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The Buoyant Plume-Driven Adiabatic Ceiling Temperature Revisited

Leonard Y. Cooper and
Anne Woodhouse (Guest Worker, Univ. of MD)

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U.S. DEPARTMENT OF COMMERCE, Malcolm Baldrige, *Secretary*
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, *Director*

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NOMENCLATURE

A, B	constants, Eq. (2)
b	nozzle-to-wall separation distance
C_p	specific heat at constant pressure
f, f_1	functions, Eqs. (7) and (A2), respectively
g	acceleration of gravity
H	plume source-to-ceiling distance
h	heat transfer coefficient, Eq. (1)
$\tilde{h}$	characteristic heat transfer coefficient, Eq. (4)
k	kinematic momentum flux of jet
Pr	Prandtl number
Q	energy release rate of fire
Q_j	enthalpy flux of jet
Q*	dimensionless Q, Eq. (4)
q	rate of heat transfer, Eq. (1)
q_c	rate of in-plate radial heat transfer
q_1	rate of heat transfer to insulation
Re_H	Reynold's number, Eq. (4)
Re_5	Reynold's number defined in [3, 5]
r	radial distance from impingement point
$r_{1/2}$	r of jet where velocity is half of axial value
T_{ad}	surface temperature for adiabatic ceiling
T_{amb}	ambient temperature
T_s	ceiling surface temperature
t	time
V	velocity profile of wall/ceiling jet
V_{max}	maximum of V

w	function, Eq. (2)
$\bar{Z}$	b divided by diameter of jet orifice
z	distance from wall
α	constant, Eq. (A4)
δ	(largest) z where $V/V_{max} = 1/2$
ε	plate emissivity
ΔE	rate of heat transfer retained in plate
ΔT^*_{ad}	dimensionless T_{ad}, Eq. (5)
λ_r	fraction of Q lost by radiation
ν	kinematic viscosity
σ	Stephan-Bolzmann constant

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Abstract

In previous works, convective heat transfer from buoyant plume-driven ceiling jets to unconfined ceilings has been estimated using a formula for the temperature distribution below an adiabatic ceiling, T_{ad}, obtained from experimental data in the range $0 \leq r/H < 0.7$ (r is the radial distance from the plume, H is the plume source-to-ceiling distance). The present study re-evaluates this data, and develops an independent estimate for T_{ad}. The analysis takes account of the effect of ceiling surface re-radiation, and use is made of the previously established similarity between plume/ceiling- and jet/wall-driven heat transfer phenomena. The latter similarity is the basis of a correlation of recently reported free jet-wall jet "recovery temperature" data into a normalized T_{ad} distribution. All of the analysis leads to new formulae for estimating the convective heat transfer to ceilings during enclosure fires. These new results confirm previous formulae, and extend them into the larger range $0 \leq r/H \leq 2.2$.

1. INTRODUCTION

The convective heat transfer to ceilings during enclosure fires can be related to the heat transfer to unconfined ceiling surfaces from buoyant plume-driven ceiling jets [1]. Results of unconfined ceiling experiments [2] suggest a relatively simple means of estimating this unconfined ceiling heat transfer rate, q, to an unconfined non-adiabatic ceiling, as

$$q = h\,(T_{ad} - T_s) \tag{1}$$

Figure 1 depicts the phenomena in question near the fire plume. In Eq. (1), T_{ad} is defined as the temperature distribution (a function of radial distance, r, from the plume-ceiling impingement point) established at the surface of an adiabatic ceiling (i.e., perfectly insulated and with zero absorbtivity and emissivity) by the gas flow from the plume of a fire of strength, Q, and effective, fire-to-ceiling distance, H, T_s is the instantaneous lower surface ceiling temperature distribution, and h is the heat transfer coefficient based on the $T_{ad} - T_s$ temperature difference. In the experiments [2] the transient thermal response of unconfined 0.00159 m thick, cold-rolled steel plate ceilings, insulated on the top side and initially at ambient temperature were measured when heated by weakly radiating premixed propane fires of constant strength between 1.17-1.53 kW (H's were between 0.58 and 0.81 m). Analysis of the data which led to the Eq. (1) formulation used long-time, "near"-ceiling gas temperatures to estimate $T_s(t \to \infty)$, which was then used as a surrogate for the T_{ad} distribution.

The original h results [2] did not distinguish between variations in the characteristic Reynold's number, Re_H, of the plume (which only varied in the narrow range $1.5(10^4) < Re_H < 2.0(10^4)$ in the laboratory experiments). In a subsequent study [3], it was noted that inertial forces compared to buoyancy forces are generally large (i.e., characteristic Richardson numbers are small) in plume-driven ceiling jets for the most interesting range of small to moderate values of r/H. For this range it was conjectured that the near-ceiling flow and heat transfer characteristics of interest could be directly related to the analogous characteristics of wall or ceiling jets driven by heated or unheated free turbulent jets (e.g., the same configuration of Figure 1, but with a turbulent jet replacing the fire and its buoyant plume). This led to a proposed correlation for h which does depend on the impingement point Reynold's number, Re_H, namely

$$h/\tilde{h} = \begin{cases} A\, Re_H^{-1/2}\ Pr^{-2/3}\ (1 + B\ r/H),\ 0 \le r/H \le 0.2; \\ \\ C\, Re_H^{-0.3}\ Pr^{-2/3}\ w(r/H),\ 0.2 \le r/H \end{cases} \tag{2}$$

where A and C are numerical constants, $B = B(Re_H)$, $h/\tilde{h}$ is a continuous function of r/H and

$$\lim_{r/H \to \infty} w \sim (r/H)^{-1.2} \tag{3}$$

In the above

$$\tilde{h} = \rho_{amb}\ C_p\ g^{1/2}\ H^{1/2}\ Q^{*1/3}$$

$$Re_H = g^{1/2}\ H^{3/2}\ Q^{*1/3}/\nu \tag{4}$$

$$Q^{*} = (1 - \lambda_r)Q/(\rho_{amb}\ C_p\ T_{amb}\ g^{1/2}\ H^{5/2})$$

where λ_r is the fraction of Q which is lost by radiation from the fire's combustion zone and from the plume itself, i.e., $(1 - \lambda_r)$ Q is the effective portion of Q which actually drives the plume's upward convection. Also, ρ_{amb}, C_p, T_{amb}, ν, and Pr are the density, specific heat at constant pressure, temperature, kinematic viscosity and Prandtl number of the ambient environment.

The above correlation was based on measured heat transfer (to an elevated-temperature wall) [5] and flow properties [10] of unheated free jet-driven wall jets (where $T_{ad} = T_{amb}$), and on the application of a criterion of equivalence for free jets and buoyant plumes at their respective sections of wall/ceiling impingement (Appendix of [3]).

With the data of [2], data from the similar "reduced-scale" fire-plume-driven heat transfer experiments of [4], and unheated, free jet-driven wall jet heat transfer measurements of [5], it was shown in [3] that with the proposed h correlation of Eq. (2), the Eq. (1) formulation would provide reasonable q estimates for Re_H at least in the range $0.73(10^4)-27(10^4)$. (Full scale hazardous fires with fire-to-ceiling distances of 2-3 m and convected energy fluxes of 200-2000 kW would have Re_H's in the range $20(10^4) < Re_H < 50(10^4)$ [1]).

In [3] the h values from the [2] and [4] experiments were derived from Eq. (1) with the use of measured values of q, T_s, and the long-time, "near"-ceiling gas temperature measurements of [2] which were assumed to be representative of T_{ad}. The h values from the free-jet experiments of [5] were derived in a similar way, but, as mentioned earlier, the values of T_{ad} in this case required no special measurements as they are identically equal to T_{amb}.

2. A RE-EVALUATION OF EXISTING DATA

The present investigation involves a re-evaluation of the data of [2] within the context of the Eq. (1) formulation. The goal is to obtain new estimates for T_{ad} and h which would 1) be consistent with an Eq. (2)-type of

Re_H-dependent h; 2) would only use the initial transient data (t = 1, 2, 3, 5 and 7 min) of [2], and not the 25-30 min, long-time, near surface gas temperatures which were previously used to represent the "true" T_{ad} [2]; 3) would include the previously neglected effect of exposed ceiling surface re-radiation to far-field surfaces; and which would 4) use recently reported data of free jet-driven wall jet heat transfer experiments to extend the r/H range of validity.

With regard to item 2) above, there are two reasons for avoiding the use of the long-time measurements in a new analysis of the data of [2]. First, there is indication from the transient data that in-plate radial conduction is important even at steady-state for r/H at least up to 0.25. Also, a non-zero plate emissivity could lead to non-negligible plate radiation to the far-field. Under either of these circumstances the steady-state plate response would differ from that of an adiabatic ceiling. Second, all the reported transient data are from thermocouples mounted directly on the plate itself, whereas the long-time temperature data in reference [2] were measured in the gas at a distance below the plate surface of 0.0016 m.

Consistent with the above remarks, another analysis of the tabulated data of [2] was carried out. The analysis involved application of energy conservation of elemental volumes of the plate, at the seven locations of plate-mounted thermocouples (placed at .07 m radial intervals from the point of plume-ceiling impingement), and at different times during the plate heating. New, dimensionless T_{ad} data were generated from the expression

$$\Delta T^*_{ad} \equiv \frac{(T_{ad} - T_{amb})}{T_{amb}\, Q^{*2/3}}$$

$$= \frac{(T_s - T_{amb})}{T_{amb}\, Q^{*2/3}} + \frac{1}{hT_{amb} Q^{*2/3}} [q_1 + q_c + \Delta E + \varepsilon\sigma(T_s^4 - T_{amb}^4)] \quad (5)$$

where σ is the Stefan-Bolzmann constant and where q_l, q_c and ΔE are the respective instantaneous rates of heat transfer per unit area of plate conducted and lost to the rear insulation, conducted radially outward, and absorbed and retained in the plate itself. The right-hand term of Eq. (5) represents the lower plate radiation to far-field ambient temperature surfaces which are taken to be at the ambient temperature, and ε is the effective plate emissivity.

A review of the data of [2] indicates that by seven minutes into the experiments a value of ε greater than a few tenths would have led to a significant contribution (on the order of 35 percent for $\varepsilon = 1$) to the net value of the bracketed term of Eq. (5). It would therefore appear that an Eq. (5)-type of analysis of the data requires an estimate for the plate's ε. Unfortunately, the literature has little to offer for ε values of cold-rolled steel plate. Indeed, while tables of the ε's of solid surfaces typically provide values of 0.6-0.8 for rolled steel plate of various descriptions, the authors found only one primary reference [6] for the ε of <u>cold</u>-rolled steel. Such an ε is reported to have been measured at the remarkably low values of 0.075 (93°C) and 0.085 (260°C). (Reference [7] reports these results of [6] incorrectly as 0.75-0.85.) To confirm this the authors had the ε of available samples of "old and uncleaned" cold-rolled steel measured at the National Bureau of Standards. A room temperature value of 0.12 was obtained [8]. As will be seen below, the new analysis of the data of [2] suggests that the "true" ε of the plate of [2] was indeed in the range of these low values.

Except for ε (assumed for now to be unknown) and h, all terms on the right hand side of Eq. (5) are available from tabular data presented in [2]. To carry out the Eq. (5) analysis the following h distribution was used

$$h/\tilde{h} = \begin{cases} 8.82\ Re_H^{-1/2}\ Pr^{-2/3}\ [1 - (5.0 - 0.284\ Re_H^{0.2})(r/H)], & 0 \le r/H < 0.2; \\ 0.283\ Re_H^{-0.3}\ Pr^{-2/3}\ (r/H)^{-1.2}\ \dfrac{(r/H - 0.0771)}{(r/H + .279)}, & 0.2 \le r/H \end{cases} \tag{6}$$

This expression, which is in the form of the correlation of Eqs. (2)-(3), was developed by using the free jet/buoyant plume equivalency relations proposed in Eqs. (A9) and (A10) of [3] and by a curve fitting of the free-jet-driven wall jet h data of [5] ($7.3(10^4) < Re_H < 27(10^4)$). Plots of the h data of [5] and the Eq. (6) correlation are presented in Figures 2 and 3 (residual standard deviations, rsd, of .87 and .04, respectively) for r/H values within ($0 \le r/H < 0.2$) and outside the stagnation zone, respectively.†

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†Note that the data of [5] were incorrectly plotted in Figures 6-8 of [3]. Instead of $Re_H^{0.3}\ Pr^{2/3}\ h/\tilde{h}$ and $Re_H^{0.5}\ Pr^{2/3}\ h/\tilde{h}$ as indicated on the ordinates of Figures 6-7 and Figure 8, respectively, the actual plots are of $Re_5^{0.3}\ Pr^{2/3}\ h/\tilde{h}$ and $Re_5^{0.5}\ Pr^{2/3}\ h/\tilde{h}$, where Re_5 is defined in [3, 5] and is approximately $Re_5 = 0.422\ Re_H$. All of the data are correctly plotted here in Figures 2 and 3.

The new ΔT^*_{ad} values were computed from Eqs. (5) and (6) and the data of Tables IV-IX of [2] for different assumed values of ε ranging from 0 to 1. Figures 4 and 5 are plots of these data for the $\varepsilon = 0$ and $\varepsilon = 1$ cases, respectively.

3. A FORMULA FOR $\Delta T^*_{ad}(r/H)$ AND AN INDEPENDENT INDICATION OF THE "TRUE" VALUE OF ε

The correspondence between the flow and heat transfer characteristics of plume-driven ceiling jets and heated or unheated free turbulent jet-driven wall jets was mentioned earlier. Relative to this correspondence which was conjectured and supported in [3], and which was introduced into the present analysis via the Eq. (6) description for h, important new experimental results have been reported recently in [9]. Using data from that work, which studied heated jet-driven wall flows, the development of the following expression for the normalized ΔT^*_{ad} distribution was presented in the Appendix

$$\frac{\Delta T^*_{ad}(r/H)}{\Delta T^*_{ad}(r/H=0)} = f(r/H) = \frac{1-1.10(r/H)^{0.8} + 0.808(r/H)^{1.6}}{1-1.10(r/H)^{0.8} + 2.20(r/H)^{1.6} + 0.690(r/H)^{2.4}} \qquad (7)$$

This result is plotted in Figure 6 together with the (equivalent) data from which it was derived.

To use an Eq. (7) correlation for the new ΔT^*_{ad} values it is necessary to determine the "best" value for the ε, $\Delta T^*_{ad}(r/H=0)$ pair. Toward this end the following procedure was carried out: 1) compute new ΔT^*_{ad} values for a given ε as discussed above; 2) obtain the corresponding $\Delta T^*_{ad}(r/H=0)$ which provides a least squares, Eq. (7) fit to the new values.

Carrying out the procedure led to the following results: for ε in the range $0 \leq \varepsilon \leq 1$, the values of $\Delta T^*_{ad}(r/H=0)$ and the rsd increase monotonically with ε over the relatively narrow ranges from 8.59 and 0.56 to 9.75 and 0.78, respectively. Thus, the present correlation is consistent with "very low" values of plate emissivity for which the smallest rds's are obtained.

The above procedure favors $\varepsilon = 0$ as the "best" or "true" value. However, in view of the above-mentioned results of [6] and [8],

$$\Delta T^*_{ad}(r/H = 0) = 8.70 \tag{8}$$

which corresponds to $\varepsilon = 0.1$ and a rsd of 0.57 would appear to be a preferred value for the normalizing factor of Eq. (7). Using Eqs. (7) and (8) leads to a new estimate for ΔT^*_{ad} which is plotted in Figure 7 along with the new, $\varepsilon = 0.1$, ΔT^*_{ad} data.

Also plotted in Figure 7 is a correlation from [3] of the measured, late-time, near-ceiling gas temperatures of [2]. As can be seen, except for small r/H values the new estimate for ΔT^*_{ad} is close to, and slightly above the measurements. Near the stagnation point, however, the measured temperatures generally suggest a plate temperature distribution with relatively large radial gradients and with an r/H = 0 value which actually exceeds the new ΔT^*_{ad}. The large gradients can be explained by the non-adiabatic, in-plate, radial conduction effect discussed earlier. However, near-ceiling gas temperatures would never have exceeded T_{ad}, and the fact that they appear to do so would seem to disqualify the Eq. (7) estimate of ΔT^*_{ad} near the stagnation point.

In [2] it was noted that "appreciable error" in the estimates of in-plate radial conduction (q_1 of Eq. (5)) "has to be expected" near r/H = 0. This fact together with the above discussion and previous remarks suggest 1) that the long-time near-surface temperature measurements of [2] continue to be the basis of the ΔT^*_{ad} estimate within the stagnation zone, albeit with some uneasiness; 2) the latter estimate be continued at r/H = 0.2 by a new Eq. (7)-type ΔT^*_{ad} estimate which would be used outside the stagnation zone; and 3) that the selection of an optimum normalizing factor to use in Eq. (7) be made without consideration of those new ΔT^*_{ad} values which correspond to the two smallest radial positions, r = 0 and r = 0.07 m, of the experiments.

The above ideas were implemented, and they resulted in the following newly recommended estimate for ΔT^*_{ad}:

$$\Delta T^*_{ad} = \begin{cases} 10.22 - 14.9\ r/H, & 0 \leq r/H \leq 0.2; \\ 8.39\ f(r/H), & 0.2 < r/H \end{cases} \tag{9}$$

where f(r/H) is defined in Eq. (7).

4. CONCLUSIONS

The new ΔT^*_{ad} estimate of Eq. (9) provides an experimentally based extension of earlier results from a maximum r/H value of 0.7 to a r/H value of 2.2. The results of the present analysis provide independent confirmation that, except for some uncertainty in the stagnation zone, the measured, long-time, near-ceiling gas temperatures of [2] accurately represent values of T_{ad}, and that ceiling re-radiation in those experiments was negligible because of very low effective values of surface emissivity. Also, the results of this paper are consistent with and provide additional support for the notion, introduced in [3], that up to moderate values of r/H there is an equivalence between the flow dynamics and heat transfer characteristics of free jet-driven wall flows and buoyant plume-driven ceiling jets.

A recommended procedure for estimating the convective heat transfer from a fire plume to an unconfined ceiling, from the point of plume impingement to moderate r/H was developed in [3]. This procedure is still recommended. However, the new results for T_{ad} and h, developed here and presented in Eqs. (6), (7) and (9) should replace the corresponding Eq. (11), (12), (17), (18) and (20) estimates of [3]. The new results should also be used to modify Eqs. (5), (6), (16) and (17) of [1] where the unconfined ceiling equations are used to estimate heat transfer to the confined ceilings of real compartment fire scenarios.

5. ACKNOWLEDGMENTS

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APPENDIX

T_{ad} for the Wall Jet Away from the Stagnation Zone

Consider a wall jet driven by a heated turbulent jet impinging on a plane adiabatic surface. Let Q_j be the constant enthalpy flux of the jet relative to T_{amb}. Assume that jet temperatures near the impingement point are close enough to T_{amb} so that, for the purpose of applying conservation of energy to the wall jet, the density field and C_p can be approximated by their ambient values. Then

$$Q_j = 2\pi r \rho_{amb} C_p V_{max} (T_{ad} - T_{amb}) \int_o^\infty \left(\frac{T - T_{amb}}{T_{ad} - T_{amb}} \right) \frac{V}{V_{max}} dz \qquad (A1)$$

where the integration is over the thickness of the wall jet at the radial position of interest. In the above, V is the distribution of the radial component of velocity and V_{max} is its maximum value. T_{ad} is the temperature distribution at the wall surface.

Let δ be the distance from the wall, at the outer portion of the wall jet, where $V/V_{max} = 1/2$. Then the experimental study of [10] for unheated jets indicates that V/V_{max} profiles at different r stations are similar in the sense that they can be closely approximated by

$$V/V_{max} = f_1(z/\delta) \text{ for } r/b > 0.75 \qquad (A2)$$

where b is the effective jet nozzle-to-wall separation distance. We assume that this latter result is applicable even for heated jets since momentum forces will typically dominate buoyancy forces in the zero to moderate r/b values of interest. Furthermore, it is reasonable to assume that $(T-T_{amb})/(T_{ad}-T_{amb})$ can also be approximated as a function of z/δ in the $r/b > 0.75$ range. Using the latter assumptions in Eq. (1) leads to

$$T_{ad} - T_{amb} = \alpha\, Q_j/(\rho_{amb} C_p V_{max} r\delta), \quad r/b > 0.75 \tag{A3}$$

$$1/\alpha = 2\pi \int_0^\infty \left(\frac{T - T_{amb}}{T_{ad} - T_{amb}}\right) \frac{V}{V_{max}}\, d\left(\frac{z}{\delta}\right) \tag{A4}$$

Besides Eq. (A2), other results in [10] of interest here are

$$V_{max}/(\sqrt{K}/b) = 1.32\ (r/b)^{-1.1}; \quad \delta/b = 0.098\ (r/b)^{0.9} \tag{A5}$$

where $\rho_{amb}K$ is the (constant) momentum flux along axial sections of the unheated jet prior to impingement.

For the case of the heated jet it is reasonable to expect that the unheated jet results of Eqs. (A5) would still be valid provided K and b are redefined. This could be done, for example, by drawing an equivalence at the jet/wall impingement point between a heated jet of interest and the unheated jets for which Eqs. (A5) are valid. This notion of equivalence was used, e.g., in [3]. There, a buoyant plume (i.e., the far field of a limiting, small-K or low-momentum jet) was taken to be equivalent to an unheated jet at respective points of wall/ceiling impingement if the total mass and momentum flux of the plume and jet were identical. Applying these criteria for equivalence led to the results in [3] that the flow of a buoyant plume-driven ceiling jet, at least up to moderate values of r/H, would simulate the flow of an unheated, free jet-driven wall jet if b and K in the results of [10] were replaced by point-heat-source-driven plume parameters as follows

$$K^{1/2} = 0.636\ g^{1/2}\ H^{3/2}\ Q^{*1/3};\ b = 1.17\ H \tag{A6}$$

Following the above, it is assumed that the idea of equivalence at impingement between the K, b pair of an unheated jet and related parameters of a heated jet has some generality. Accordingly, Eqs. (A5) are used in Eq. (A3) with the result

$$\frac{T_{ad} - T_{amb}}{Q_j/\left(\rho_{amb} C_p b\ \sqrt{K}\right)} = 7.73\ \alpha(r/b)^{-0.8};\ r/b < 0.75 \tag{A7}$$

For the case of a buoyant plume, the equivalence relationships of Eq. (A6) together with the substitution of $(1 - \lambda_r)Q$ for Q_j are used in Eq. (A7) to yield

$$\Delta T^*_{ad} = 11.8\ \alpha(r/H)^{-0.8};\ r/H > 0.88 \tag{A8}$$

A Normalized ΔT^*_{ad} Distribution

T_{ad} data for heated jet-driven wall jets have been recently acquired and plotted in Figure 9 of [9]. This plot is in the form

$$\frac{T_{ad}(r) - T_{amb}}{T_{ad}(0) - T_{amb}} = f_2\left(r/r_{1/2}\right) \tag{A9}$$

where $r_{1/2}$ is defined as the radial position of the impinging jet, at the plane of impingement, where the jet velocity, in the absence of the wall, would have dropped to 1/2 of its value on the jet axis. Plotted in this way, the data ($r/r_{1/2} < 20.6$) correlate well for all test cases where the ratio of jet orifice-to-wall-distance to jet orifice diameter, $\bar{Z}$, exceeds 1 (i.e., for jets of different Q_j and with $\bar{Z}$'s of 5, 10, and 15). Interpreting Eq. (A7) as an asymptotic result for large $r/r_{1/2}$, the following curve fit to the $\bar{Z} > 1$ data of [9] has been obtained with a residual standard deviation of 0.02

$$\frac{T_{ad}(r/r_{1/2})-T_{amb}}{T_{ad}(0)-T_{amb}} = \frac{1-0.199(r/r_{1/2})^{0.8} + 0.0258(r/r_{1/2})^{1.6}}{1-0.199(r/r_{1/2})^{0.8} + 0.0640(r/r_{1/2})^{1.6} + 0.00443\ (r/r_{1/2})^{2.4}} \tag{A10}$$

In the experiments of [9], buoyancy forces compared to inertial forces in the impinging downward-directed jet were reported to be small enough so that within jet axis distances of interest, centerline velocities varied on the order of a few percent at most when the jet was turned upward. Accordingly it is reasonable to assume that the velocity distributions (e.g., the values of $r_{1/2}$) of the jets at impingement would be closely approximated by the corresponding velocity distributions of similarly configured but unheated free jets. Equation (A9) of [3] suggests the following equivalence between the $r_{1/2}$ of an unheated jet and the H (distance from the source) of a buoyant plume

$$r_{1/2} = 0.109\ H \tag{A11}$$

Using this in Eq. (A10) suggests the normalized ΔT^*_{ad} distribution for plume-driven ceiling jets which is presented in Eq. (7). This together with the data of [6] (made equivalent with the use of Eq. (A11)) is plotted in Figure 6.

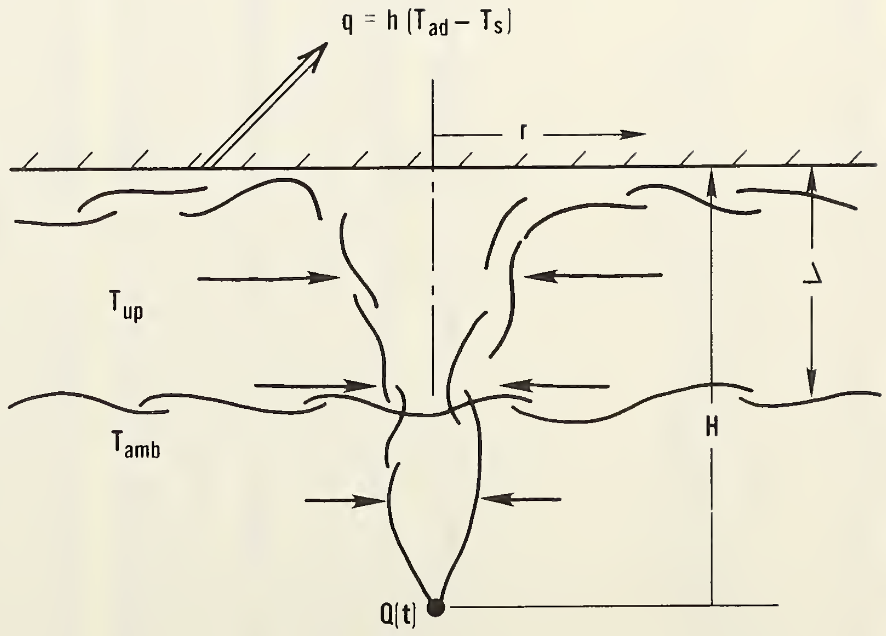

Figure 1. Heat transfer to an unconfined ceiling

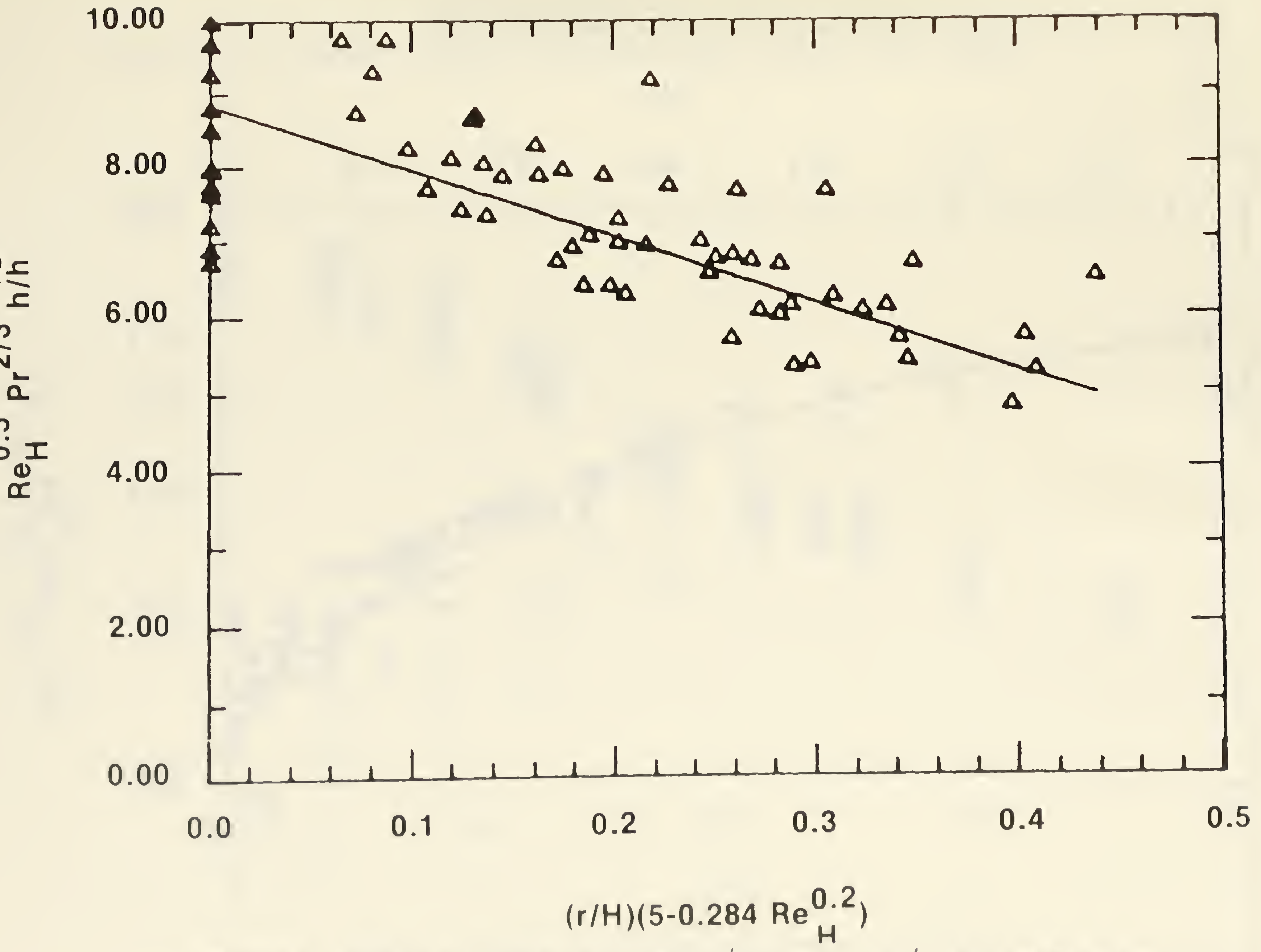

Figure 2. Plot of the h data of [5], $7.3(10^4) < Re_H < 27(10^4)$, and the estimate of Eq. (6) - in the stagnation zone, $0 \leq r/H < 0.2$

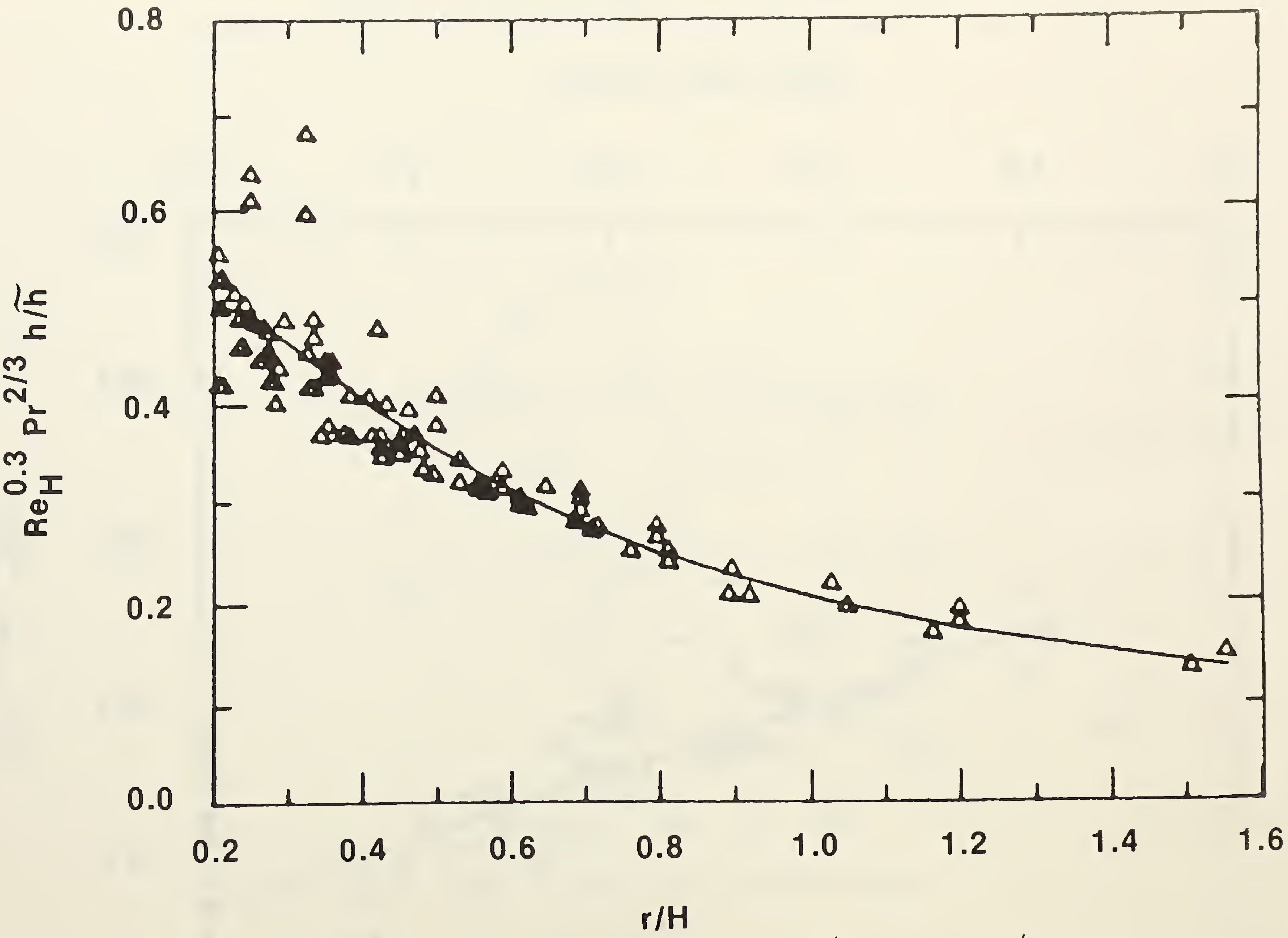

Figure 3. Plot of the h data of [5], $7.3(10^4) < Re_H < 27(10^4)$, and the estimate of Eq. (6) - outside the stagnation zone, $0.2 \leq r/H$

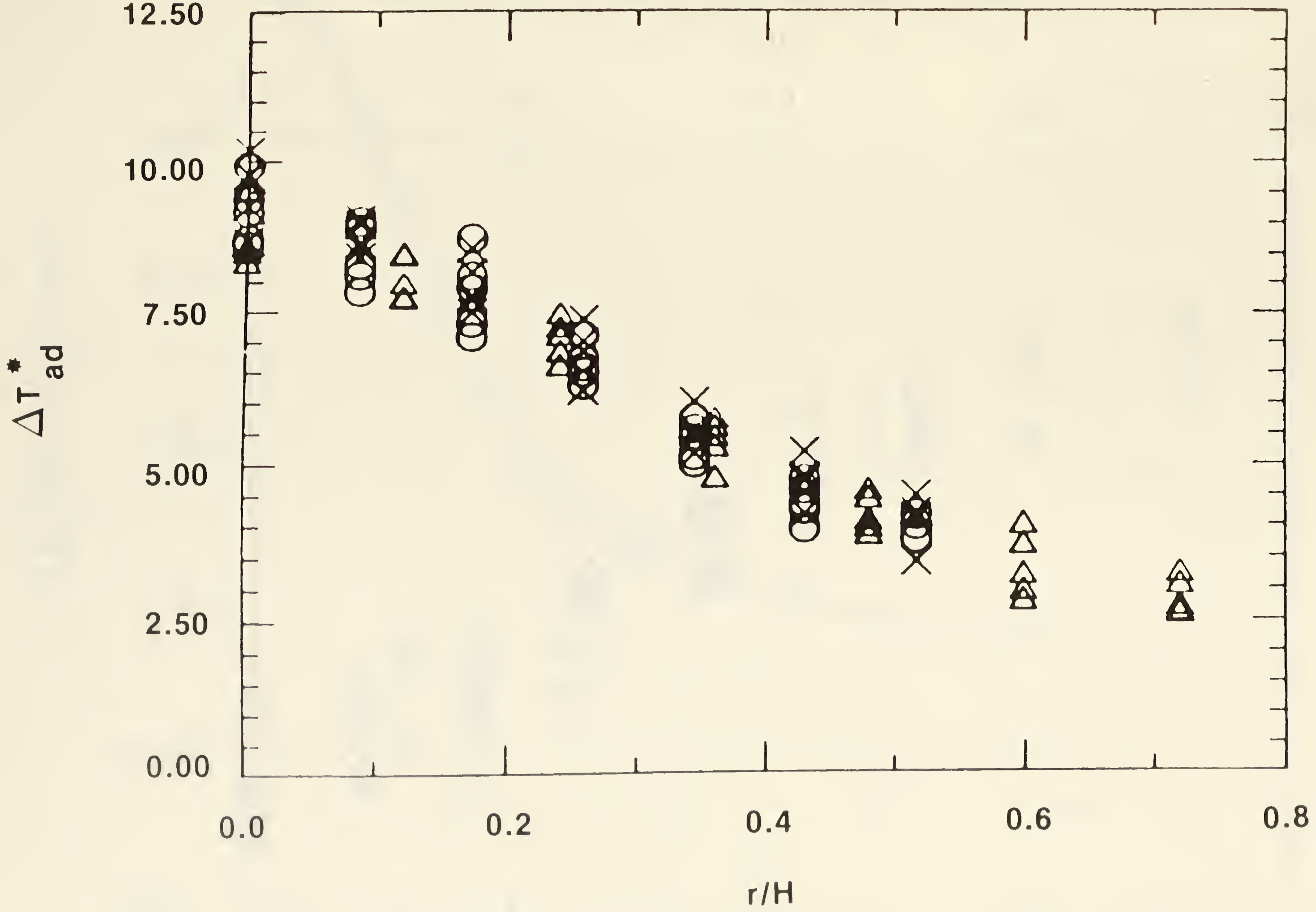

Figure 4. Plots of the new ΔT^*_{ad} data for $\varepsilon = 0$ (test 1: O, test 2: Δ, test 3: X [2])

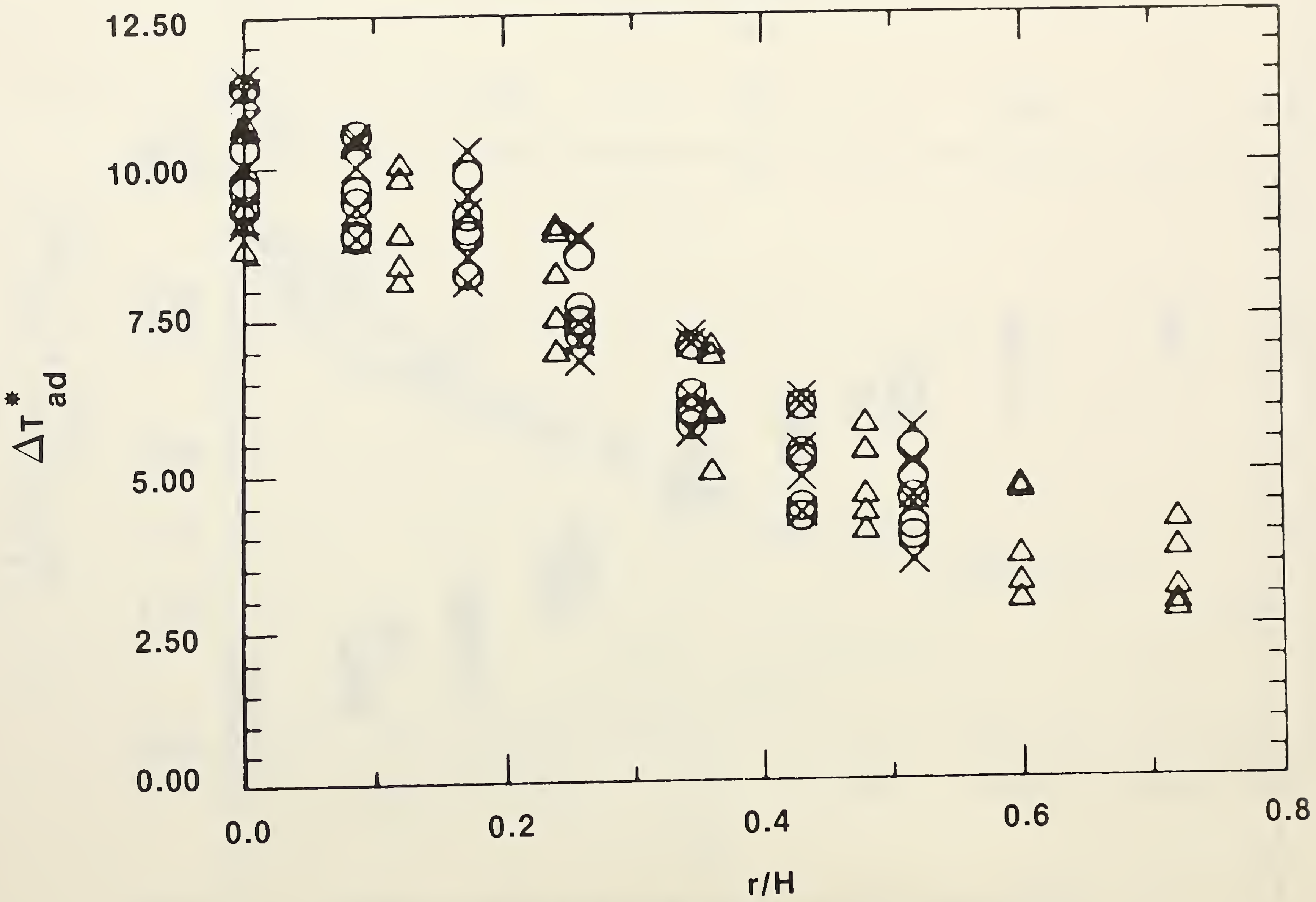

Figure 5. Plots of the new ΔT^*_{ad} data for $\varepsilon = 1$ (test 1: O, test 2: Δ, test 3: X [2])

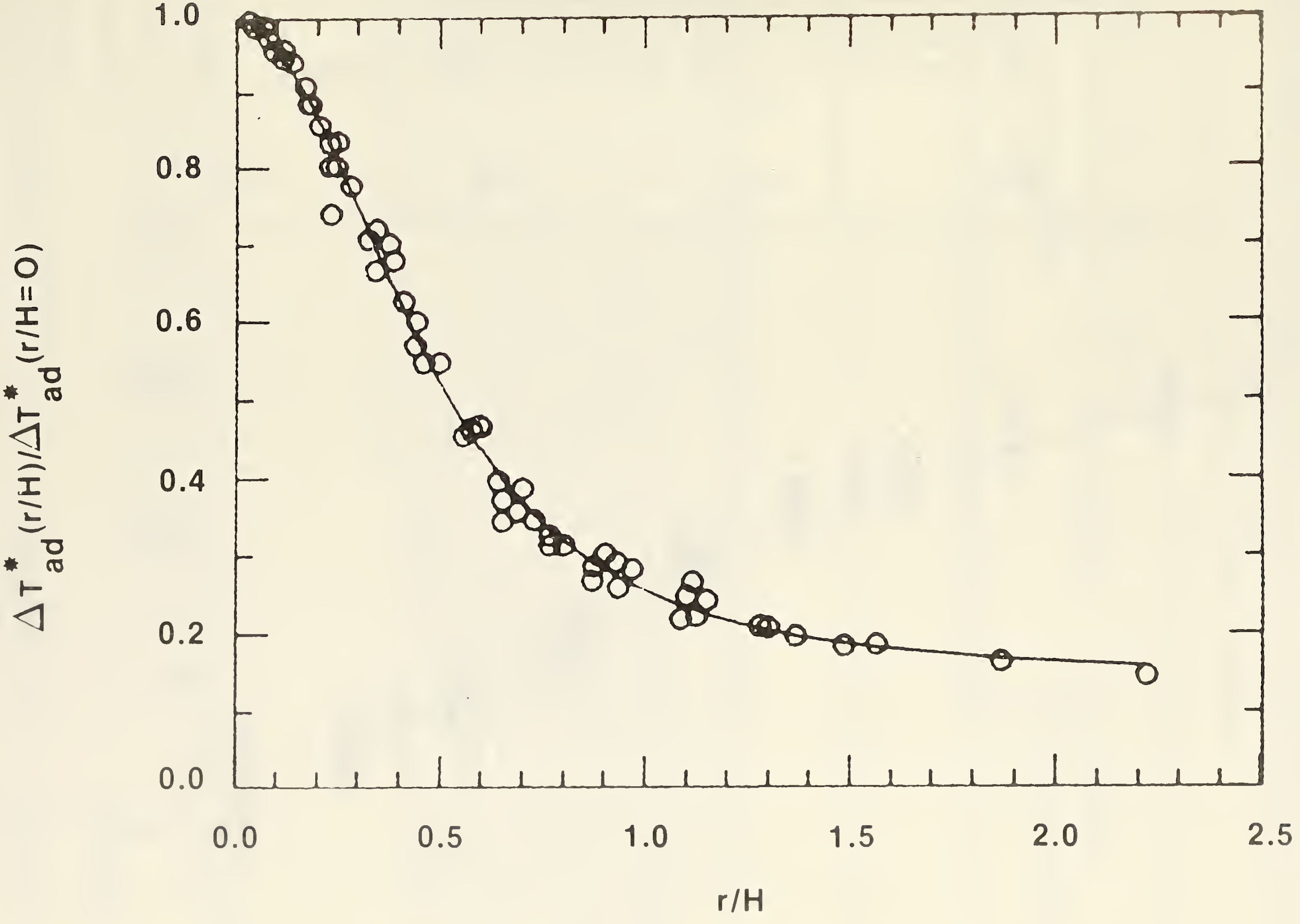

Figure 6. Plots of the normalized ΔT^*_{ad} data of [9] and the estimate of Eq. (7)

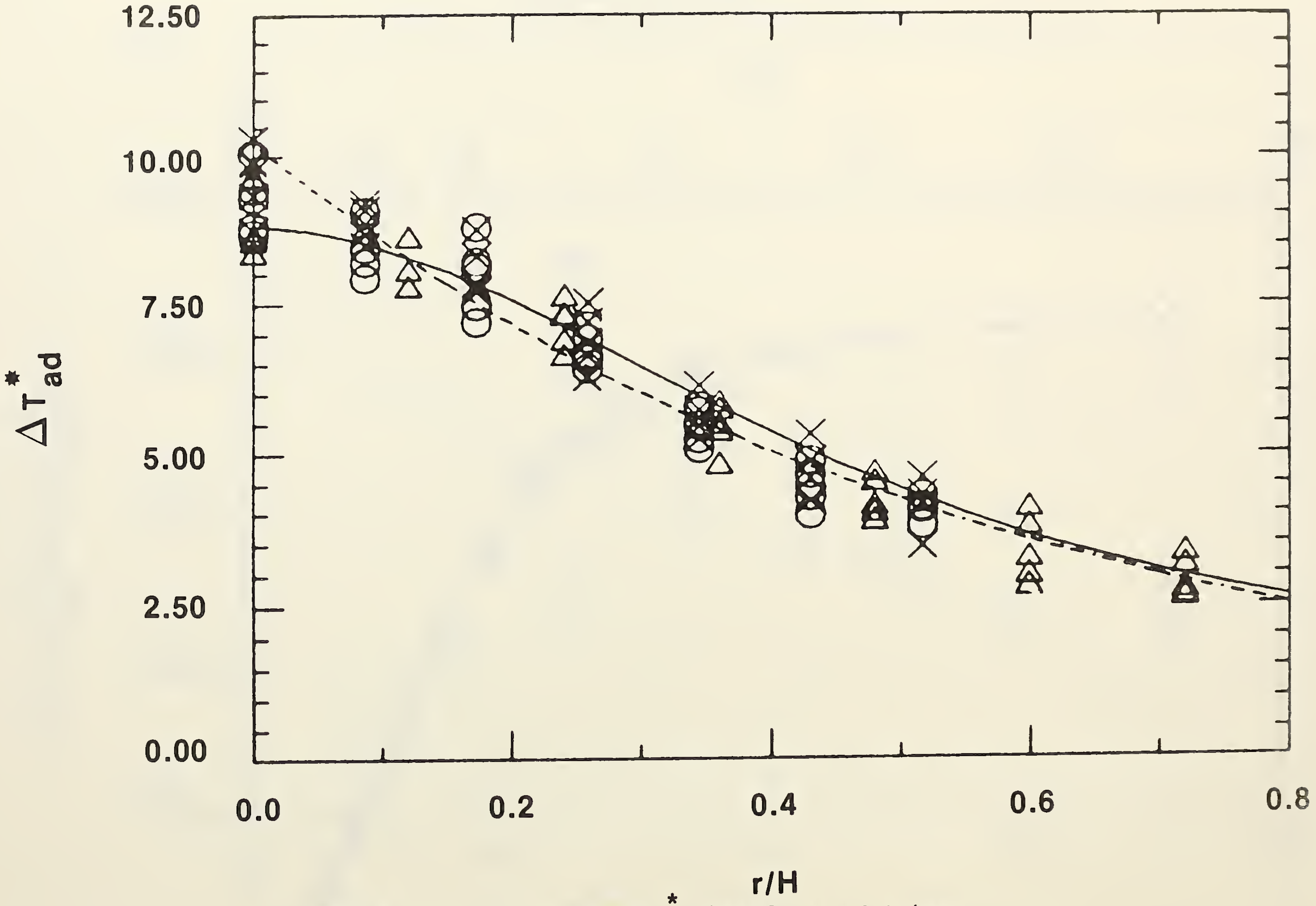

Figure 7. Plot of the new ΔT^*_{ad} data for $\epsilon = 0.1$ (test 1: O, test 2: Δ, test 3: X [2]), the ΔT^*_a estimate of Eqs. (7) and (8) (______), and the correlation (Eqs. (11) and (12) of [3]) for the measured, long-time, near-ceiling gas temperatures of [2] (---)

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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)*

In previous works, the convective heat transfer from buoyant plume-driven ceiling jets to unconfined ceilings has been estimated using a formula for the temperature distribution below an adiabatic ceiling, T_{ad}, obtained from experimental data in the range $0 \leq r/H < 0.7$ (r is the radical distance from the plume, H is the plume source-to-ceiling distance). The present study re-evaluates this data, and develops an independent estimate for T_{ad}. The analysis takes account of the effect of ceiling surface re-radiation, and use is made of the previously established similarity between plume/ceiling- and jet/wall-driven heat transfer phenomena. The latter similarity is the basis of a correlation of recently reported free jet-wall jet "recovery temperature" data into a normalized T_{ad} distribution. All of the analysis leads to new formulae for estimating the convective heat transfer to ceilings during enclosure fires. These new results confirm previous formulae, and extend them into the larger range $0 \leq r/H < 2.2$.

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

Ceilings; compartment fires; convective heat transfer; enclosure fires; fire plumes; fire modeling; heat transfer; room fires; walls

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