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NBS TECHNICAL NOTE 981

U.S. DEPARTMENT OF COMMERCE / National Bureau of Standards

The Calibration of a Burn Room for Fire Tests on Furnishings

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NO.981
1978
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Issued December 1978

National Bureau of Standards Technical Note 981
Nat. Bur. Stand. (U.S.), Tech. Note 981, 59 pages (Dec. 1978)
CODEN: NBTNAE

U.S. GOVERNMENT PRINTING OFFICE
WASHINGTON: 1978

For sale by the Superintendent of Documents, U.S. Government Printing Office, Washington, D.C. 20402
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NOMENCLATURE

A	Doorway area (m^2)
C	Discharge coefficient (-)
C_p	Heat capacity (J/kg-K)
d	Diameter (m)
D	Normalized thermal discontinuity height (-)
f	Friction factor (-)
g	Acceleration of gravity (m/s^2)
Gr	Grashof number (-)
h	Enthalpy (J)
h_c	Convective transfer coefficient (W/m^2-K)
Δh_c	Heat of combustion (J/kg)
H	Room height (m)
H_D	Doorway height (m)
k	Thermal conductivity (W/m-K)
k_e	Entrainment coefficient (-)
L	Distance (m)
m	Mass (kg)
N	Normalized neutral plane height (-)
Nu	Nusselt number (-)
P	Pressure (Pa)
Pr	Prandtl number (-)
q	Heat (J)
Q	Plume heat flow (W)
r	Radial distance (m)
R	Thermal resistance (m^2-K/W); ratio of upper to lower gas space temperatures (-)
Re	Reynolds number (-)
T	Temperature (K, °C)
u	Velocity (m/s)
Z_D	Thermal discontinuity height (m)
Z_N	Neutral plane height (m)

ε Emissivity (-)

ν Kinematic viscosity (m^2/s)

ρ Density (kg/m^3)

σ Stefan-Boltzmann constant

Superscripts

$\cdot$ Per unit time

" Per unit area

Subscripts

c Convected

f Fire gas

i Indicated; inflow

l Lower (cold) part

o Outflow

p Plume

s Surface

w Wall

∞ Ambient

THE CALIBRATION OF A BURN ROOM FOR FIRE TESTS ON FURNISHINGS

King-Mon Tu and Vytenis Babrauskas

A series of ten tests, using a diffusion flame gas burner as heat source, was conducted in a full-size room designed for furnishings flammability tests. The gas burner was used to release known heat rates and to permit steady state measurements of energy and mass flow. The gas burner fires simulated preflashover conditions, with peak temperatures outside the burner plume of around 300°C. The measurements obtained were compared with available theoretical room fire descriptions and published heat transfer values. The results showed the importance of a precise determination of the inflow and exhaust velocities at the doorway. It was demonstrated that a large number of doorway velocity probes is required to accurately obtain a room heat and mass balance. A calculational procedure was developed for analyzing the results which should be useful for future analysis of furnishings experiments.

Key words: Buoyant plumes; convection; fire tests; flammability; furnishings; heat transfer; radiation.

1. INTRODUCTION

One of the objectives of the Program for Fire Control-Furnishings at the National Bureau of Standards (NBS) is to measure the fire contribution of interior furnishings involved in room fires and to evaluate their potential hazards. Fire incidence data show that upholstered furniture and other furnishings are the major contributors to all fire deaths. Full-scale experiments are necessary to identify the sequential process of fire development. For this purpose, a burn room was constructed in which a wide variety of tests could be conducted and instrumentation was installed to measure the appropriate physical and thermal changes. A gas burner was used as a heat source to characterize and calibrate the burn room for the full-scale experiments. Temperatures, gas velocities, and heat fluxes were recorded during the tests.

The data obtained were analyzed in terms of available theory of room fires to determine the usefulness and applicability of theoretical models.

A general methodology for analysis has also been developed in the course of this study. Thus, the procedures outlined will be useful in making heat and mass flow analyses for any full-size burn-room experiments, not just for those in the present test room.

2. EXPERIMENTAL ARRANGEMENT

2.1 Exterior Environment and Ventilation

Tests were conducted in a room-corridor complex located inside the NBS Fire Test Facility Building. The large building is 57 m (187 ft) long and 26.6 m (87 ft 4 in) wide, with a ceiling height of 11 m (36 ft) in the high bay area over the test space. The general plan and section view are shown in figures 1 and 2. The heat and smoke generated in the room-corridor complex was ducted through an exhaust hood to an after-burner for smoke abatement purposes. The air needed for combustion in the burn room entered the doorways through the corridor. Air conditioning was provided in the laboratory building to maintain an average ambient temperature of 20 to 25°C, with a relative humidity of approximately 40% at the start of each test.

2.2 Test Compartment

The experiments were conducted in a full-scale burn room sized to represent a bedroom or small living room. An adjacent room along the corridor was provided for measuring and potentially controlling the flow of air. The burn-room-corridor complex is shown in figures 3 and 4. The length of the burn room was 3.52 m (11.5 ft). The adjacent room was 3.05 m (10 ft) long and the corridor was 6.10 m (20 ft) in length. The rooms were 3.40 m (11 ft 2 in) wide and the corridor was 2.44 m (8 ft) wide; they all had the same ceiling height of 2.44 m (8 ft). An exhaust hood at the end of the corridor funneled the exhaust products at low velocity to a stack containing an after-burner for smoke abatement. Doorways 2.13 m (7 ft) high and 0.91 m (3 ft) wide separated the burn room, the adjacent room, and the corridor. The rooms had a concrete slab floor. Walls and ceilings were constructed of 16 mm (nominal 5/8 inch) Type X gypsum wallboard applied on steel studs. All the surfaces of the rooms, including the floors, were lined with 13 mm cement-asbestos board (Atlas Asbestos Co. "Superbestos")[1]. The cement-asbestos boards were staggered over the gypsum wallboard to minimize air leaks and heat loss. All joints, both gypsum and cement-asbestos, were spackled.

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[1] Certain commercial equipment and materials are identified in this report in order to adequately specify the experimental procedure. In no case does such identification imply recommendation or endorsement by the National Bureau of Standards, nor does it imply that the material or equipment identified is the best available for the purpose.

2.3 Instrumentation

The room-corridor complex was instrumented with 86 thermocouples, six bidirectional velocity probes, four radiometers, and four total heat flux meters. Test data were recorded by a high speed digital data acquisition system at the rate of one scan every 10 seconds. The locations of the instrumentation are given in table 1 and shown in figures 3 and 4.

Chromel-alumel (Type K) thermocouples of 18-gage (1.02 mm) wire were used. Radiometers and total heat flux meters were Gardon-foil type water cooled units. Bidirectional velocity probes were located in the doorway to measure flows. This type of probe was developed by Heskestad [1][2] for obtaining accurate low-velocity flow measurements under fire conditions which can involve water condensation, soot deposition, and flow reversal. McCaffrey and Heskestad [2] have provided calibration techniques for these probes. The probes used were 12.7 mm in diameter, with construction details as given in the above reference. The basic equation gives that

$$\sqrt{\frac{2\Delta p/\rho}{u}} = C(Re)$$

where ΔP = measured differential pressure,

ρ = gas density (obtained from thermocouple reading adjacent to the probe and the use of the ideal gas law),

u = the gas velocity,

and $C(Re)$ = is a constant which depends on the Reynolds number. For Reynolds numbers in the range of interest the constant can be approximately taken as C = 1.08 according to the recommendation of McCaffrey and Heskestad.

Pressures were sensed with variable reluctance pressure transducers. The velocity probes, together with their associated thermocouples, were initially placed along the doorway centerline. After steady state was reached and centerline values recorded, the probes were moved to make a horizontal traverse of the doorway at 0.076 m intervals. Only half the doorway was traversed since use was made of symmetry.

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

2.4 The Gas Burner

A diffusion flame gas burner was used for this study. Since no primary air was injected, a tall, yellowish flame, somewhat typical of furnishings fires, was obtained. The burner[3] measured 0.84 m by 0.51 m by 0.46 m high and was placed on the floor in one of three positions — center, back wall, or corner — shown in figure 3. The fuel used was natural gas. Gas flow was measured by the pressure drop across a calibrated orifice. A gas burner was chosen for this calibration work because of the ease of controlling and measuring the output of the burner. It is realized, of course, that a gas burner does not model a real fire in that there is no feedback from the flame increasing the rate of fuel liberation.

3. EXPERIMENTS CONDUCTED

3.1 Experimental Plan

Ten experiments were conducted in the test room, varying the burner position and the fuel supply rate. A list of conditions studied is given in table 2. The heat release rate was obtained by multiplying the gas flow rate by the lower calorific value of the fuel gas, which averaged 34.2 MJ/m^3 (normal temperature and pressure, NTP).

3.2 Results

Since no attempt was made to model realistic fire development rates with the gas burner, only steady state results are presented. Readings were taken when it was determined that steady state was reached, some 2-3 hours after ignition. A constant temperature on the unexposed wall surfaces was used to indicate steady state.

The measurements made are given in table 3. Vertical temperature profiles at various locations, going from west to east are shown in figures 5 through 14. Within the room, air temperatures were measured in vertical groupings at six locations: at the four corners, at the room center, and midway between the center and a corner (the latter designated as "tree"). The channels plotted are the following:

outer doorway -- 70, 71, 72, 73
outer-room center -- 67, 68, 69
burn-room doorway -- 56, 57, 83, 58, 84, 59, 85, 60, 86, 61, 87, 62, 65, 63, 88
NW corner -- 55, 13, 14, 32, 15, 16, 33, 76
SW corner -- 51, 17, 18, 19, 20, 77
burn-room center -- 46, 01, 02, 03, 04
tree -- 52, 21, 22, 23, 24, 25, 26, 27, 28, 29, 74

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[3] A Thoro-Mix Model U4-43 burner, manufactured by Thoro-Mix Power-Flame Division, Parsons, Kansas, was found suitable for this purpose.

NE corner -- 53, 09, 10, 11, 12

SE corner -- 52, 05, 06, 07, 08

The profiles given in figures 5 through 14 are to be read with each labeled ordinate as starting at a base line 0°C value.

With the exception of tests with the burner in the corner position, the test geometry was nearly symmetrical about the east-west axis. The close agreement for the NW-SW readings and also for the NE-SE ones bears this out. It might be observed that in the general case the highest air temperatures were recorded at about 0.1 m below the ceiling. This peak is best visible at the tree location because of the closer thermocouple spacing there. The ceiling temperature was generally lower than the peak by 10 to 40°C. In the lower half of the room the measured temperatures were fairly uniform; the readings at the floor surface, however, were higher than in the surrounding air. This is not an error; in fact, it shows the effect of combined radiative heating and convective cooling.

4. MASS BALANCE

In five experiments where a satisfactory record of doorway velocities was obtained a mass balance for the burn room could be made. An analysis has been made for tests 102, 104, 105, 106, and 107.

4.1 Thermocouple Radiation Correction

The air densities for the doorway flows are determined from measured temperatures and the ideal gas law. The measured temperatures differ somewhat from the actual temperatures because of radiation error. An attempt was made to correct for this error.

The procedure used was as follows. Heat balance for the thermocouple bead is

$$\dot{q}''_c + \dot{q}''_r = 0$$

The sum of convective and radiative transfer rates are equal to zero since no heat is stored in steady state. The convective transfer can be expressed as

$$\dot{q}''_c = Nu \left(\frac{k}{d}\right) (T_f - T_i)$$

where Nu = Nusselt number

k = conductivity of air (W/m-K)

d = bead diameter (m)

T_f = actual air temperature (K)

T_i = indicated thermocouple temperature (K)

The Nusselt number for airflow past spheres in the velocity range of interest (Reynolds number $\leq$ 250) is given by [3]:

$$Nu = 2.0 + 0.53 \sqrt{Re}$$

And the Reynolds number is

$$Re = \frac{ud}{\nu}$$

where u = velocity (m/s)

ν = kinematic viscosity (m^2/s).

For the present experiments the bead diameter can be taken as approximately twice the wire diameter, or $d = 2 \times 10^{-3}$ m. The thermal conductivity of air, which is temperature dependent, was taken as 0.03 W/m-K for the cold air inflow and 0.04 W/m-K for the hot gas outflow. The kinematic viscosity was taken as 0.2×10^{-4} m^2/s for the inflow and 0.5×10^{-4} m^2/s for the outflow. Combining the above expressions gives

$$\dot{q}''_c = (30 + 80 \sqrt{u}) \Delta T \ (W/m^2) \qquad \underline{\text{inflow}}$$

$$= (40 + 67 \sqrt{u}) \Delta T \ (W/m^2) \qquad \underline{\text{outflow}}$$

where $\Delta T = T_f - T_i$ is the correction term.

The radiative term can be expressed in a simplified way as follows. For an environment at a uniform temperature

$$\dot{q}''_r = \varepsilon \sigma (T_w^4 - T_i^4)$$

where T_w = the bounding surface temperature

σ = Stefan-Boltzmann constant = 5.67×10^{-8} W/m^2K^4

ε = effective emissivity.

The emissivity of a metal varies widely, depending on the surface preparation. Sooty fuels had been burned in the test room prior to the present test series. It is therefore, reasonable to assume that the thermocouple surface will be covered with soot and therefore $\varepsilon \approx 1$. The wall emissivity is not well known, but is fairly high. The effective emissivity has been set to 1.0, a simplification that is reasonable in the present circumstance.

The bounding surfaces viewed by the thermocouple basically consist of the hot ceiling and the cool floor. Thus the radiation term becomes

$$\dot{q}''_r = 5.7 \times 10^{-8} \frac{(T_t^4 - T_i^4) + (T_b^4 - T_i^4)}{2} \quad (W/m^2)$$

where the subscripts t and b denote ceiling and floor, respectively. Combining all terms gives

$$\Delta T = \frac{9.5 \times 10^{-10}}{1 + 2.7 \sqrt{u}} (2T_i^4 - T_t^4 - T_b^4) \qquad \underline{\text{inflow}}$$

$$= \frac{7.1 \times 10^{-10}}{1 + 1.7 \sqrt{u}} (2T_i^4 - T_t^4 - T_b^4) \qquad \underline{\text{outflow}}$$

The results are given in tables 4 through 8. The corrected temperatures for the outflow portion of the doorway averaged about 10°C higher than measured (range: 3-29°C), while the inflow temperatures averaged about 20°C lower than measured (range: 11-28°C). It can thus be seen that part of the reason why the measured doorway temperature distribution is not square-edged, as might be expected from theoretical consideration [4], is because of the radiation error. The corrected profile is not completely square-edged, however, even though the correction appears to slightly over-correct. (Ambient temperature for this series of tests was approximately 20°C; those corrected inflow values that are lower than 20°C can be presumed to be over-corrected.)

4.2 Measured Mass Flows

The computed mass flows are given in tables 4 through 8, where a record of horizontal velocity distributions is also given. Figure 15 illustrates a typical velocity distribution. Readings are reported at five heights in the doorway. Measurements from velocity probe 112 have not been used since it appears that they were systematically depressed. The differences for the five tests range between 7% and 25% of the mean, with a median of 8%. The cause for the major deviation of 25% for test 107 is not obvious. In general, the precision of mass flow measurements in this test series was limited by the small number of velocity probes. It would be desirable in future tests to use at least a total of 10 probes. The thermocouple radiation corrections did help to reduce the imbalance, but it is clear that two or three probes for inflow or outflow are insufficient to provide a complete and accurate flow mapping.

4.3 Theoretical Mass Flows

A theory for steady state flow through compartment openings has been developed by Rockett [4]. The flow depends on the gas temperatures and opening dimensions and is calculable from Bernoulli's equation. The air in the compartment away from the opening is assumed to be at ambient temperature below a certain height, Z_D, and at a uniform elevated temperature above Z_D.

The neutral (zero flow) plane at the opening is at a different height, Z_N, which must be greater than Z_D. The flow is known when the Z_N is specified. Conservation of mass relates heights Z_N and Z_D. In the present case fuel flow is only ≈ 0.3% of the doorway flow and can be neglected in comparison. The governing relationship is then

$$\left(\frac{1-N}{N}\right)^3 = \left(1 - \frac{D}{N}\right)\left(1 + \frac{D}{2N}\right)^2 R$$

where $N = Z_N/H_D$ = normalized neutral plane height

$D = Z_D/H_D$ = normalized thermal discontinuity height

H_D = doorway height (m)

$R = T_f/T_1$ = ratio of absolute upper to lower gas temperatures

and the flow equation is

$$\dot{m} = C\,\frac{2}{3}\,\rho_\infty A\,\sqrt{H_D}\,\sqrt{2g}\left(\frac{1}{R}\left[1-\frac{1}{R}\right]\right)^{1/2}(1-N)^{3/2}$$

where $\dot{m}$ = mass flow (kg/s)

C = discharge coefficient

ρ_∞ = ambient air density (kg/m^3)

A = doorway area (m^2)

g = acceleration of gravity (m/s^2)

The flows in and out are approximately equal, differing only by the fuel flow amount. To solve for flows from basic principles a third relationship would be needed that gives either N or D. In the present series measured values of N are readily available for five of the experiments. A value of D can be calculated from them and checked for reasonableness. It is hard, however, in the absence of measurement of N to deduce it from D. As is readily seen in figures 4-14 the vertical temperature distributions, while definitely indicating hot upper and cold lower zones, are not sharp enough to accurately assign a thermal discontinuity height.

Values for R, N, D, and $\dot{m}/C$ are given in table 9. By dividing measured flows by the theoretical $\dot{m}/C$ a value for the discharge coefficient is computed. The results show outflow coefficients C_o ranging from 0.58 to 0.75 and inflow coefficients from 0.39 to 0.60. The average C_o is 0.66 and the average C_i, discounting the 0.39 value due to poor mass balance, is 0.56. Literature values [5] give for Re > 5000 (based on opening height), $C_o \simeq C_i \simeq 0.68$. The present C_o value of 0.66 agrees well, but the C_i value is somewhat low. The experiments on which the 0.68 recommended value was based were all done at lower doorway Re numbers than the present room burns ($Re \simeq 4\text{-}6 \times 10^4$). To the extent that there was a measured difference, however, the experiments of reference 5 did show $C_i < C_o$.

4.4 Plume Flow

According to simple compartment fire theory [4] the gases from the hot upper layer and the cold lower layer are assumed not to mix along their interface except at the source plume. It is assumed that under steady state conditions all of the compartment outflow has come in through the plume, and therefore the outflow should equal the plume flow, at the height Z_D, where it intercepts the hot gas layer. The plume flow equation, as given by Rockett [4], is:

$$\dot{m}_p = \dot{m}_f \left\{ 1 + \omega \frac{\rho_o}{\rho_f} \left[\left(\frac{4}{5} k_e^{4/5} \left(\frac{5}{12} \frac{(1-\omega)}{\omega^3} \pi^2 g \rho_f^2 \right)^{1/5} \left(\frac{Z_D}{\dot{m}_f^{2/5}} \right) + 1 \right)^{5/2} - 1 \right] \right\}$$

where subscript f refers to the source (burner) conditions; ω is a constant, for the fuel and is equal to 0.091 for methane. The height Z_D should properly be corrected by subtracting the burner distance above the floor and adding a virtual source distance to account for finite burner mouth area. Since the latter is not well known and since the corrections are of opposite sign, neither correction has been applied. The equation can be solved for the only unknown factor, k_e, the entrainment coefficient. The results are given in table 10.

For those cases where the burner is located near a room boundary, a circularly symmetrical flow does not result and a modified analysis must be made. When the burner is located against a wall, a construction using an image burner can be made. The wall situation represents one-half of the flow that would result from a burner of twice the flow rate in a room double the size. Thus, $\dot{m}_p$ is obtained by inserting into the above equation an $\dot{m}_f$ equal to twice the actual value and then multiplying the calculated $\dot{m}_p$ by 0.5. The practical effect of this is that only the $\dot{m}_f^{2/5}$ term is modified. The values of k_e in table 10 have been calculated on this basis. A similar procedure, using three image burners, is used for analyzing the corner position flows.

When the above procedure is followed, the calculated k_e should be independent of burner placement. From the results in table 10, it can be concluded that a value of $k_e = 0.56$ is indicated by the data. Tests 102 and 106, however, give differing results, for which no explanation is available. A value of 0.56 is, of course, considerably higher than the $k_e \simeq 0.16$ obtained [6] when regions of a plume far away from the burner are considered. In a room fire it is generally not possible to be far enough above the burner for the far region expressions to hold. McCaffrey and Rockett [7] have recently examined in more detail the experimental verification of entrainment constants. Instead of using the image burner technique they expressed the different entrainment in wall and corner positions simply as part of an effective k_e.

5. HEAT LOSSES AND HEAT BALANCE

5.1 Convective Ceiling Heat Transfer

The test series was originally not designed with the objective of verifying ceiling heat transfer relationships. However, suitable ceiling heat flux measurements were available for five tests. Table 11 lists the radiative and total heat flux measurements taken at the center of the ceiling. The convective component is obtained by subtraction. Since maximum gas temperatures were not measured in the immediate vicinity of the heat flux meters, a certain amount of extrapolation had to be performed. The peak gas temperature, T_f, below the heat flux meters was assumed to be equal to the reading of TC01 multiplied by the ratio of the reading at TC21 divided by the TC23 values. (TC21 is located near the height of peak temperatures, while TC23 is at the same height as TC01.) Ratios were taken using temperature rise values, $(T_f - T_\infty)$, with $T_\infty = 20°C$. The surface temperature of the heat flux gage is equal to that of the cooling water, T = 15°C, plus the average rise in foil temperature, which was taken as 5°C, giving $T_s = 20°C$. Heat average coefficients were then calculated according to the relation

$$h_c = \frac{\dot{q}_c}{T_f - T_s}$$

Measured values ranged from 6.1 to 24.6 W/m^2-K.

Several references are available for empirical and theoretical ceiling heat transfer coefficients. Alpert [8] has studied the ceiling flows for a situation where the walls are effectively very far away from the region of interest. For that case, a nondimensional heat transfer coefficient can be derived:

$$h_c^* = h_c \left(\frac{T_\infty H}{gQ \rho_\infty^2 C_p^2} \right)^{1/3}$$

where Q is the plume heat flow. Alpert then gives the radial dependence of $h_c^* = 0.67\ f\ \frac{r}{H}^{-0.69}$, valid outside the plume region.

The f is a friction factor and will most likely be in the range of 0.02 - 0.04 according to Alpert's results. Values of f calculated from the heat transfer coefficients in the present case are given in table 11 and range from 0.005 to 0.017.

Comparison can also be made with the direct measurements of Zukoski and Kubota [9] who studied both the infinite ceiling and the wall-obstructed case. Zukoski reports his results as a band of values within which the infinite ceiling measurements fell. The obstructed ceiling results fell near the lower limit of the infinite ceiling measurements. Table 11 also shows a comparison between experimentally determined h_c^* and Zukoski's values. The results from test 104 show a significant disagreement with both Alpert's and Zukoski's findings. Since the ratio of the radiant to the convective fluxes is also quite different for test 104 than for the four other tests, the discrepancy may be attributable to instrument malfunction in this case. The remaining four tests all show h_c^* values which fall right around Zukoski's lower limit. These data also agree well with Alpert's equation, if it is taken that in the obstructed ceiling case a value of $f \simeq 0.013$ is appropriate.

Use of expressions for h_c based on plume flow knowledge would be difficult in a room with realistic furnishings because neither the Q nor the r/H values would be well defined. It would be much more useful in that case to use some expression that depends only on locally measureable quantities. An expression can be derived that is dependent on H and T_f, but not on Q, by using Alpert's results for T_f as a function of Q. The resulting expression is

$$h_c = C_1\ f\left(\frac{r}{H}\right)^{-0.44} H^{1/2}\ \Delta T_f^{\ 1/2}$$

where $\Delta T_f = T_f - T_\infty$. For a Gaussian ceiling jet profile the value of C_1 would be 85. The present data are, instead, best fitted by $C_1 = 47$.

It is also of some interest to compare the experimental results against the simple relationship for turbulent heat transfer by natural convection to a horizontal flat plate. Accepted literature values [10] are that

$$Nu = \frac{h_c L}{k} = 0.14 \ (Gr \ Pr)^{1/3}$$

where $$Gr = \frac{gL^3 \Delta T}{T\nu^2}$$

And T is some average temperature for the problem. The gas viscosity ν, conductivity k, and Prandtl number Pr are to be evaluated at T. The convective transfer coefficient is then

$$h_c = 0.14 \ k \left(\frac{g \ Pr}{\nu^2} \ \frac{\Delta T}{T} \right)^{1/3}$$

Calculations for flat plate h_c on the above basis are also shown in table 11. For tests 105-110 the experimental values average twice the literature values. Such a difference is not unexpected in view of the differences between the velocity profiles in the ceiling jet, and in the flat plate situations. Still, if an equation of this form might be useful in estimating the ceiling fluxes, then the k and ν factors can well be approximated by power law expressions. When multiplied by 2.0 this gives

$$h_c \simeq 162 \ T^{-2/3} \Delta T^{1/3} \quad (W/m^2\text{-}K)$$

This expression can then be used for approximate calculations. Although it is cruder than Alpert's and Zukoski's expressions, it does not require the knowledge of the distance r from the plume center.

5.2 Wall and Ceiling Heat Losses

In the steady state the heat transferred to the ceiling from the room fire gases is equal to the heat conducted through the ceiling and is equal to the heat leaving from the unexposed surface. The desired heat flux is, of course, not what is measured by the total heat flux meter, since its surface temperature is ≃20°C and is not the ceiling surface temperature.

Theoretically, the ceiling heat loss might be determined by using the measurements on the exposed side. The radiative transfer would be the measured incident flux minus reradiation at the surface temperature. The convective term could be obtained by using measured temperatures and the best h_c value. Practically, this procedure is not very useful since it would entail subtracting two large numbers to get a small difference. The desired heat flux can best be evaluated by considering the conditions at the unexposed surface. The heat leaving the unexposed surface can be expressed as

$$q_w = q_{cw} + q_{rw}$$

$$= 0.081\ T^{-2/3}\ \Delta T^{4/3} + \sigma\varepsilon_w(T_w^4 - T_\infty^4) \quad (\mathrm{kW/m^2})$$

The gypsum wallboard emissivity can be taken as $\varepsilon_w \simeq 0.9$ for the temperatures of interest. To get average ceiling unexposed surface values, T_w can be averaged over the readings for TC47, TC49, TC54, and TC81. The calculations are given in table 12. According to these results the ceiling heat losses ranged from 0.18 to 0.40 kW/m^2. A similar calculation can be made for the wall losses. It is assumed that losses are non-negligible only through the upper half of the walls. For the walls, T_w is taken as the average from the readings of TC40 and TC44. In the vertical plate case [10], the constant in the Nusselt number expression is 0.12, giving

$$h_c \simeq 0.069\ T^{-2/3}\ \Delta T^{1/3}$$

The resultant wall fluxes, which were of similar magnitude as the ceiling fluxes have been multiplied by the effective wall area, 16.0 m^2, to yield the total wall losses. Total room surface losses — wall and ceiling — are added up and tabulated in table 12. These losses range from 3% to 6% of the fuel heat release rate.

5.3 Wall Thermal Conductance

The wall (and ceiling) material in these tests consisted of two layers: 13 mm of cement-asbestos board on the inside and 16 mm of Type X gypsum wallboard on the outside of the room. The ambient temperature values for the conductivities are taken as

Material	K (W/m-K)
Cement-asbestos board	0.14
Gypsum wallboard	0.20

These conductivities are not strongly dependent on temperature. The total thermal resistance, for the thickness used, is R = 0.17 m^2-K/W.

This composite resistance should agree with the measured resistance, which can be gotten as

$$R = \frac{T_s - T_w}{\dot{q}_w}$$

The average measured resistance is seen from the computations in table 13 to be R = 0.74, leaving a net resistance of R = 0.74 - 0.17 = 0.57 m^2-K/W unaccounted for.

The above computation was based on the implicit assumption that thermal contact between the two wall strata is perfect. Since no special pains were taken to achieve a tight fit, however, it is likely that with repeated heating-induced deformation some gaps could exist. It is of interest to calculate the possible resistance that such a gap could introduce. The maximum gap thickness that can be expected is on the order of L = 2 mm. The thermal conductivity of air is k = 0.036 W/m-K at an average interface temperature of 428 K. The actual gap conductance is the sum of the gas conductance plus the radiative conductance

$$\frac{1}{R} = \frac{1}{R_c} + \frac{1}{R_r}$$

$$= \frac{k}{L} + 4\,\varepsilon\sigma T^3$$

$$= 18 + 15$$

$$R = 0.03 \text{ m}^2\text{-K/W}$$

Thus, contact resistance can account for up to 5% of the difference between measured and calculated R values. The remaining discrepancy must be attributed to either changed material properties with heating or to inapplicability of the flat plate expressions for the external heat transfer.

5.4 Heat Balance

The heat balance can be written for the system consisting of the gas within the room as follows:

$$\dot{h}_{in} - \dot{h}_{out} - \dot{q}_{loss} = 0$$

where h denotes enthalpy. Assumptions of steady state and negligible kinetic energy have been included. The combination reaction is

$$1 \text{ kg fuel} + r \text{ kg air} \longrightarrow (1+r) \text{ kg products}$$

and $\Delta h_c = h_f + rh_a - (1+r)h_p$

where f, a, and p denote fuel, air, and products, respectively, and all enthalpy values are taken at a reference temperature of 298 K.

The heat balance becomes

$$\dot{m}_f \Delta h_c - \dot{m}_p \left(h_{p,T_f} - h_{p,298K}\right) - \dot{q}_{loss} = 0$$

It simplifies analysis considerably to assume that the heat capacity of the products is the same as of air. Also, the heat capacity of air at 300°C is only 1% different than at room temperature; thus it is useful to assume that $\Delta h_{air} = C_p \Delta T$, with C_p being taken as 1.0 kJ/kg-K. Then

$$\dot{m}_f \Delta h_c - \dot{m}_o (1.00) \Delta T - \dot{q}_{loss} = 0$$

where $\dot{m}_o$ is the total outflow. The calorific value, Δh_c, to be used is the lower, or net, value which excludes the latent heat of water vaporization. This is most convenient since otherwise the h_p term could not be simply represented with a constant C_p but would have to include a vaporization contribution. ΔT can be computed on a flow-weighted basis at the doorway. The calculations are given in table 14.

The wall surface losses were considered in section 5.2 and are also included in table 14. The radiant losses through the doorway can be estimated by the expression

$$\dot{q} \simeq \sigma \sum_i \left(T_i^4 - T_\infty^4\right) \Delta A_i$$

where the summation is taken over temperature readings at the various heights of the doorway.

The error in the heat balance is also shown in table 14. The range is from 6.8 to 13.8%, with a mean of 10.5%. The errors consistently show that the losses exceeded the heat gain. No allowance was made for incomplete combustion losses. Incomplete combustion losses would further serve to increase the disparity in the measurements. The dominant source of error is presumed to be in the doorway velocity measurements. The poor agreement shows that using two or three points to measure a distribution which, according to theory, is expected to be parabolic, is quite insufficient. In future work more vertical sampling positions should be provided to improve the accuracy. It should be observed, however, that in a test of actual furnishings steady state conditions will not exist. This means that a horizontal traversing of the velocity probes is not readily feasible and, therefore, that greater reliance will have to be placed on accuracy of the vertical distribution.

6. EFFECT OF AFTER-BURNER ON BURN ROOM FIRE

The test facility exhaust hood is equipped with a draft-inducing exhaust blower and an after-burner for smoke abatement. Tests were conducted to determine if the induced flow due to after-burner operation had an effect on burn-room measurements. Repeat room fires were conducted for tests 101 and 110 with the after-burner on and off. The results for test 101 are given in table 15; the results of test 110 were similar. The temperature rises with the after-burner were, on the average, 7% lower than without. This difference was judged not significant. Since the test burner flames were not smoky, and since the effect of the after-burner on the temperature fields was small, and in the interest of energy conservation (the after-burner consumed some ten times more fuel than the test burner) all the tests reported in the previous sections were conducted with the after-burner off.

7. DISCUSSION

The measuring of the room fire variables using a steady state gas burner as fuel source represents idealized conditions for measurement. A room fire fueled by furnishings will, by contrast, be strongly time dependent and poorly characterized in its release rate. Yet the present series of tests has shown numerous difficulties in accurate measurement and in theoretical analysis.

The most significant shortcoming was the small number of velocity measurement probes in the doorway. It is clear that two or three probes are insufficient for characterizing the inflow or the outflow, both of which are nonlinear functions of height. It is apparent that in future work a minimum of 10 probes should be used, distributed about equally between inflow and outflow. The failure to obtain adequate mass and heat balances can be attributed directly to the insufficiency of velocity sampling points.

Plume entrainment and ceiling heat transfer measurements are intrinsically more difficult to make than doorway flows. The agreements with theory and literature values achieved for these measurements can be considered satisfactory. Doorway discharge coefficients determined were also as expected.

Heat transfer through the ceiling and upper wall surfaces could not be satisfactorily compared with material property values and literature heat transfer coefficients. This might be due to degraded material properties, errors in surface temperature measurement, or inapplicability of the literature heat transfer coefficients.

8. SUMMARY

A series of detailed steady state measurements was taken in a furnishings burn room fueled with a gas burner. These measurements were intended to be used as a calibration for the constants and procedures used in analysis since only with a controlled fuel source, such as a gas burner, can heat release rates be adequately characterized.

The major observations were the following:

- A general calculational procedure for making heat and mass flow balance in a test burn compartment has been evolved. The procedures adopted are quite general and will be useful for situations where steady state flows are to be determined in a room with a vertical single ventilation opening and a single horizontal fuel source located in the lower half of the room. For time-dependent flows only a minor extension is needed in the analysis of wall losses.
- Agreement with room fire theories is good, generally within the error bounds ($\simeq$ factor of 2) of the theories.
- Because of insufficient precision in the doorway velocity measurements a satisfactory heat balance could not be obtained. The doorway flow is the dominant loss term at steady state; thus accuracy in determining wall losses and doorway radiation was not limiting.
- A plume entrainment coefficient of 0.56 was calculated from the data. This is in general agreement with other data showing high entrainment coefficient in the source region of combusting plumes.
- Discharge coefficients were calculated which were quite close to the literature value of 0.68 on outflow, but less on inflow.
- Doorway mass inflows and outflows showed an imbalance of about 8%, largely attributable to insufficient velocity probe positions.
- Good agreement was obtained for the measured ceiling heat transfer compared with literature values.
- No agreement was obtained between wall loss calculations based on (a) the measured unexposed surface temperatures, using standard convective coefficient formulas, and (b) the wall material conductance and measured temperature drops. Several explanations are possible, but a single cause was not reliably identified.
- Required thermocouple radiation corrections in the doorway were about -10°C for inflow, and about +20°C for outflow temperatures on the order of 300°C. In the early stages of heating the inflow correction would be more and the outflow correction less.

9. ACKNOWLEDGMENTS

The authors would like to thank both Mr. R. Smith and Mr. T. Prather for their assistance in the experimental work, and Mr. R. Smith for his assistance in developing a computer program and reduction of experimental data.

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Table 1. List of instrumentation

Channel Number	Thermocouples
01	Center of room, 0.31 m from ceiling
02	Center of room, 0.92 m from ceiling
03	Center of room, 1.53 m from ceiling
04	Center of room, 2.14 m from ceiling
05	SE corner of room, 0.31 m from ceiling
06	SE corner of room, 0.92 m from ceiling
07	SE corner of room, 1.53 m from ceiling
08	SE corner of room, 2.14 m from ceiling
09	NE corner of room, 0.31 m from ceiling
10	NE corner of room, 0.92 m from ceiling
11	NE corner of room, 1.53 m from ceiling
12	NE corner of room, 2.14 m from ceiling
13	NW corner of room, 0.31 m from ceiling
14	NW corner of room, 0.92 m from ceiling
15	NW corner of room, 1.53 m from ceiling
16	NW corner of room, 1.83 m from ceiling
17	SW corner of room, 0.31 m from ceiling
18	SW corner of room, 0.92 m from ceiling
19	SW corner of room, 1.53 m from ceiling
20	SW corner of room, 2.14 m from ceiling
21	0.91 m from E wall, 0.91 m from S wall, 0.08 m from ceiling
22	0.91 m from E wall, 0.91 m from S wall, 0.10 m from ceiling
23	0.91 m from E wall, 0.91 m from S wall, 0.31 m from ceiling
24	0.91 m from E wall, 0.91 m from S wall, 0.61 m from ceiling
25	0.91 m from E wall, 0.91 m from S wall, 0.92 m from ceiling
26	0.91 m from E wall, 0.91 m from S wall, 1.22 m from ceiling
27	0.91 m from E wall, 0.91 m from S wall, 1.52 m from ceiling
28	0.91 m from E wall, 0.91 m from S wall, 1.83 m from ceiling
29	0.91 m from E wall, 0.91 m from S wall, 2.13 m from ceiling
32	NW corner of room, 1.22 m from ceiling
33	NW corner of room, 2.13 m from ceiling
35	On S wall, 2.28 m from E wall, 1.83 m from ceiling
36	On S wall, 1.06 m from E wall, 0.61 m from ceiling
37	On E wall, 1.22 m from S wall, 1.83 m from ceiling
38	On rear wall surface, behind TC 37
39	On E wall, 1.22 m from S wall, 0.61 m from ceiling
40	On unexposed wall surface, behind TC 39
41	On N wall, 1.22 m from E wall, 1.83 m from ceiling
42	On unexposed wall surface, behind TC 41
43	On N wall, 2.43 m from E wall, 0.61 m from ceiling
44	On unexposed wall surface, behind TC 43
45	On ceiling, 1.70 m from S wall, 0.31 m from E wall
46	On ceiling, 1.70 m from S wall, 1.83 m from E wall
47	On unexposed ceiling surface, behind TC 46
48	On ceiling, 1.70 m from S wall, 2.74 m from E wall
49	On unexposed ceiling surface, behind TC 48
50	On ceiling, 1.70 m from S wall, 3.35 m from E wall
51	On ceiling, 0.91 m from S wall, 2.59 m from E wall
52	On ceiling, 0.91 m from S wall, 0.91 m from E wall
53	On ceiling, 2.49 m from S wall, 0.91 m from E wall
54	On unexposed ceiling surface, behind TC 53
55	On ceiling, 2.49 m from S wall, 2.59 m from E wall
56	At doorway centerline, 0.09 m below top
57	At doorway centerline, 0.13 m below top
58	At doorway centerline, 0.30 m below top
59	At doorway centerline, 0.61 m below top
60	At doorway centerline, 0.91 m below top
61	At doorway centerline, 1.22 m below top
62	At doorway centerline, 1.52 m below top
63	At doorway centerline, 1.83 m below top

Table 1. (continued)

Channel Number	Thermocouples
65	At doorway, 1.47 m from S wall, 1.52 m below top
67	Center of adjacent room, 0.91 m from ceiling
68	Center of adjacent room, 1.52 m from ceiling
69	Center of adjacent room, 2.13 m from ceiling
70	Adjacent room doorway centerline, 0.30 m below top
71	Adjacent room doorway centerline, 0.91 m below top
72	Adjacent room doorway centerline, 1.52 m below top
73	Adjacent room doorway centerline, 2.13 m below top
74	On floor, 0.91 m from S wall, 0.91 m from E wall
75	On floor, 2.33 m from S wall, 1.07 m from E wall
76	NW corner of room, on floor
77	SW corner of room, on floor
78	On floor, 1.70 m from S wall, 0.61 m from E wall
79	On floor, 1.70 m from S wall, 2.59 m from E wall
80	On floor, 1.70 m from S wall, 3.05 m from E wall
81	On unexposed ceiling surface, behind TC 45
82	Quick-response thermocouple, adjacent to TC 48
83	At velocity probe 113, 0.13 m below doorway top
84	At velocity probe 112, 0.30 m below doorway top
85	At velocity probe 111, 0.66 m below doorway top
86	At velocity probe 110, 1.07 m below doorway top
87	At velocity probe 109, 1.37 m below doorway top
88	At velocity probe 108, 1.91 m below doorway top
89	On unexposed ceiling surface, behind TC 90
90	Ceiling of adjacent room, 0.61 m W of test room
91	Floor of adjacent room, 0.61 m W of test room

	Heat Flux Meters
100	Radiometer, facing East, in adjacent room, 0.91 m W of test room, 1.53 m from ceiling
101	Radiometer, facing East, on W wall, 0.61 m from S wall, 0.61 m from ceiling
103	Total heat flux meter, on ceiling, 1.70 m from S wall, 2.89 m from E wall
104	Total heat flux meter, on ceiling, center of room
105	Radiometer, same location as heat flux meter 104
106	Radiometer, on floor, 1.70 m from S wall, 2.59 m from E wall
107	Total heat flux meter, same location as heat flux meter 106
114	Total heat flux meter, same location as heat flux meter 101

	Velocity Probes
108	At doorway, 1.91 m below doorway top
109	At doorway, 1.37 m below doorway top
110	At doorway, 1.07 m below doorway top
111	At doorway, 0.66 m below doorway top
112	At doorway, 0.30 m below doorway top
113	At doorway, 0.13 m below doorway top

Table 2. Burn-room tests conducted

Test No.	Burner Position	Heat Release Rate (kW)
101	Center	164
102	Center	184
103	Center	203
104	Corner	164
105	Corner	184
106	Corner	203
107	Back	164
108	Back	184
109	Back	203
110	Back	228

Table 3. Test data (at steady state)

Channel	Thermocouples (C)									
	Test Number									
	101	102	103	104	105	106	107	108	109	110
01	314	360	377	224	273	289	235	270	291	308
02	325	445	458	172	222	237	168	197	214	225
03	648	718	748	52	81	87	62	73	84	87
04	65	61	64	36	51	54	42	48	54	55
05	--	271	288	241	290	315	271	313	337	337
06	250	265	281	211	262	281	233	268	289	306
07	81	87	92	51	77	84	69	83	100	107
08	61	29	27	22	30	29	27	26	31	29
09	238	263	280	261	322	349	260	299	324	347
10	234	253	270	239	300	320	229	260	282	298
11	63	68	72	70	98	106	55	65	77	79
12	56	62	66	56	79	87	50	58	70	72
13	252	272	287	218	272	291	230	259	280	295
14	242	257	273	201	250	265	212	241	263	276
15	92	98	101	59	91	100	67	78	94	99
16	70	78	82	50	69	77	57	66	78	83
17	237	259	276	190	237	253	217	246	267	282
18	250	266	284	199	248	265	222	253	274	292
19	82	87	94	55	83	92	65	77	91	97
20	64	69	75	23	64	69	27	55	67	72
21	264	288	306	235	287	308	264	303	330	348
22	273	299	318	238	291	313	--	--	--	--
23	--	276	293	226	275	296	235	269	290	305
24	247	267	283	202	258	276	219	250	272	285
25	--	245	265	184	232	247	204	234	256	273
26	158	141	158	113	163	177	132	160	182	193
27	75	82	86	62	93	103	80	96	111	120
28	80	86	92	49	68	72	69	81	92	97
29	70	75	81	42	60	64	58	70	80	87
32	169	159	178	118	169	180	100	125	143	153
33	57	64	66	40	55	57	43	50	60	61
35	64	71	74	39	60	64	46	53	62	64
36	226	242	257	190	234	252	229	263	287	300
37	100	96	102	57	82	92	145	173	194	206
38	31	26	25	20	25	24	25	26	28	28
39	46	234	248	197	248	266	203	234	260	273
40	53	50	50	40	60	62	41	47	55	55
41	58	105	112	93	130	144	85	101	120	126
42	37	33	32	25	33	33	32	34	38	37
43	220	252	269	198	247	265	211	240	261	275
44	63	58	58	41	56	57	48	53	58	58
45	242	253	267	234	282	305	306	352	378	406
46	323	379	416	234	295	314	240	276	298	314
47	44	88	86	31	56	57	35	41	53	48
48	286	--	--	204	--	--	212	245	264	276
49	62	66	66	46	61	63	51	52	58	55
50	242	259	277	166	213	228	177	204	223	237
51	250	274	293	184	234	250	204	232	253	267
52	254	265	282	207	252	271	247	284	308	328
53	263	280	291	275	337	364	260	296	320	343

Table 3. (continued)

Channel	Thermocouples (C)									
	Test Number									
	101	102	103	104	105	106	107	108	109	110
54	40	38	39	30	38	37	35	36	39	40
55	282	303	321	214	274	292	216	247	268	282
56	248	257	274	197	249	266	210	238	255	272
57	262	278	297	211	260	278	231	263	283	300
58	249	--	--	208	--	--	227	257	279	294
59	229	248	264	198	250	268	219	245	268	281
60	109	103	115	70	110	124	70	84	95	103
61	61	86	93	34	83	91	40	44	53	62
62	44	41	43	28	39	42	32	36	41	41
63	41	38	41	28	37	37	33	35	39	38
65	--	41	43	29	39	40	33	36	41	41
67	126	111	128	48	118	124	49	55	63	68
68	42	41	42	28	43	43	34	38	42	42
69	33	29	29	20	31	29	23	27	27	27
70	30	26	26	20	28	27	23	24	24	25
71	29	27	26	25	28	27	29	30	31	31
72	28	24	24	19	26	25	22	23	24	25
73	26	23	22	18	25	23	21	21	22	22
74	107	105	113	64	90	100	111	137	159	168
75	76	67	65	38	55	60	59	70	88	89
76	75	61	65	41	53	56	47	50	60	62
77	88	104	118	58	92	102	61	85	98	98
78	95	113	119	96	136	153	139	162	185	192
79	83	99	105	--	87	93	--	--	--	--
80	75	65	78	34	60	63	41	46	52	55
81	52	52	52	44	58	57	53	59	66	67
82	254	225	235	130	180	192	136	152	167	171
83	253	272	291	208	257	273	226	257	280	292
84	245	259	274	--	256	275	--	--	--	--
85	224	227	235	177	234	249	180	202	245	243
86	58	69	70	33	60	80	43	44	60	59
87	46	44	43	35	40	40	41	47	55	56
88	39	53	43	28	37	38	33	36	39	41
89	188	42	43	--	44	44	--	--	--	--
90	207	216	228	--	210	223	--	--	--	--
91	53	32	32	--	32	32	--	--	--	--
	Heat Flux Meters (kW/m^2)									
100	1.5	1.4	1.7	0.7	1.2	1.4	0.9	--	--	1.5
101	2.8	3.2	3.1	1.9	2.9	3.7	0.9	--	--	5.1
103	8.2	10.4	11.7	6.4	5.1	6.2	6.8	--	--	9.8
104	9.2	15.8	16.3	3.1	6.7	8.7	6.1	--	--	7.6
105	11.8	3.1	3.7	1.8	1.4	1.8	2.0	--	--	3.2
106	2.2	2.9	3.3	1.3	2.0	2.4	1.5	--	--	2.8
107	2.8	3.3	3.8	1.5	2.4	2.9	1.7	--	--	3.1
114	--	5.4	6.4	2.0	2.9	3.4	4.3	--	--	5.7

Table 3. (continued)

Channel	Velocity Probes* (m/s)									
	Test Number									
	101	102	103	104	105	106	107	108	109	110
108	--	-0.90	-0.97	-0.75	-0.98	-1.05	-0.73	--	--	-0.88
109	-	-0.53	-0.63	-0.42	-0.74	-1.01	-0.35	--	--	-0.50
110	--	-0.40	-0.41	-0.41	-0.48	-0.28	-0.20	--	--	-0.43
111	1.70	1.89	2.00	1.40	2.25	2.31	1.36	--	--	1.72
112	1.96	1.31	1.52	1.34	1.81	1.77	1.32	--	--	1.42
113	2.26	2.67	2.44	1.82	2.53	2.46	2.05	--	--	2.19

*Values at doorway centerline. See tables 4-8 for other values.

Table 4. Mass flow for test 102

Distance from doorway top (m)	Quantity	Distance from doorway centerline					
		0	0.076 m	0.152 m	0.229 m	0.305 m	0.381 m
0.13	u(m/s)	2.28	2.28	2.31	2.21	2.04	1.89
	T_i(°C) measured	272	271	271	262	257	254
	T_f(°C) corrected	289	288	288	277	271	268
	ρ (kg/m^3)	0.63	0.63	0.63	0.64	0.65	0.65
	ρuA (kg/s)	0.0859	0.0861	0.0872	0.0852	0.0795	0.0741
0.66	u	1.89	1.88	1.91	1.83	1.78	1.19
	T_i	228	229	229	235	229	169
	T_f	236	237	237	244	237	166
	ρ	0.69	0.69	0.69	0.68	0.69	0.80
	ρuA	0.0945	0.0928	0.0943	0.0894	0.0878	0.0683
1.07	u	-0.40	-0.25	-0.34	-0.30	-0.32	-0.23
	T_i	69	67	75	73	70	73
	T_f	47	41	52	49	46	49
	ρ	1.10	1.12	1.08	1.10	1.11	1.10
	ρuA	-0.0237	-0.0151	-0.0198	-0.0176	-0.0190	-0.0135
1.37	u	-0.53	-0.58	-0.48	-0.52	-0.50	-0.52
	T_i	44	42	40	39	40	46
	T_f	22	20	16	16	17	24
	ρ	1.20	1.20	1.22	1.22	1.22	1.19
	ρuA	-0.0405	-0.0445	-0.0373	-0.0405	-0.0388	-0.0394
1.91	u	-0.90	-0.94	-0.98	-1.04	-1.12	-1.26
	T_i	53	53	46	53	48	54
	T_f	35	35	28	36	31	38
	ρ	1.15	1.15	1.17	1.14	1.16	1.13
	ρuA	-0.0767	-0.0800	-0.0854	-0.0883	-0.0967	-0.1062

Total flow out = 1.024 kg/s
Total flow in = 0.883 kg/s
Average = 0.954 kg/s

Difference = 7%

Table 5. Mass flow for test 104

Distance from doorway top (m)	Quantity	Distance from doorway centerline					
		0	0.076 m	0.152 m	0.229 m	0.305 m	0.381 m
0.13	u(m/s)	2.12	2.39	2.39	2.40	2.42	2.20
	T_i(°C) measured	208	209	211	212	214	202
	T_f(°C) corrected	218	219	221	222	224	211
	ρ(kg/m^3)	0.72	0.72	0.71	0.71	0.71	0.73
	ρuA(kg/s)	0.0915	0.1030	0.1025	0.1027	0.1031	0.0963
0.66	u	1.64	1.60	1.71	1.71	1.72	1.74
	T_i	177	182	188	184	184	183
	T_f	182	188	195	180	190	189
	ρ	0.78	0.77	0.75	0.76	0.76	0.76
	ρuA	0.0907	0.0873	0.0920	0.0929	0.0934	0.0948
1.07	u	-0.48	-0.51	-0.47	-0.42	-0.37	-0.40
	T_i	33	35	36	36	36	33
	T_f	19	22	22	22	21	19
	ρ	1.21	1.20	1.19	1.20	1.20	1.21
	ρuA	-0.0312	-0.0328	-0.0302	-0.0270	-0.0239	-0.0261
1.37	u	-0.49	-0.40	-0.51	-0.49	-0.50	-0.45
	T_i	35	36	37	35	30	30
	T_f	22	22	24	22	16	16
	ρ	1.20	1.20	1.19	1.20	1.22	1.22
	ρuA	-0.0374	-0.0306	-0.0386	-0.0374	-0.0388	-0.0350
1.91	u	-0.87	-0.87	-0.96	-1.00	-1.08	-1.15
	T_i	28	28	28	27	27	27
	T_f	17	17	17	16	16	17
	ρ	1.22	1.22	1.22	1.22	1.22	1.22
	ρuA	-0.0788	-0.0788	-0.0868	-0.0907	-0.0979	-0.1041

Total flow out = 1.150 kg/s
Total flow in = 0.926 kg/s
Average = 1.038 kg/s

Difference = 11%

Table 6. Mass flow for test 105

Distance from doorway top (m)	Quantity	Distance from doorway centerline: 0	0.076 m	0.152 m	0.229 m	0.305 m	0.381 m
0.13	u (m/s)	2.16	2.33	2.24	2.25	2.28	2.20
	T_i (°C) measured	257	260	260	259	259	248
	T_f (°C) corrected	271	274	275	273	273	260
	ρ (kg/m^3)	0.65	0.64	0.64	0.65	0.65	0.66
	ρuA (kg/s)	0.0841	0.0902	0.0867	0.0872	0.0884	0.0874
0.66	u	1.92	1.79	1.83	1.85	1.99	1.30
	T_i	234	230	222	241	241	208
	T_f	243	239	229	252	252	212
	ρ	0.68	0.69	0.70	0.67	0.67	0.73
	ρuA	0.0936	0.0880	0.0918	0.0887	0.0955	0.0674
1.07	u	-0.42	-0.38	-0.33	-0.31	-0.29	-0.32
	T_i	60	63	67	68	68	73
	T_f	38	40	44	45	44	50
	ρ	1.14	1.13	1.11	1.11	1.11	1.09
	ρuA	-0.0257	-0.0230	-0.0198	-0.0185	-0.0173	-0.0188
1.37	u	-0.63	-0.50	-0.55	-0.64	-0.56	-0.54
	T_i	40	40	39	37	36	36
	T_f	19	17	17	16	14	14
	ρ	1.21	1.22	1.22	1.22	1.23	1.23
	ρuA	-0.0485	-0.0387	-0.0427	-0.0498	-0.0439	-0.0424
1.91	u	-0.84	-0.91	-0.94	-1.02	-1.05	-1.15
	T_i	37	37	37	37	39	38
	T_f	18	18	19	19	21	21
	ρ	1.21	1.21	1.21	1.21	1.20	1.20
	ρuA	-0.0758	-0.0720	-0.0845	-0.0916	-0.0935	-0.1026

Total flow out = 1.049 kg/s
Total flow in = 0.909 kg/s

Average = 0.979 kg/s

Difference = 7%

Table 7. Mass flow for test 106

Distance from doorway top (m)	Quantity	Distance from doorway centerline					
		0	0.076 m	0.152 m	0.229 m	0.305 m	0.381 m
0.13	u(m/s)	2.10	2.25	2.27	2.34	2.33	2.25
	T_i(°C) measured	273	275	276	277	278	274
	T_f(°C) corrected	290	292	293	294	295	291
	ρ (kg/m^3)	0.63	0.62	0.62	0.62	0.62	0.63
	ρuA(kg/s)	0.0791	0.0844	0.0850	0.0873	0.0869	0.0846
0.66	u	1.97	1.91	1.94	1.91	2.09	1.39
	T_i	249	241	236	254	253	243
	T_f	260	250	244	266	265	254
	ρ	0.66	0.67	0.68	0.65	0.66	0.67
	ρuA	0.0930	0.0919	0.0944	0.0891	0.0978	0.0664
1.07	u	-0.24	0.10	0.15	0.29	0.15	0.23
	T_i	80	87	84	86	85	84
	T_f	53	58	57	63	58	59
	ρ	1.08	1.07	1.07	1.05	1.07	1.06
	ρuA	-0.0140	0.0058	0.0086	0.0164	0.0086	0.0131
1.37	u	-0.86	-0.73	-0.72	-0.77	-0.78	-0.74
	T_i	41	39	38	38	38	39
	T_f	20	17	16	16	16	17
	ρ	1.20	1.22	1.22	1.22	1.22	1.22
	ρuA	-0.0660	-0.0566	-0.0561	-0.0599	-0.0606	-0.0574
1.91	u	-0.89	-0.90	-1.00	-1.05	-1.13	-1.21
	T_i	38	37	39	38	37	36
	T_f	17	16	19	18	18	17
	ρ	1.22	1.22	1.21	1.21	1.21	1.22
	ρuA	-0.0805	-0.0816	-0.0898	-0.0945	-0.1019	-0.1093

Total flow out = 1.092 kg/s
Total flow in = 0.928 kg/s

Average flow = 1.010 kg/s

Difference = 8%

Table 8. Mass flow for test 107

Distance from doorway top (m)	Quantity	Distance from doorway centerline 0	0.076 m	0.152 m	0.229 m	0.305 m	0.381 m
0.13	u(m/s)	2.40	2.44	2.47	2.46	2.51	1.79
	T_i(°C) measured	226	225	228	228	228	191
	T_f(°C) corrected	237	236	240	240	240	197
	ρ(kg/m³)	0.69	0.69	0.69	0.69	0.69	0.75
	ρuA(kg/s)	0.0995	0.1015	0.1020	0.1016	0.1037	0.0807
0.66	u	1.59	1.62	1.67	1.71	1.77	1.76
	T_i	180	187	197	192	191	181
	T_f	184	192	204	198	197	185
	ρ	0.77	0.76	0.74	0.75	0.75	0.77
	ρuA	0.0875	0.0876	0.0881	0.0913	0.0948	0.0967
1.07	u	-0.24	-0.04	0.06	0.03	0.15	0.28
	T_i	43	40	41	41	41	39
	T_f	24	12	18	16	21	22
	ρ	1.19	1.24	1.21	1.22	1.20	1.20
	ρuA	-0.0153	-0.0026	0.0039	0.0020	0.0097	0.0180
1.37	u	-1.41	-0.29	-0.43	-0.44	-0.39	-0.45
	T_i	41	41	43	43	37	37
	T_f	25	23	27	28	20	21
	ρ	1.18	1.19	1.17	1.17	1.20	1.20
	ρuA	-0.0310	-0.0220	-0.0321	-0.0329	-0.0299	-0.0344
1.91	u	-0.86	-0.85	-0.86	-0.93	-1.02	-1.14
	T_i	33	34	33	32	32	31
	T_f	20	21	20	19	20	19
	ρ	1.20	1.20	1.20	1.21	1.21	1.21
	ρuA	-0.0770	-0.0758	-0.0770	-0.0835	-0.0914	-0.1023

Total flow out = 1.169 kg/s
Total flow in = 0.707 kg/s

Average flow = 0.938 kg/s

Difference = 25%

Table 9. Theoretical mass flows

Test	Z_N (m)	N	D	R	Theoretical $\dot{m}/C$ (kg/s)	Measured $\dot{m}_o$ (kg/s)	C_o	Measured $\dot{m}_i$ (kg/s)	C_i
102	1.14	0.54	0.45	1.89	1.56	1.02	0.65	0.88	0.56
104	1.14	0.54	0.44	1.68	1.54	1.15	0.75	0.93	0.60
105	1.13	0.53	0.42	1.86	1.61	1.05	0.65	0.91	0.57
106	1.02	0.48	0.30	1.93	1.88	1.09	0.58	0.93	0.49
107	1.03	0.49	0.31	1.72	1.80	1.17	0.65	0.71	(0.39)

Table 10. Plume flows

Test	Burner position	$\dot{m}_f$ (kg/s)	$\frac{\dot{m}_p}{\dot{m}_f}$	$\frac{Z_D}{\dot{m}_f^{2/5}}$	k_e
102	Center	0.0035	273	9.20	0.26
104	Corner	0.0032	324	9.33	0.55
105	Corner	0.0035	280	8.59	0.56
106	Corner	0.0039	259	5.88	0.93
107	Back	0.0032	293	6.37	0.57

ω = 0.091

ρ_f = 0.68 kg/m^3 at 15°C

Table 11. Ceiling heat transfer

11a. Experimental measurements

Test	Flux meter readings						
	$\dot{q}_r$ (kW/m^2)	$\dot{q}_{total}$ (kW/m^2)	$\dot{q}_c$ (kW/m^2)	T_f (°C)	T_s (°C)	ΔT_c (K)	h_c (W/m^2-K)
104	1.8	3.1	1.3	233	20	213	6.1
105	1.4	6.7	5.3	285	20	265	20.0
106	1.8	8.7	6.9	301	20	281	24.6
107	2.0	6.1	4.1	264	20	244	16.8
110	3.2	7.6	4.4	351	20	331	13.3

11b. Comparison with Alpert's theory

Test	$q^{-1/3}$ ($W^{-1/3}$)	h_c^* (-)	r/H (-)	f measured (-)	f Alpert (-)	C_1 (-)
104	0.0183	0.004	0.67	0.005	0.02 - 0.04	17
105	0.0176	0.013	0.67	0.015	0.02 - 0.04	51
106	0.0170	0.015	0.67	0.017	0.02 - 0.04	61
107	0.0183	0.011	0.58	0.012	0.02 - 0.04	42
110	0.0164	0.008	0.58	0.008	0.02 - 0.04	28

11c. Comparison with Zukoski's theory

Test	h_c^* measured (-)	h_c^* Zukoski (-)
104	0.004	0.012 - 0.021
105	0.013	0.012 - 0.021
106	0.015	0.012 - 0.021
107	0.011	0.012 - 0.020
110	0.008	0.012 - 0.020

11d. Comparison with flat plate values

Test	h_c measured (W/m^2-K)	h_c flat plate (W/m^2-K)
104	6.1	9.1
105	20.0	9.5
106	24.6	9.7
107	16.8	9.6
110	13.3	10.1

Table 12. Heat losses through room surfaces

Test	Ceiling losses						Wall losses							
	$\dot{q}''_{cw}$ (kW/m^2)	$\dot{q}''_{rw}$ (kW/m^2)	$\dot{q}''_{w}$ (kW/m^2)	T_w (K)	ΔT (K)	$\dot{q}_{ceiling}$ (kW)	$\dot{q}''_{cw}$ (kW/m^2)	$\dot{q}''_{rw}$ (kW/m^2)	$\dot{q}''_{w}$ (kW/m^2)	T_w (K)	ΔT (K)	$\dot{q}_{walls}$ (kW)	$\dot{q}_{total}$ (kW)	$\frac{\dot{q}_{total}}{\dot{q}_{fuel}}$
104	0.08	0.10	0.18	311	18	2.1	0.09	0.12	0.21	314	21	3.4	5.5	0.03
105	0.18	0.20	0.38	326	33	4.5	0.19	0.24	0.37	331	38	6.9	11.4	0.06
106	0.19	0.21	0.40	327	34	4.8	0.20	0.25	0.39	333	40	7.2	12.0	0.06
107	0.12	0.14	0.26	317	24	3.1	0.11	0.15	0.22	318	25	4.2	7.3	0.04
110	0.18	0.20	0.38	326	33	4.5	0.19	0.23	0.37	330	37	6.7	11.2	0.05

Table 13. Thermal resistance values

Test	Surface	$\dot{q}''_w$ (W/m^2)	T_s (°C)	T_w (°C)	R measured (m^2-K/W)	R calculated (m^2-K/W)	Difference (m^2-K/W)
104	Ceiling	180	237	38	1.10	0.17	0.93
	Walls	210	198	41	0.75	0.17	0.58
105	Ceiling	380	305	53	0.66	0.17	0.49
	Walls	370	248	58	0.51	0.17	0.34
106	Ceiling	400	328	54	0.69	0.17	0.52
	Walls	390	266	60	0.53	0.17	0.36
107	Ceiling	260	339	44	1.13	0.17	0.96
	Walls	220	207	45	0.74	0.17	0.57
110	Ceiling	380	335	53	0.74	0.17	0.57
	Walls	370	274	57	0.59	0.17	0.42

Table 14. Heat balance

Test	$\dot{m}_f$ (kg/s)	Δh_c (MJ/kg)	Heat release $\dot{m}_f \Delta h_c$ (kW)	$\dot{m}_o$ (kg/s)	ΔT (K)	Losses: $\dot{m}_o C_p \Delta T$ (kW)	Wall surface losses (kW)	Doorway radiation (kW)	Total (kW)	Error (%)
102	0.0035	52.6	184	0.954	220	210	5.5	3.2	219	-17
104	0.0032	52.6	164	1.038	186	193	11.4	2.0	206	-23
105	0.0035	52.6	184	0.979	233	228	12.0	3.2	243	-28
106	0.0039	52.6	205	1.010	246	248	7.3	3.8	259	-23
107	0.0032	52.6	164	0.938	187	175	11.2	2.2	188	-14

Table 15. Influence of after-burner for test 101 conditions

Channel	Temperature (°C) without after-burner	Temperature (°C) with after-burner
01	314	316
02	325	364
03	648	702
06	250	239
07	81	74
08	61	56
09	238	230
10	234	226
11	63	57
12	56	51
13	252	241
14	242	230
15	92	83
16	70	64
17	237	228
19	82	74
20	64	57
21	264	255
22	273	264
24	247	238
26	158	140
27	75	70
28	80	75
29	70	64
36	226	214
37	100	92
38	31	27
40	53	48
41	58	53
42	37	34
43	220	210
44	63	60
45	242	235
46	323	308
48	286	276
49	62	58
50	242	230
51	250	240
52	254	245
53	263	255
54	40	37
55	282	269
56	248	240
57	262	254
58	249	239
59	229	220
60	109	95
61	61	60
62	44	40
63	41	36
67	126	107
68	42	37
69	33	27
70	30	28
71	29	26
72	28	23
73	26	22
74	107	99
75	76	70
76	75	64
77	88	79
78	95	88
79	83	75
80	75	80
81	52	50
82	254	244
83	253	238
84	245	238
85	224	200
86	58	72
87	46	42
88	39	33
90	207	199
91	53	49

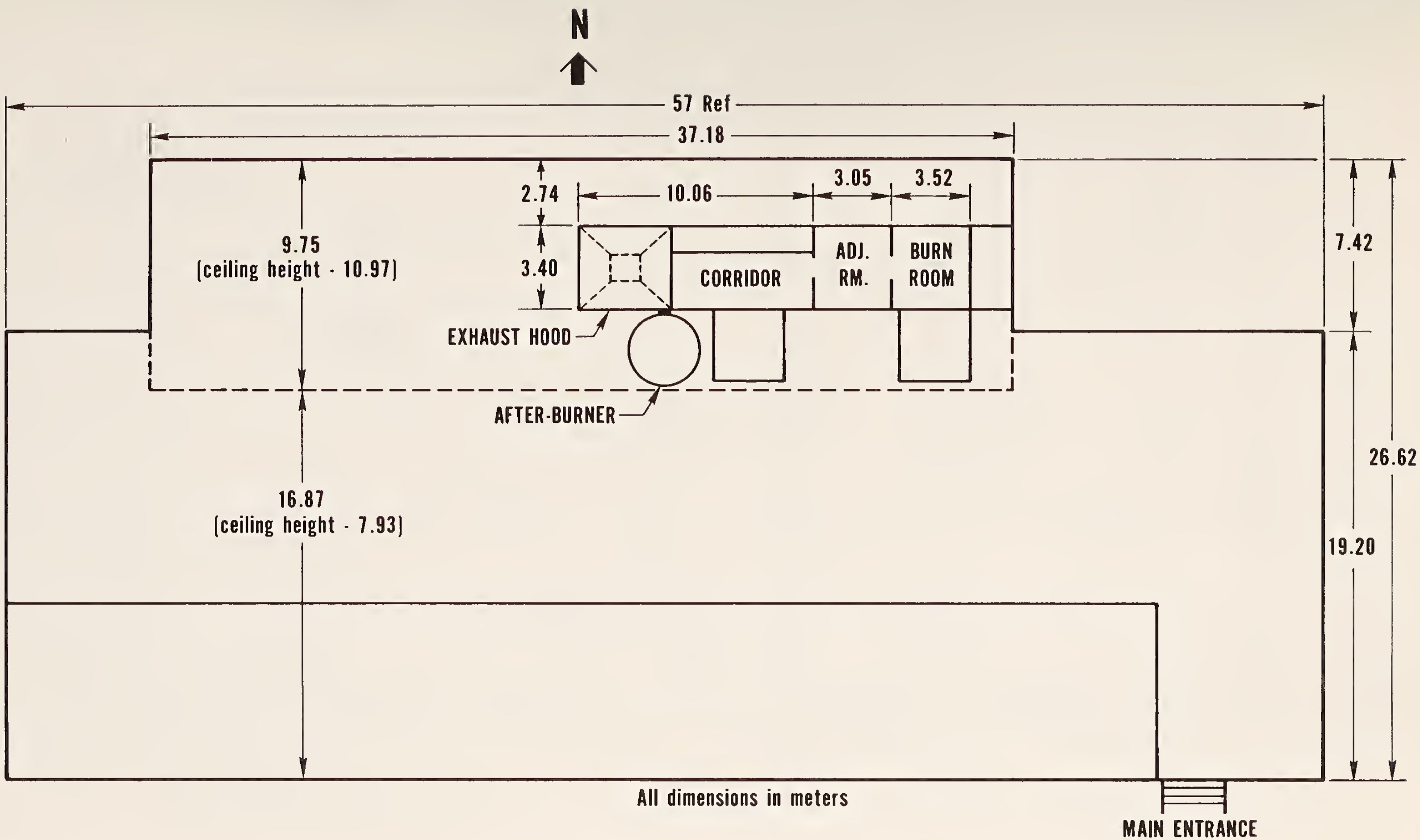

Figure 1. Plan view of test facility

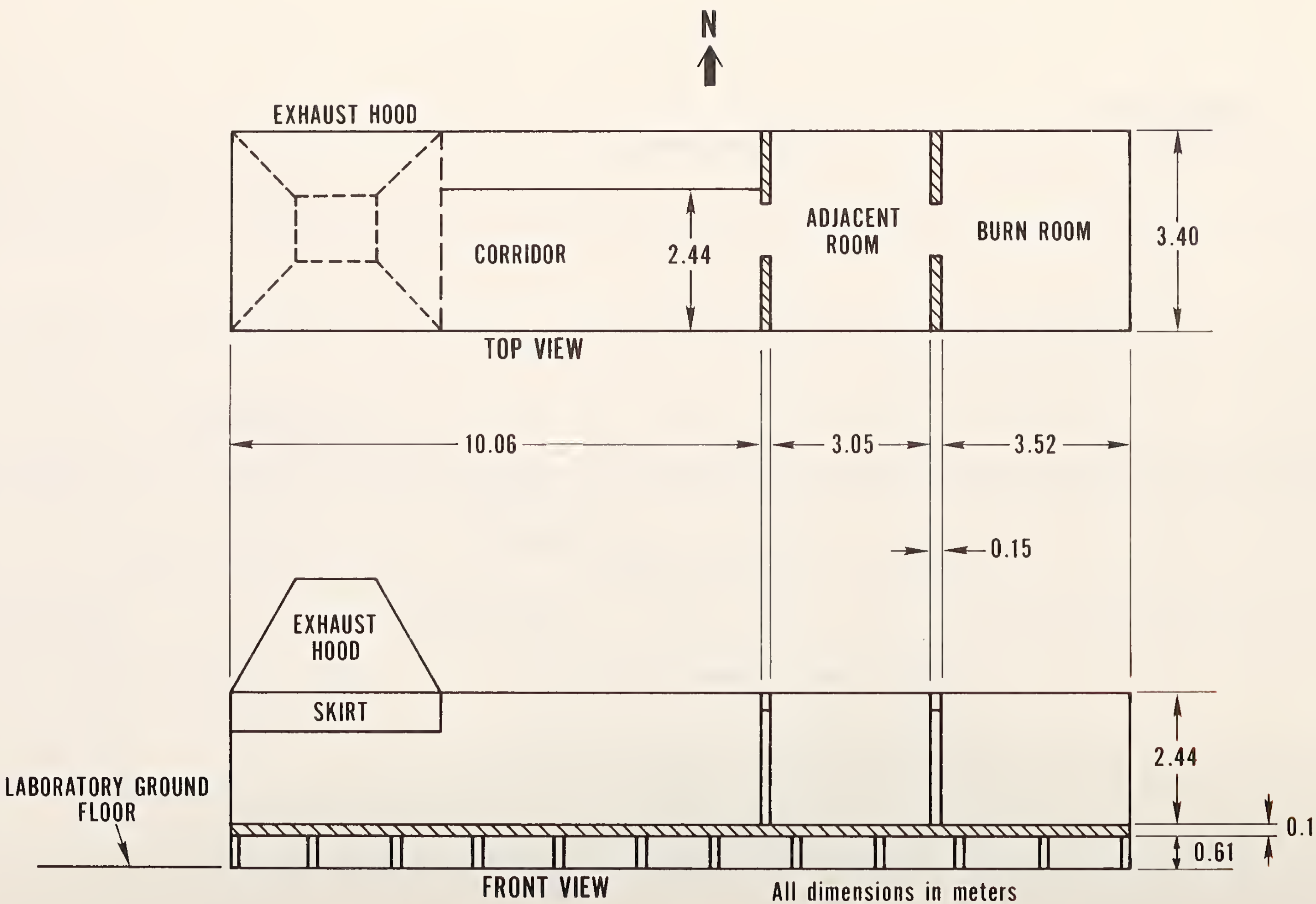

Figure 2. General view of room-corridor complex

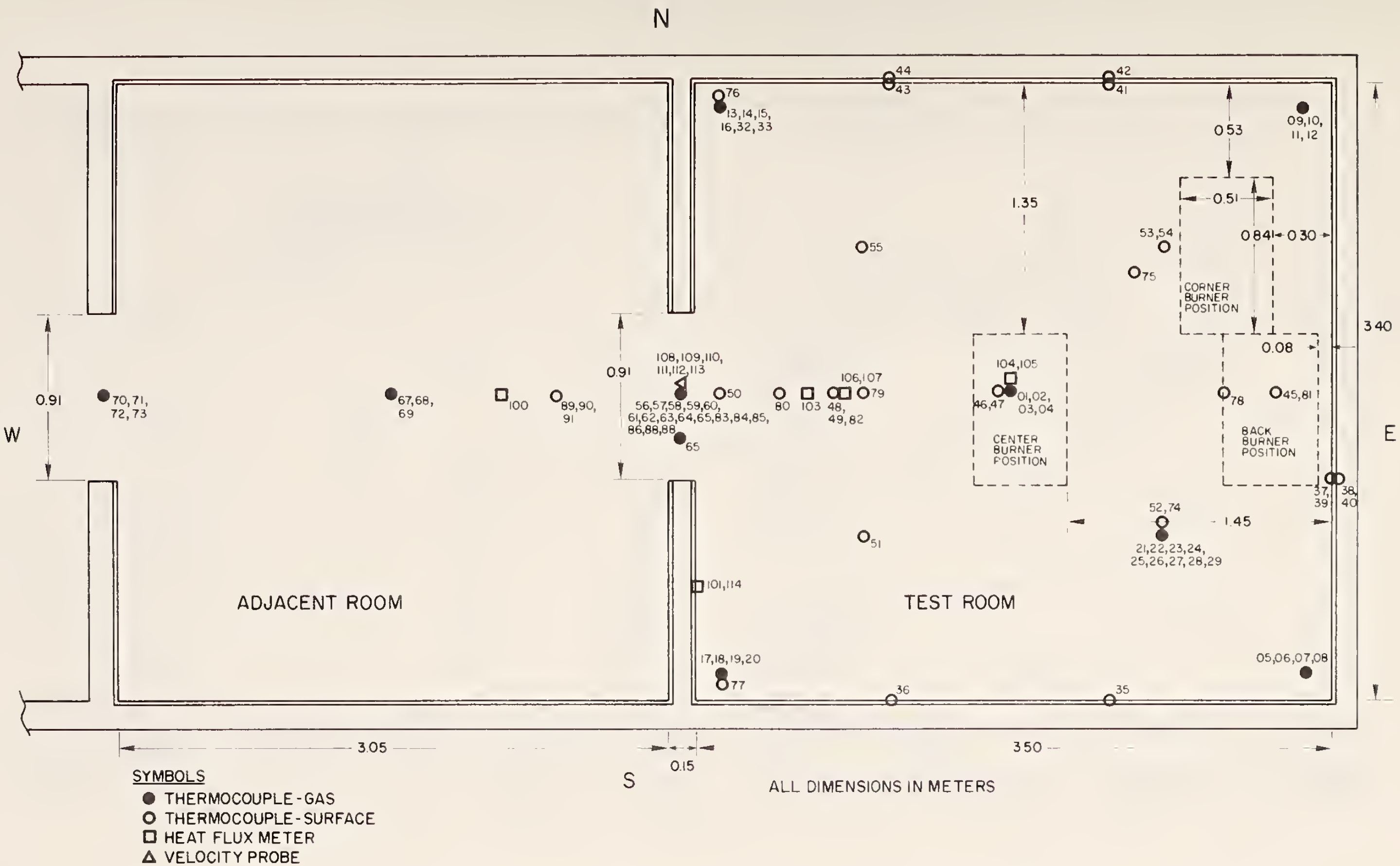

Figure 3. Plan view of room-corridor complex

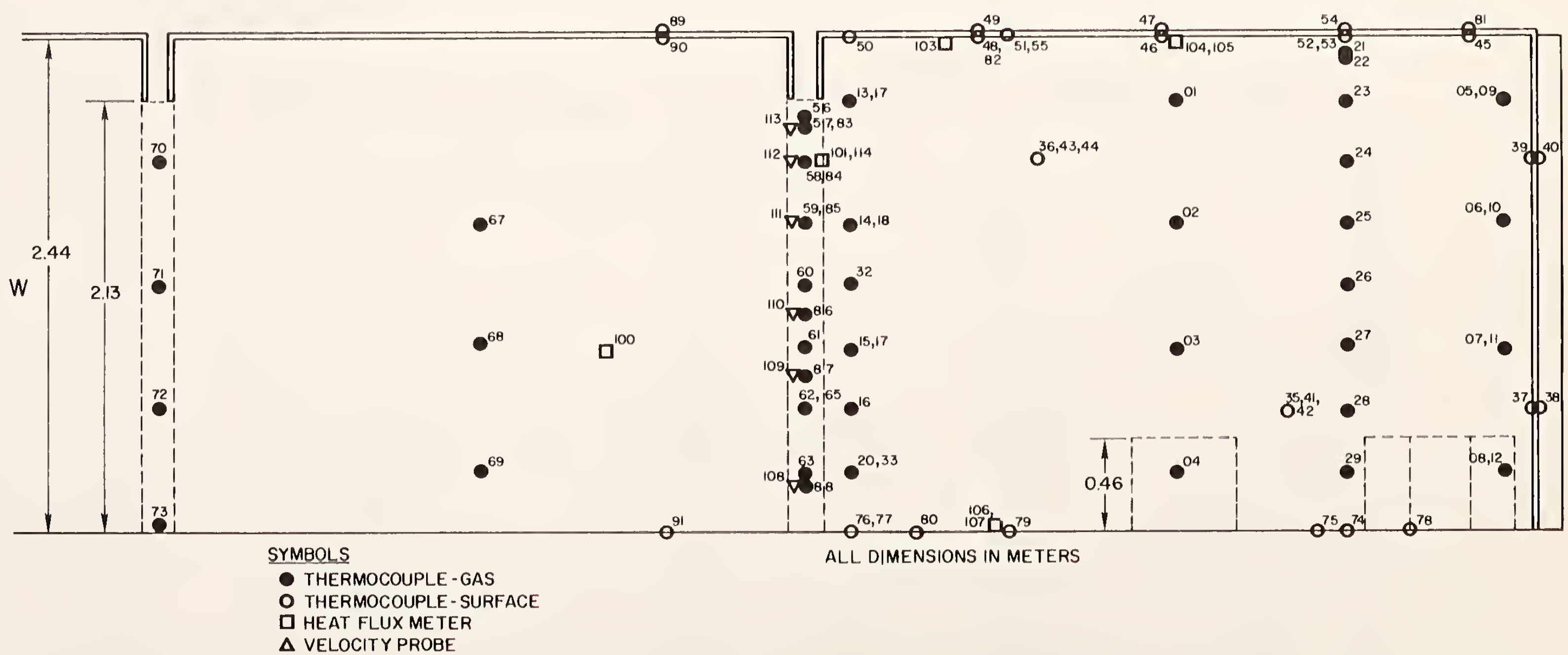

Figure 4. Elevation of room-corridor complex

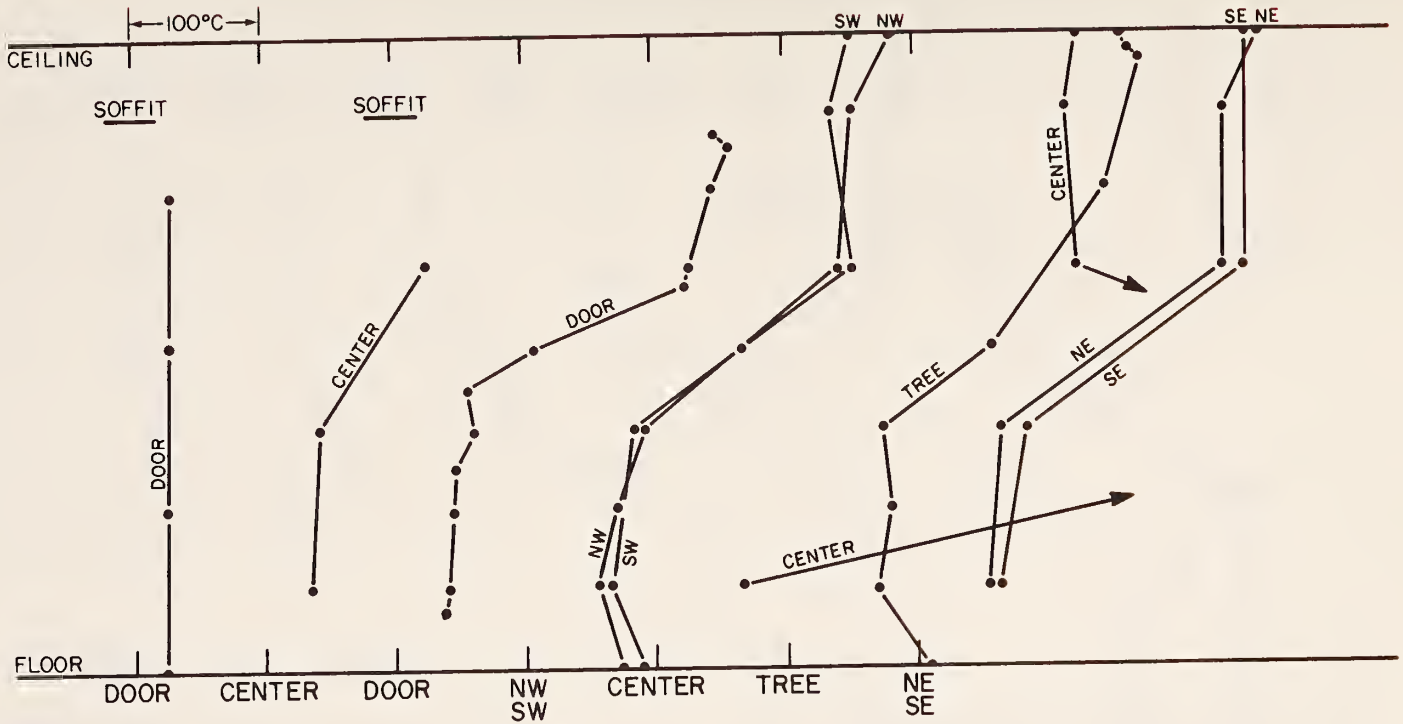

Figure 5. Vertical temperature profiles for test 101

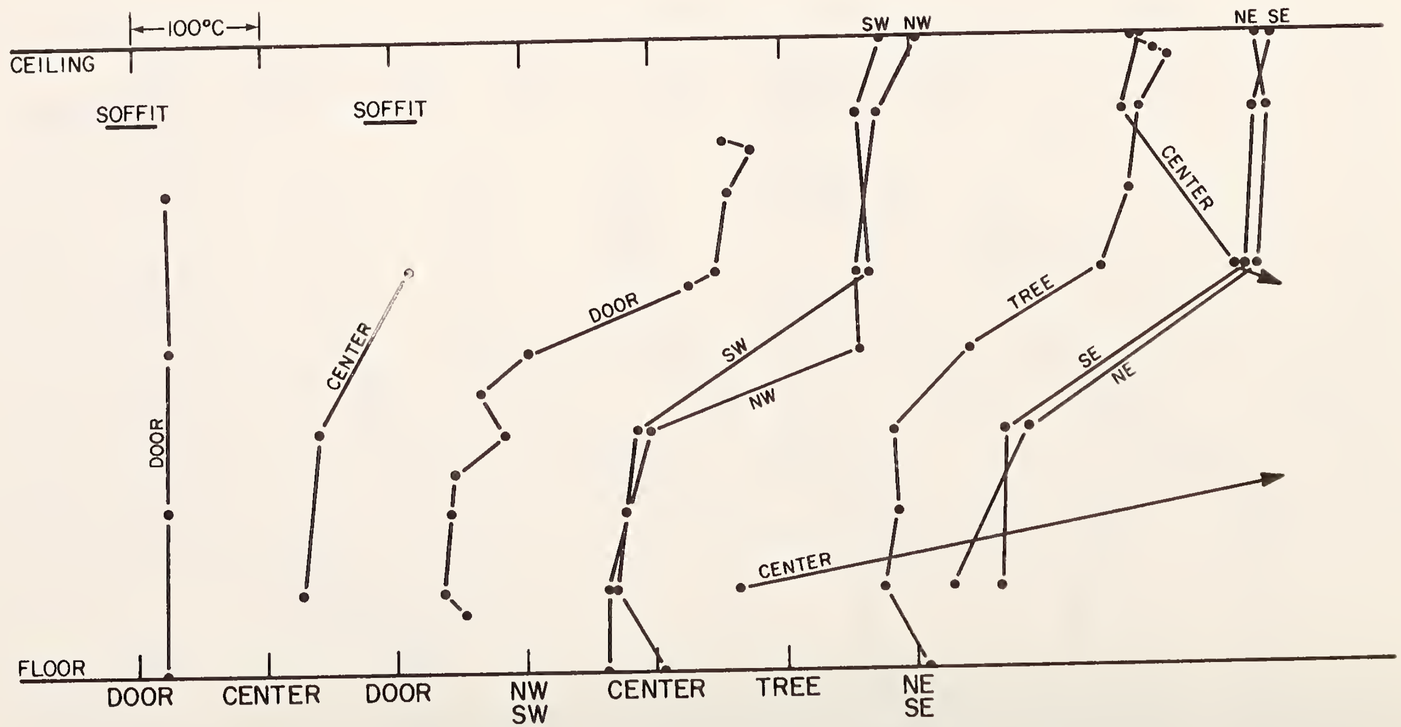

Figure 6. Vertical temperature profiles for test 102

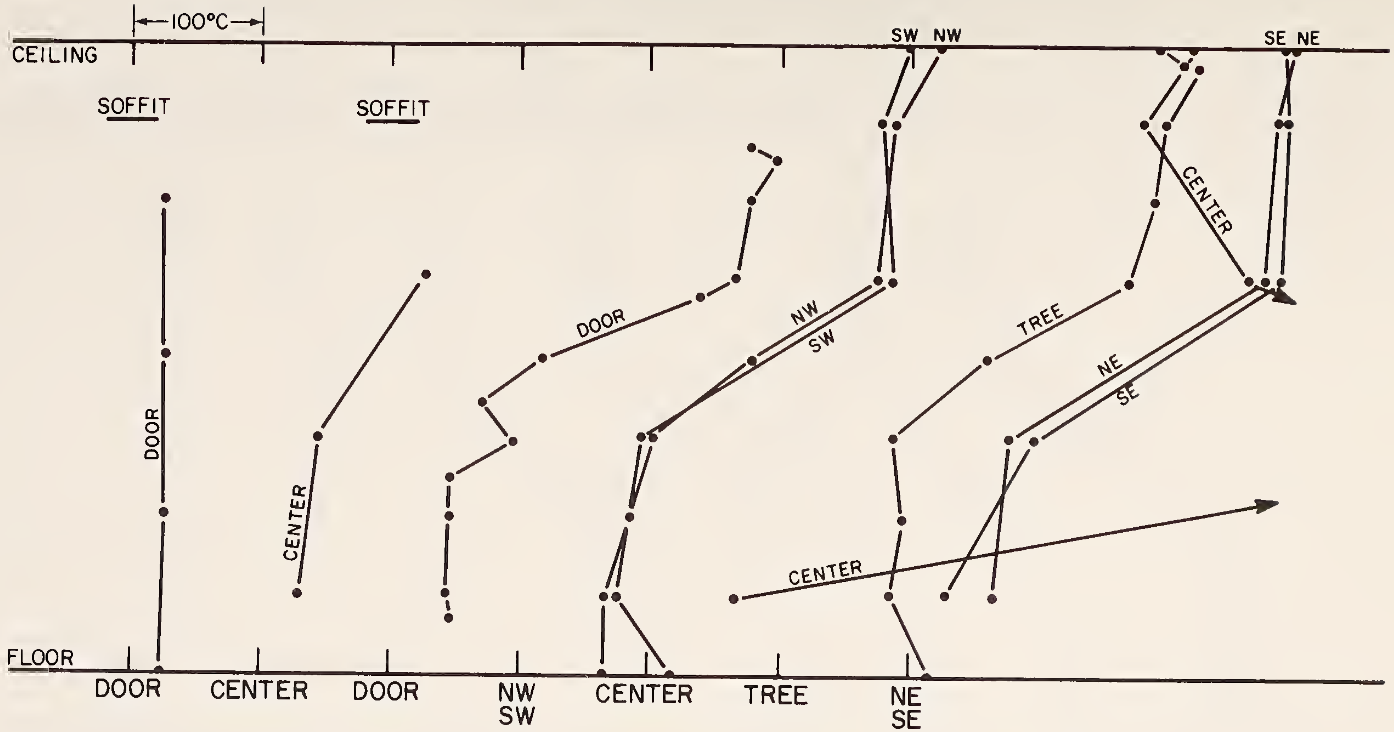

Figure 7. Vertical temperature profiles for test 103

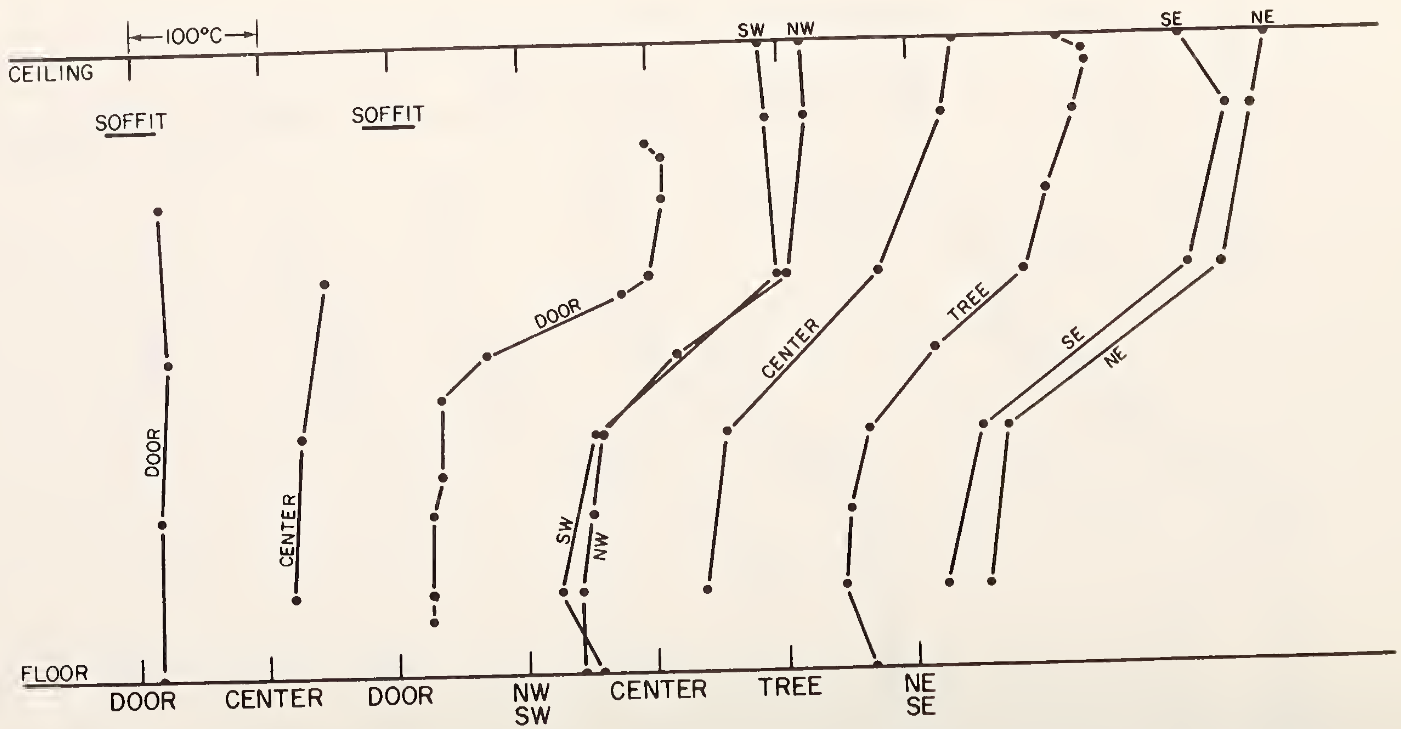

Figure 8. Vertical temperature profiles for test 104

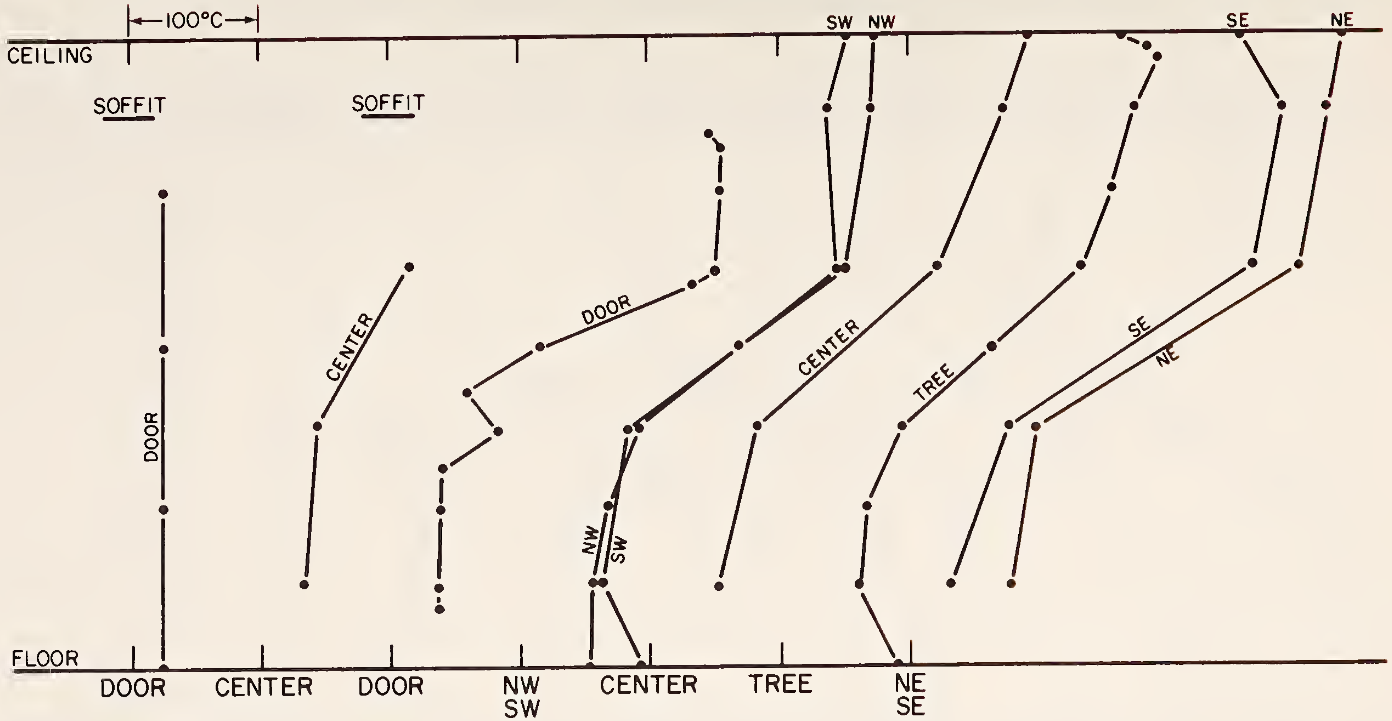

Figure 9. Vertical temperature profiles for test 105

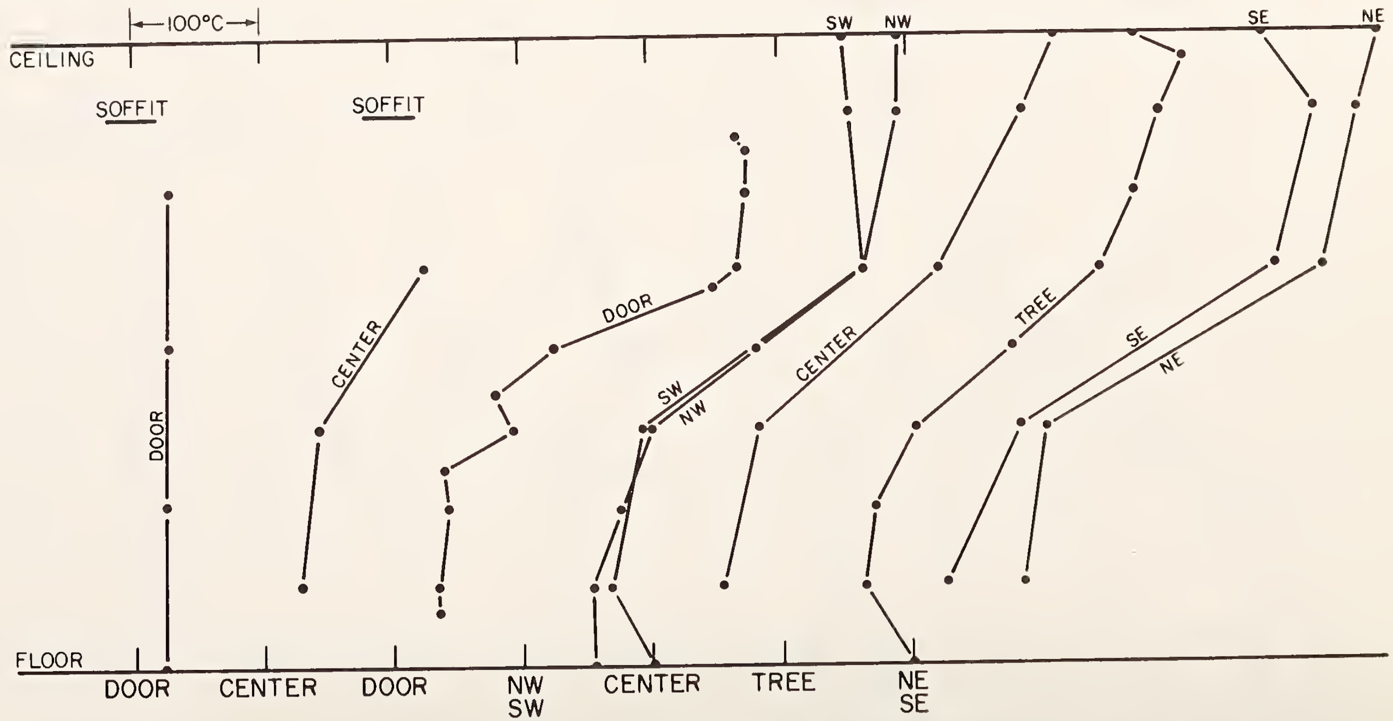

Figure 10. Vertical temperature profiles for test 106

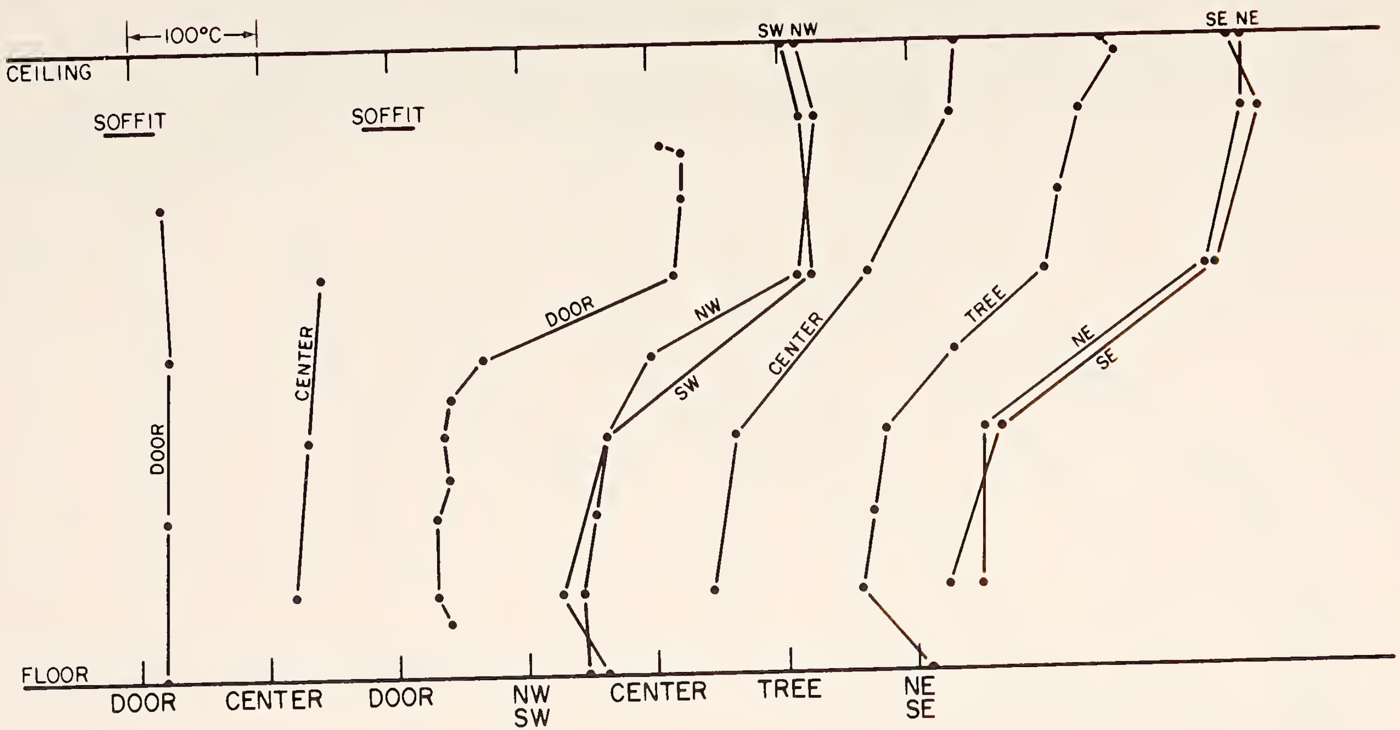

Figure 11. Vertical temperature profiles for test 107

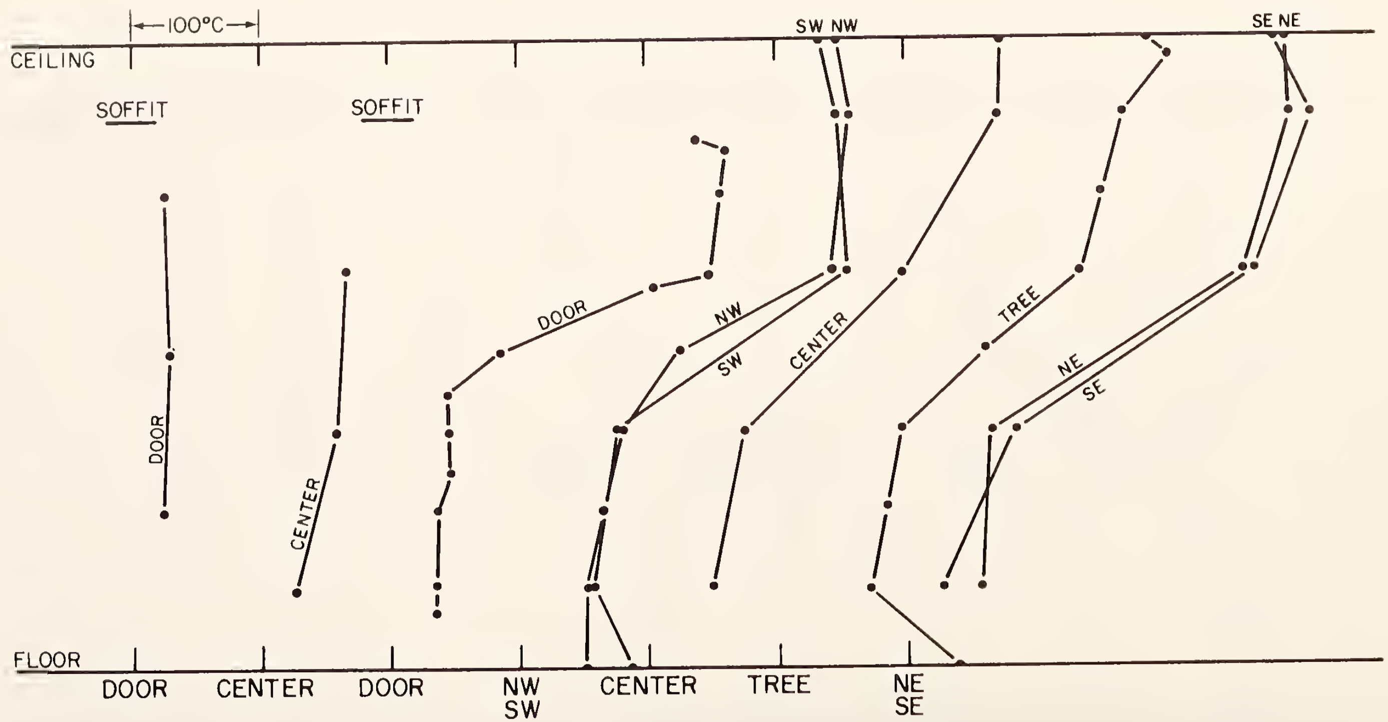

Figure 12. Vertical temperature profiles for test 108

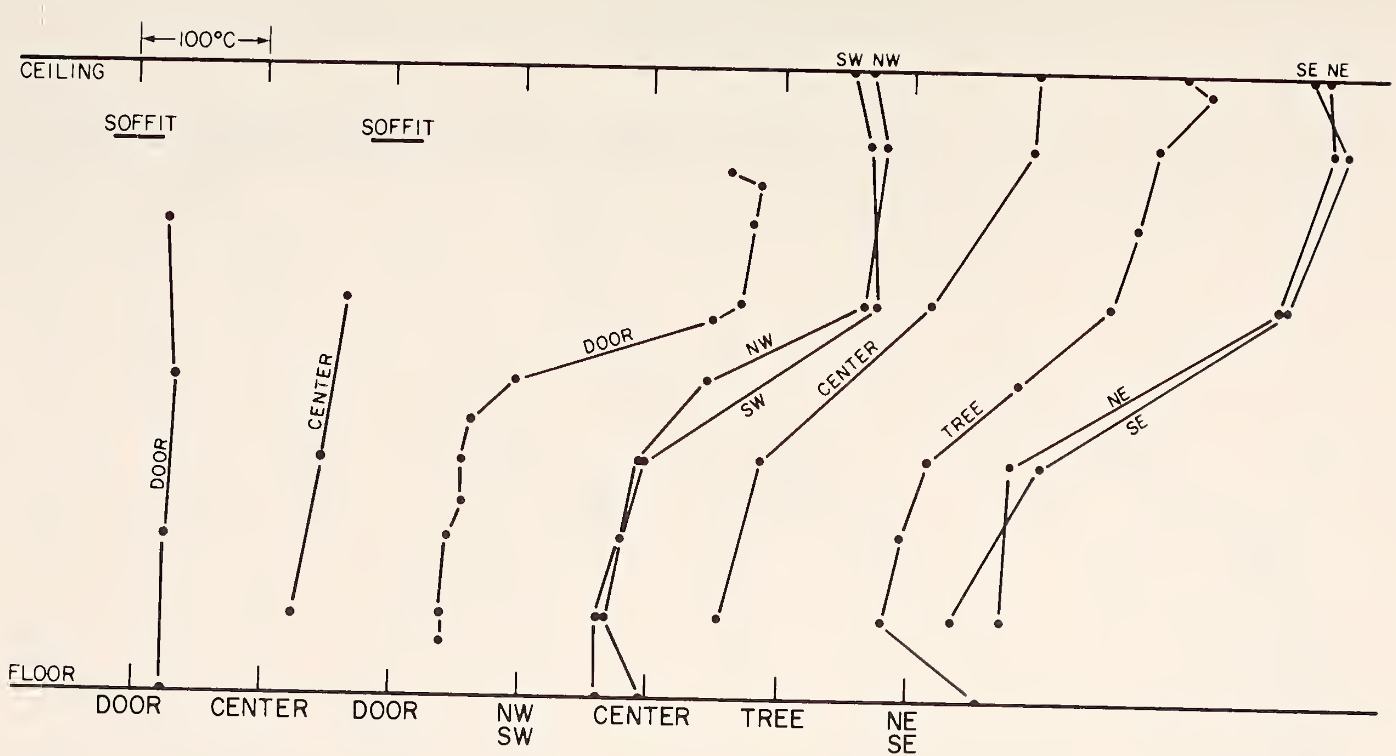

Figure 13. Vertical temperature profiles for test 109

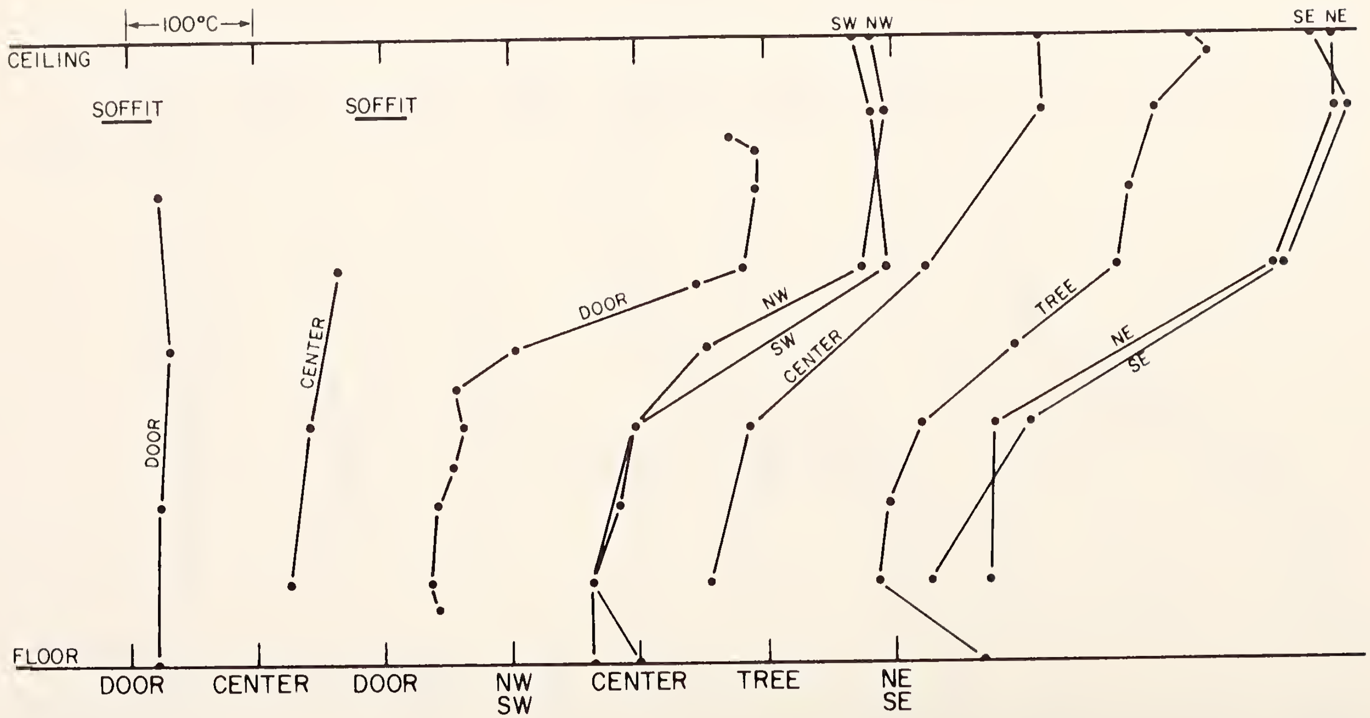

Figure 14. Vertical temperature profiles for test 110

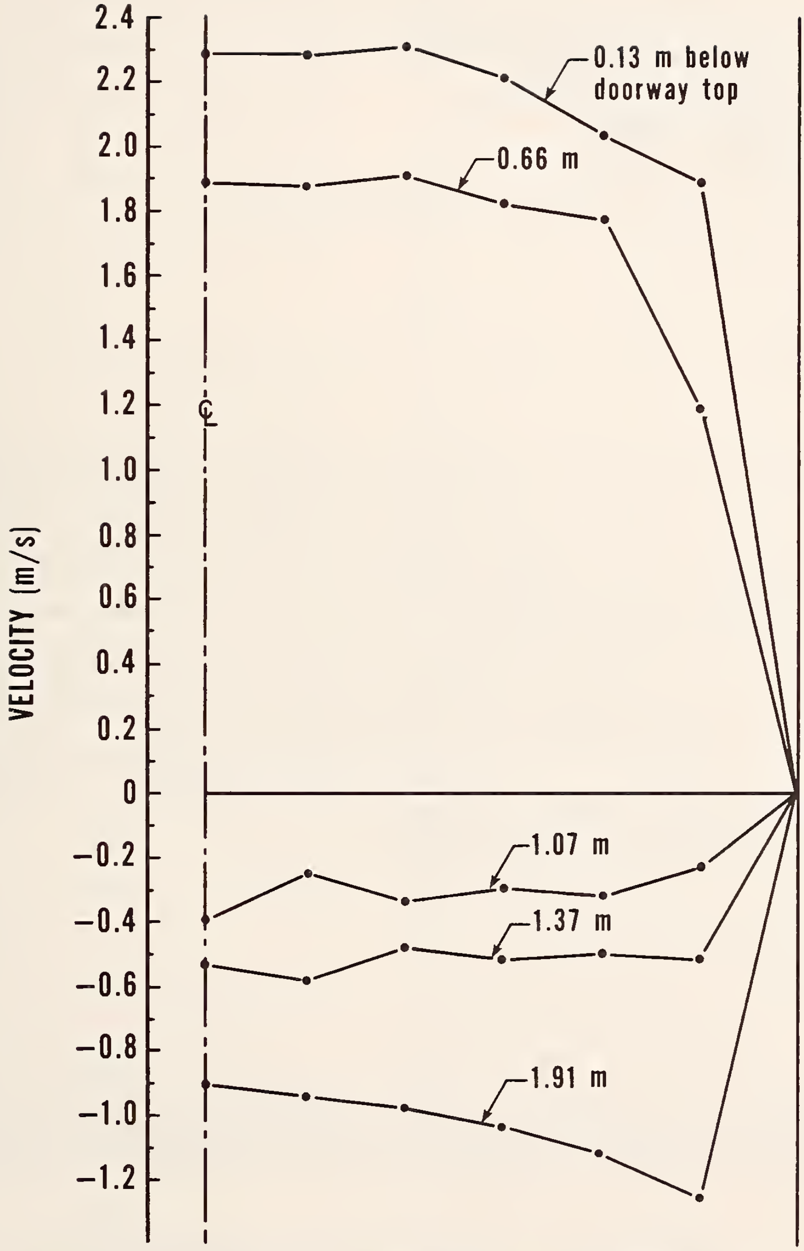

Figure 15. Typical doorway velocity distribution

NBS-114A (REV. 7-73)

U.S. DEPT. OF COMM. **BIBLIOGRAPHIC DATA SHEET**	1. PUBLICATION OR REPORT NO. NBS TN 981	2. Gov't Accession No.	3. Recipient's Accession No.
4. TITLE AND SUBTITLE THE CALIBRATION OF A BURN ROOM FOR FIRE TESTS ON FURNISHINGS			5. Publication Date December 1978 6. Performing Organization Code
7. AUTHOR(S) King-Mon Tu and Vytenis Babrauskas			8. Performing Organ. Report No.
9. PERFORMING ORGANIZATION NAME AND ADDRESS NATIONAL BUREAU OF STANDARDS DEPARTMENT OF COMMERCE WASHINGTON, D.C. 20234			10. Project/Task/Work Unit No. 7527677 11. Contract/Grant No.
12. Sponsoring Organization Name and Complete Address *(Street, City, State, ZIP)* Same as No. 9			13. Type of Report & Period Covered Final 14. Sponsoring Agency Code
15. SUPPLEMENTARY NOTES			

16. 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 series of ten tests, using a diffusion flame gas burner as heat source, was conducted in a full-size room designed for furnishings flammability tests. The gas burner was used to release known heat rates and to permit steady state measurements of energy and mass flow. The gas burner fires simulated preflash-over conditions, with peak temperatures outside the burner plume of around 300°C. The measurements obtained were compared with available theoretical room fire descriptions and published heat transfer values. The results showed the importance of a precise determination of the inflow and exhaust velocities at the doorway. It was demonstrated that a large number of doorway velocity probes is required to accurately obtain a room heat and mass balance. A calculational procedure was developed for analyzing the results which should be useful for future analysis of furnishings experiments.

17. KEY WORDS *(six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons)* Buoyant plumes; convection; fire tests; flammability; furnishings; heat transfer; radiation.

18. AVAILABILITY [X] Unlimited	19. SECURITY CLASS (THIS REPORT) UNCLASSIFIED	21. NO. OF PAGES 59
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USCOMM-DC 29042-P74

☆ U. S. GOVERNMENT PRINTING OFFICE : 1978 281-067/324

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