambient conditions to form dimensionless variables for the substitute source $Q_2^*$.

## 4. EXPERIMENT

A simple laboratory scale cylindrical enclosure was built to study the flow produced by steady axisymmetric fires. This enclosure consisted of a 1.22 m diameter, 13 mm thick ceramic fiber board circular ceiling surrounded by a 0.29 m deep clear rigid plastic curtain. This enclosure is supported by rods so that gas may flow freely out around the lower edge of the plastic curtain sidewall into the laboratory. The commercial laboratory burner that was used had a 0.0365 m diameter screened outlet and was located directly under the center of the circular ceiling and below the lower edge of the sidewall. Flow from the burner outlet would entrain laboratory air until it reached the warm layer of gas accumulated under the ceiling within the sidewall. Laboratory natural gas was burned, premixed with laboratory air to minimize flame heights. Protective screening was installed around the apparatus to help dampen room air drafts.

Data is available from a small series of tests conducted to map the radial distribution of temperatures within and just below the upper layer of gas contained by the sidewall [21]. This data may be used to determine an average temperature for the warm layer contained by the plastic sidewall and ceiling, temperatures in the fire plume before entering the warm gas layer, and the maximum temperatures in the ceiling-jet flow. In all cases, a steady fire was established for at least 20 minutes before data sampling was started.

Figure 4 shows data from four tests having different combinations of two heat release rates and three burner to ceiling heights. For this modest series of tests, the depth of the plastic sidewall was fixed at 0.29 m. Average warm layer temperatures established naturally by the plume flow from the burner impinging on the ceiling and circulating through the enclosure ranged from 21°C to 56°C above the lower layer (laboratory) ambient of 20°C.

The depth of the warm gas layer was assumed for calculation purposes to be equal to the depth of the sidewall curtain. Heat release from the burner was assumed to be totally convective and equal to the value obtained from complete combustion of the metered natural gas flow.

Virtual source locations for all fires were checked using temperature measurements on the plume axis in the laboratory ambient. For all of the 0.63 kW fires the virtual source was calculated to be within 0.01 m of the actual burner outlet. For the 1.25 kW fire, Nixon's [21] recorded temperature in the plume before entering the upper layer is labeled with large uncertainty because of signal fluctuations and is 100 K greater than expected from extrapolation of the adjacent measurement. The calculated virtual source location using this measurement was 0.04 m above the burner outlet. Using the adjacent measured value 0.04 m into the warm layer, the virtual source location was found to be at the burner outlet elevation, as in the 0.63 kW fire tests. In all calculations of ceiling layer flow, to be presented in this paper, the virtual source location was assumed to be at the burner outlet.

Using the above characterizations of the enclosure fire; upper and lower layer temperatures, the heat release rate of the burner, and upper layer depth of 0.29 m corresponding to the curtain depth, a substitute source location $Z_{I,2}$ and strength $Q^*_{I,2}$ for the flow in the upper layer were calculated using equations 20 and 21. Values of dimensional $\dot{Q}_c$ and H calculated from these substitute source parameters were used to form values of $\Delta T^*_m$ following equation 5 from the measured values of maximum temperature in the ceiling-jet, $\Delta T_m = T_m - T_{\infty,2}$ for radial positions no closer than 0.20 m to the sidewall. At distances within 0.20 m of the sidewall the ceiling-jet flow enters the ceiling-sidewall corner region which is not modeled. These dimensionless maximum excess temperatures are shown plotted vs. dimensionless radius in figure 4. Other data obtained in a 0.91 m diameter enclosure by Zukoski and Kubota [22] were reduced in the same manner as Nixon's data with the exception that the upper layer temperature was determined from the average of only one vertical traverse in the upper layer at mid-radius. These data from three experiments are shown in figure 4. For comparison, the two temperature correlations developed by Alpert [2], and Heskestad and Delichatsios [4] are also shown in figure 4.

Measurements from the laboratory scale tests ($\dot{Q}_c \approx 1$ kW and $H \approx 0.5$ m) are seen to fall close to, but at consistently higher temperatures than would be expected from the correlations obtained in much larger tests ($\dot{Q}_c \approx 1$ MW, $H \approx 10$ m). This could be simply a result of the relatively weaker turbulent mixing of the plume flow with ambient air in these laboratory scale tests compared with the much larger fires and ceiling heights used to develop the empirical correlations. Therefore, it is recommended that the existing correlations developed by Alpert [2] and Heskestad and Delichatsios [4] be used in application to fires that are larger then those used in these laboratory experiments.

## 5. DISCUSSION AND CONCLUSIONS

A method has been demonstrated to estimate maximum excess temperature and maximum velocity for steady axisymmetric ceiling-jet flow in an upper warm layer, from a fire source in a lower cool layer of a two-layer ambient. The method involves calculating an equivalent single layer source and using existing correlations. When combined with measured thermal response characteristics, such as the Response Time Index, and warm upper layer temperatures and depths from a zone fire model, this method may be a useful tool for calculating operation times of convectively heated thermally activated sprinklers and heat detectors that are often installed near ceilings. It is expected that the steady-state results may be useful in estimating conditions for slowly growing fires in enclosures where a quiescent warm gas layer can accumulate below the ceiling before operation of a thermally activated device, although limitations have not been quantified. Large scale tests with both steady and growing fires are needed to test the limitations of these calculation methods.

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# 7. NOTATION

| | |
|---|---|
| $A$ | area |
| $b$ | half width in Gaussian profile for velocity |
| $C_\ell, C_w, C_T$ | constants in equations 14-16 |
| $c$ | specific heat |
| $g$ | gravitational constant |
| $h_c$ | convective heat transfer coefficient |
| $H$ | heat source to ceiling distance |
| $m$ | mass |
| $\dot{Q}_c$ | convective fraction of heat release rate from flame |
| $Q^*$ | dimensionless heat addition parameter $\dot{Q}_c/\rho_\infty c_{p,\infty} T_\infty g^{1/2} Z^{5/2}$ |
| $r$ | radius |
| $T$ | temperature |
| $T^*$ | dimensionless temperature, $T(c_{p,\infty}{}^2 \rho_\infty{}^2 g H^5 / T_\infty \dot{Q}_c{}^2)^{1/3}$ |
| RTI | Response Time Index |
| $\Delta T$ | $T - T_\infty$ |
| $t$ | time |
| $U$ | radial velocity |
| $U^*$ | dimensionless velocity, $U(c_{p,\infty} T_\infty \rho_\infty H/g\ \dot{Q}_c)^{1/3}$ |
| $W$ | vertical velocity in the plume |
| $Z$ | distance above point source |
| $\beta$ | ratio of the Gaussian profile half widths, velocity to temperature |
| $\xi$ | ratio of upper to lower layer ambient temperatures |
| $\rho$ | density |
| $\tau$ | thermal response time constant |

Subscripts

| | |
|---|---|
| c | convective portion |
| I | interface |
| g | gas |
| L | sensing element, link |
| m | maximum value at given height for plume or given radius for ceiling-jet |

| | |
|---|---|
| o | reference values |
| p | constant pressure |
| r | of reaction or operation |
| l | lower, cool layer |
| 2 | upper, warm layer |
| $\infty$ | ambient conditions |

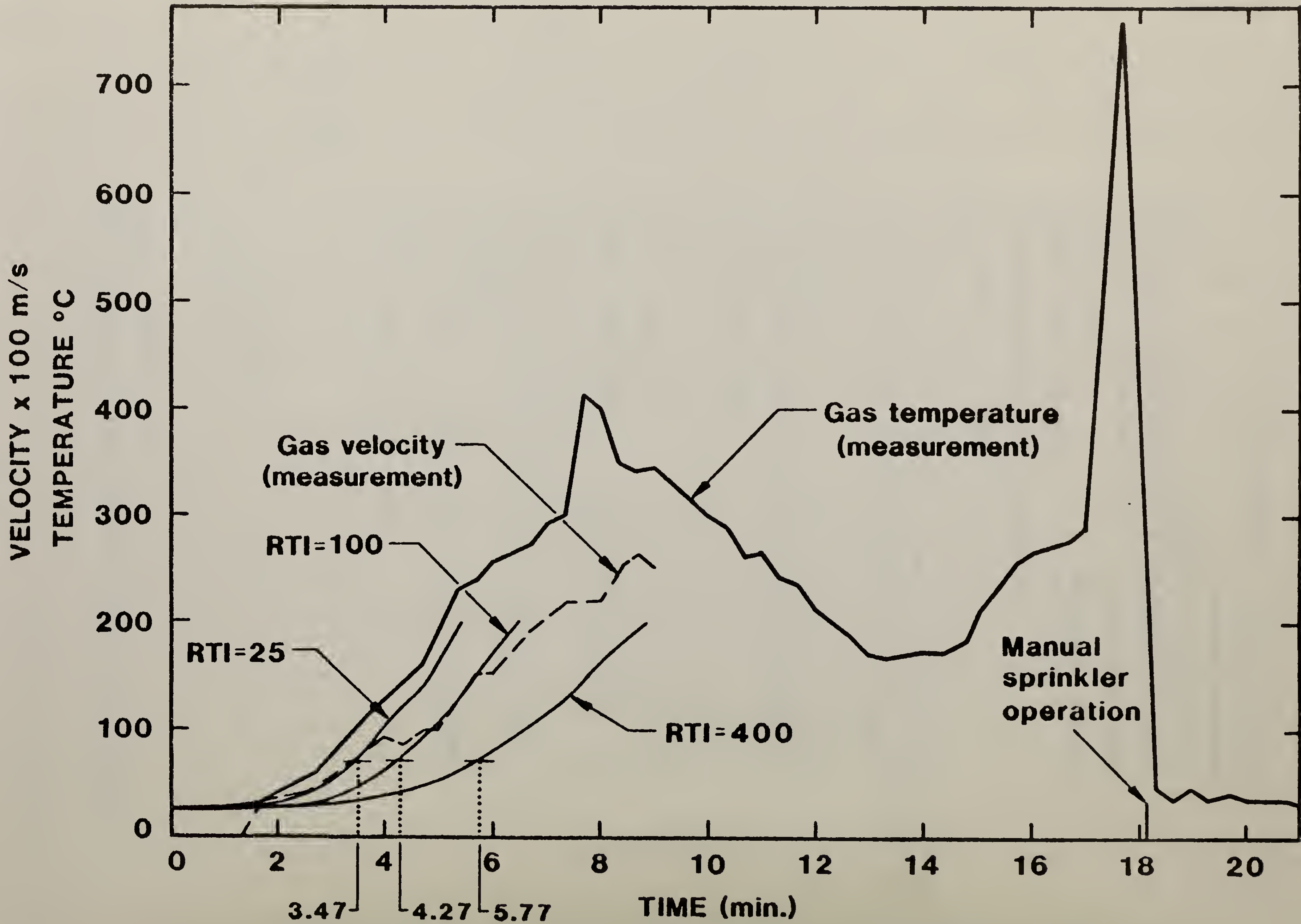


Figure 1. Calculated Temperature Rise of Sprinkler Links (RTI = 25, 100, and 400) in Response to Measured near Ceiling Temperature and Velocity Histories from the First Large Scale NSF-Rann Bedroom Fire Test July 11, 1983

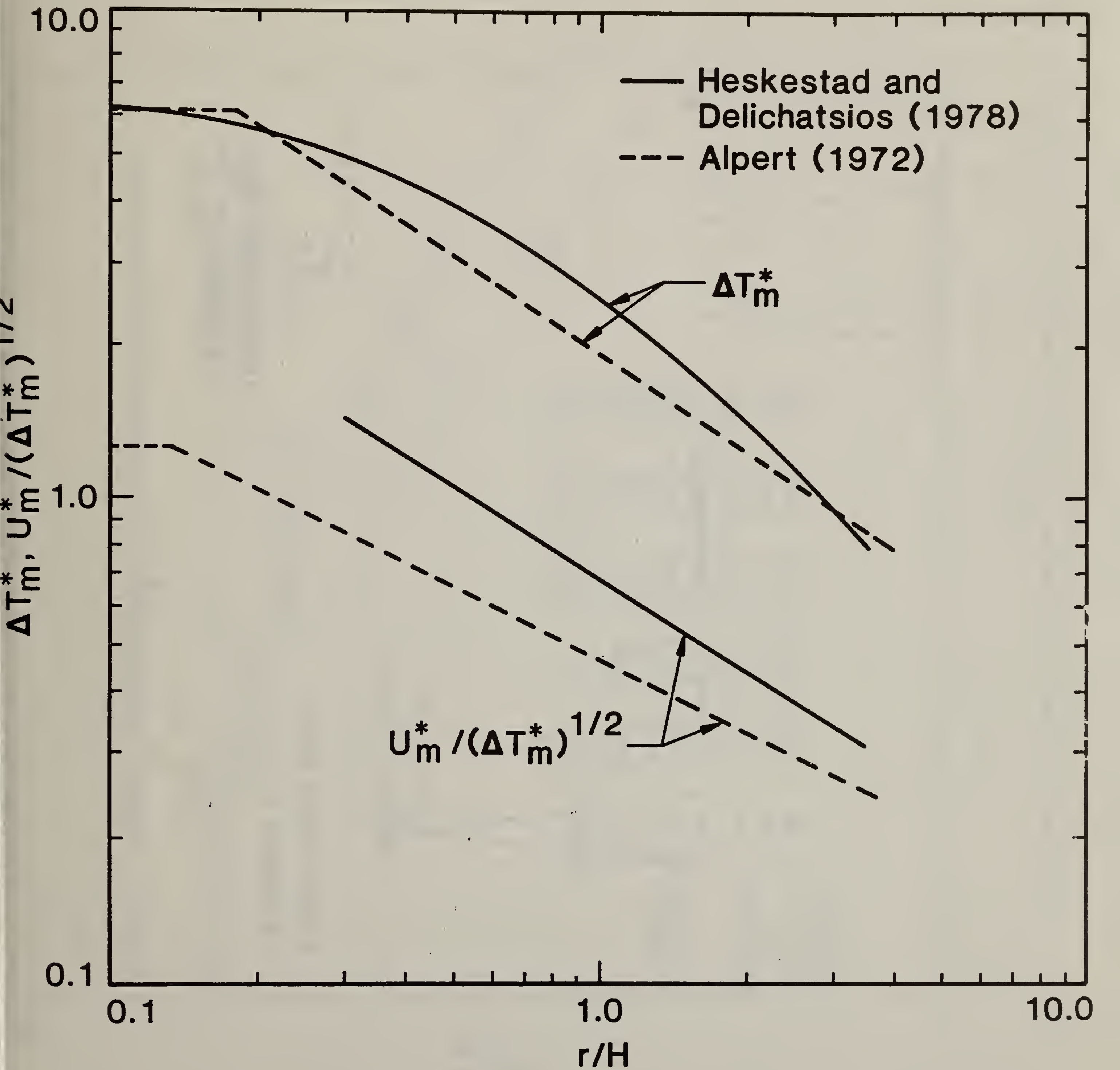


Figure 2. Dimensionless Correlations for Maximum Ceiling-Jet Temperatures and Velocities

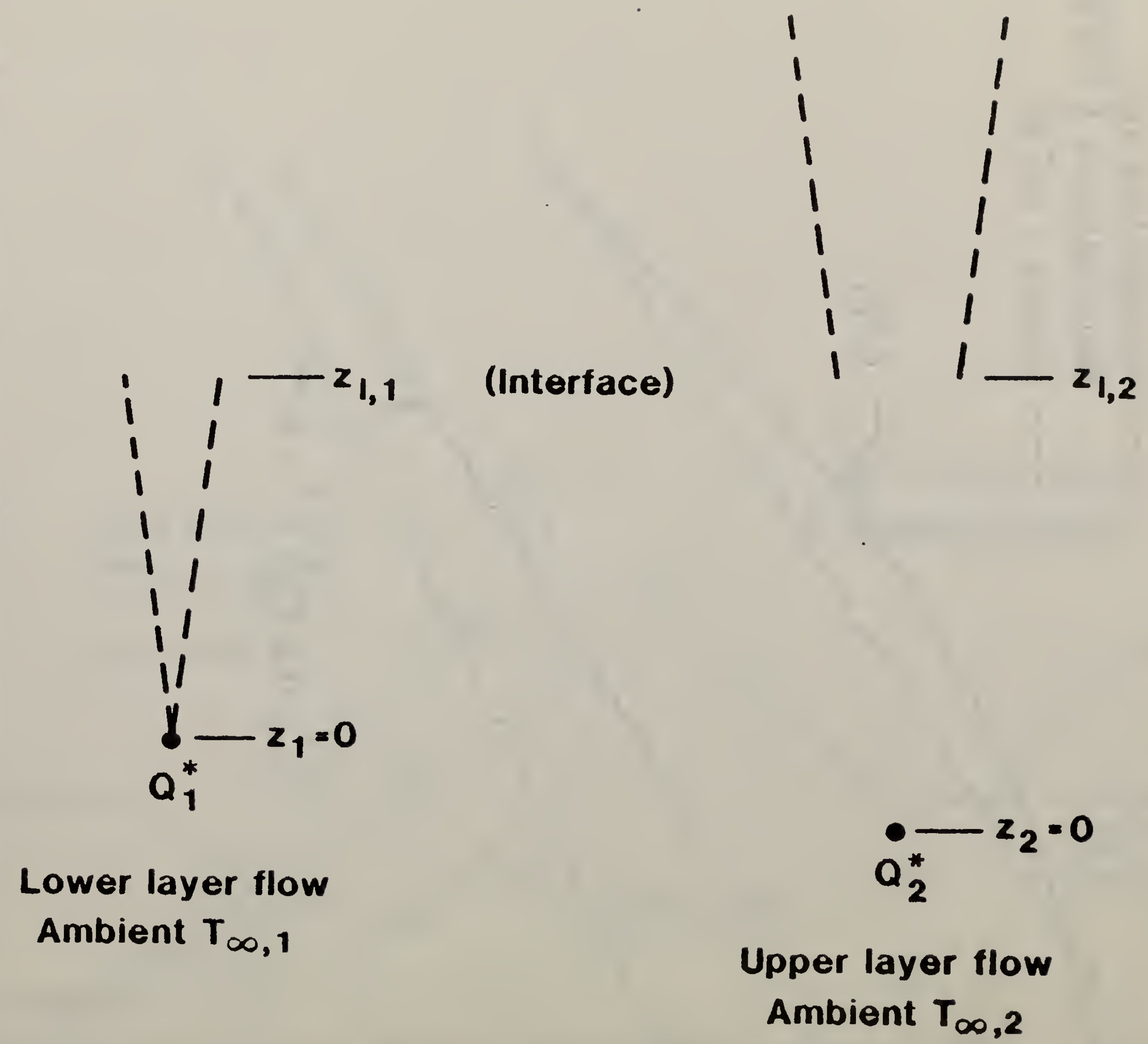


Figure 3. Schematic Representation of the Substitute Plume Calculation

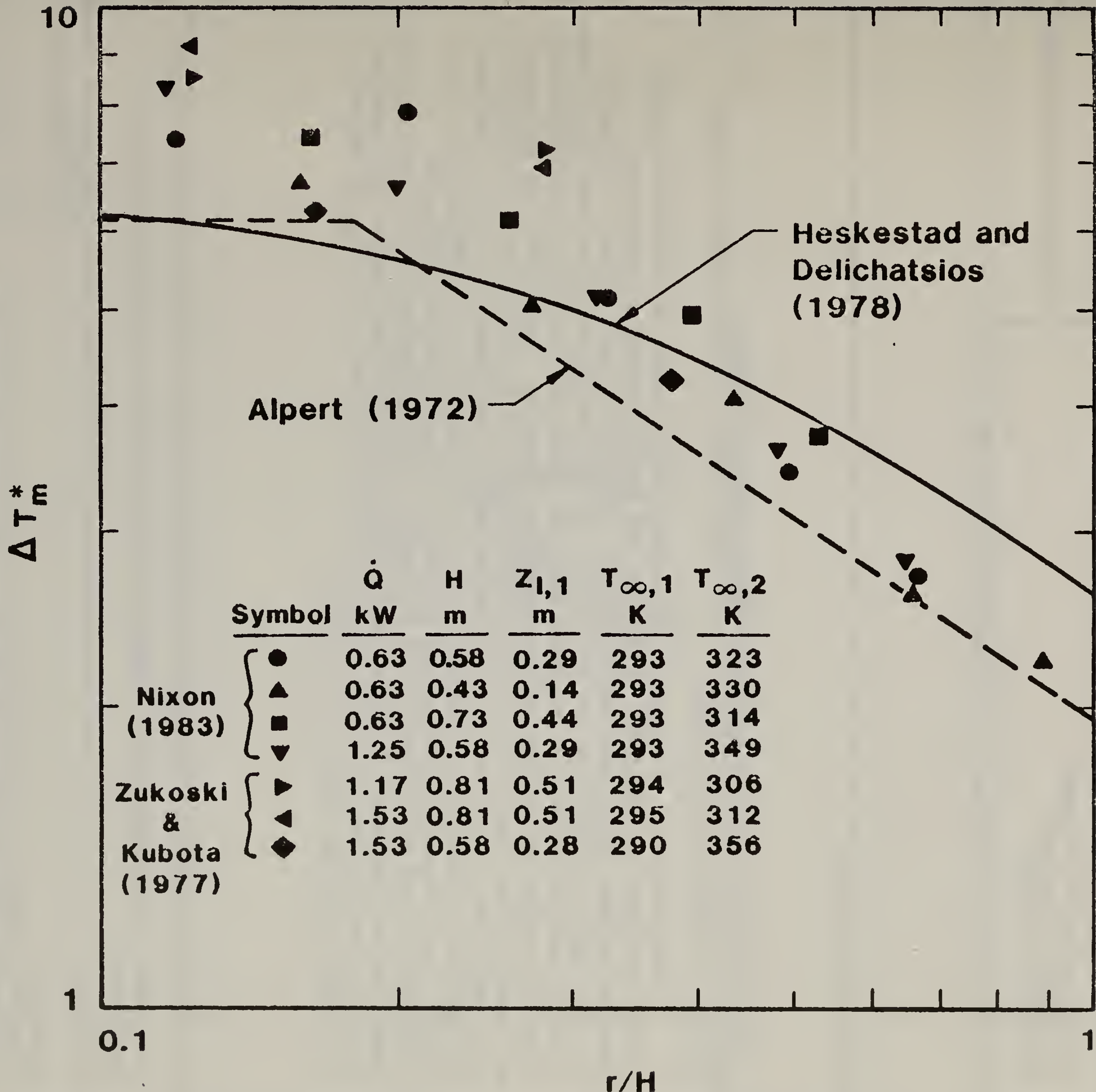


Figure 4. Comparison of Dimensionless Maximum Ceiling-Jet Temperature from Laboratory Experiment with Existing Correlations

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11. ABSTRACT (A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here)

A brief review of the Response Time Index (RTI) method of characterizing the thermal response of commercial sprinklers and heat detectors is presented. Measured ceiling layer flow temperature and velocity histories from a bedroom fire test are used to illustrate the use of RTI in calculating sprinkler operation times. In small enclosure fires, a quiescent warm gas layer confined by the room walls may accumulate below the ceiling before sprinkler operation. The effects of this warm gas layer on the fire plume and ceiling-jet flows are accounted for by substitution of an equivalent point source fire. Relationships are given for the location and strength of the substitute source relative to a point source representation of the actual fire. Encouraging agreement was found between measured ceiling-jet temperatures from steady fires in a laboratory scale cylindrical enclosure put into dimensionless form based on parameters of the substitute fire source, and existing empirical correlations from fire tests in large enclosures in which a quiescent warm upper gas layer does not accumulate.

12. KEY WORDS (Six to twelve entries; alphabetical order; capitalize only proper names; and separate key words by semicolons)

fire models; fire plumes; fire protection; heat detectors; sprinkler systems; zone models

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