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An Assessment of Correlations Between Laboratory and Full-Scale Experiments for the FAA Aircraft Fire Safety Program, Part 6: Reduced-Scale Modeling of Compartments at Atmospheric Pressure

U.S. DEPARTMENT OF COMMERCE
National Bureau of Standards
National Engineering Laboratory
Center for Fire Research
Washington, DC 20234

March 1983

Sponsored by:

U.S. Department of Transportation
Federal Aviation Administration
Technical Center
Atlantic City Airport, NJ 08405

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AN ASSESSEMENT OF CORRELATIONS BETWEEN LABORATORY AND FULL-SCALE EXPERIMENTS FOR THE FAA AIRCRAFT FIRE SAFETY PROGRAM, PART 6: REDUCED-SCALE MODELING OF COMPARTMENTS AT ATMOSPHERIC PRESSURE

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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	Flame heat transfer coefficient $\dot{q}''_L/T^*$ (kW/m$^2\cdot$K)
A	Area of room (m^2)
A_w	Area of wall (m^2)
A_c	Area of ceiling (m^2)
A_f	Area of flame (m^2)
A_{fl}	Area of floor (m^2)
B	Expansion coefficient of gases - 1/T (K^{-1})
C_p	Heat capacity of air (kJ/kg$\cdot$K)
C_{p_s}	Heat capacity of room lining material (kJ/kg$\cdot$K)
D	Diameter of pool (m)
E	Heat release rate per unit mass of oxygen consumed (13.1 MJ/kg)
f	Function Dependence
Fr	Froude number - u^2/gD (dimensionless)
g	Acceleration due to gravity (9.8 m/s^2)
h	Height of doorway (m)
h_1	Distance from top of doorway down to the neutral pressure plane (m)
H	Height of room (m)
H_c	Heat of combustion of fuel (kJ/kg)
K	thermal conductivity of the wall (kW/m$\cdot$K)
ℓ	Flame height above the pool (m)
L	Length of room (m)
$\dot{m}'_a$	Mass flow rate per unit width of the air in the boundary layer (kg/m$\cdot$s)
$\dot{q}''_c$	Heat flux per unit area measured at the center of the ceiling (kW/m^2)
$\dot{q}''_f$	Heat release rate per unit area in the flame (kW/m^2)
$\dot{q}''_{f_1}$	$\dot{q}''_f$ for full scale room
$\dot{q}''_L$	Heat loss rate per unit area of the flame (kW/m^2)
$\bar{\dot{q}}''_c$	Average heat release rate per unit area of ceiling (kW/m^2)

$\bar{\dot{q}}''_w$	Average heat release rate per unit area of wall (kW/m^2)
$\dot{Q}$	Total rate of heat release (kW)
$\dot{Q}_B$	Rate of heat release from burner (kW)
$\dot{Q}'$	Rate of heat release per unit width of flame (kW/m)
r	ratio of the mass of oxygen to the mass of fuel consumed in complete combustion
S	Scale factor
T	Temperature (K)
T_{ad}	Adiabatic flame temperature (K)
T_∞	Free stream temperature in room and ambient temperature surrounding the pool fire (K)
T^*	$T - T_\infty$ (K)
T^*_{ad}	$T_{ad} - T_\infty$ (K)
T_c	Ceiling temperature (K)
T^*_1	T^* for full-scale test (K)
u	Velocity in boundary layer or in plume above pool (m/s)
u_∞	Free stream velocity (m/s)
u_w	Wind velocity (m/s)
$\dot{V}$	Volumetric air flow rate (m^3/s)
w	Width of door (m)
W	Width of room (m)
x	Distance along surface (m)
x_f	Flame length in room (m)
y_{O_2}	Mass fraction of oxygen in the free stream
y	Distance from surface (m)
M.C.	Thickness of lining material

ΔT Temperature rise of flame just above the pool (K)

δ Boundary layer thickness (m)

ϕ $\dot{Q}^2/gD^5B\Delta T$

λ Heat loss rate to room surfaces divided by upper air temperature rise ($kW/m^2 \cdot K$)

ν Kinematic viscosity (m^2/s)

ρ Density of the hot gas (kg/m^3)

ρ_∞ Density of air at ambient temperature or in the free stream (kg/m^3)

ρ_s Density of room lining material (kg/m^3)

σ Stefan Boltzmann constant (5.67×10^{-11} kW/m^2T^4)

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Abstract

The temperatures, heat fluxes, air velocities, and times to flashover are compared between a number of previously reported full- and reduced-scale room fire tests. The model tests are usually similar but somewhat less severe than their full-scale counterparts. A simplified analysis is presented to account for the lower temperatures observed in the models. Some recommendations are made with regard to physical modeling of the aircraft postcrash fire.

Key Words: Aircraft fires; fire tests; flashover; reduced-scale model; room fire tests

1. INTRODUCTION

The large quantities of aviation fuel carried aboard commercial aircraft have the potential of causing large pool fires after an otherwise survivable crash. Thermal radiation and the ingestion of flames through the open doorways can quickly lead to the buildup of untenable conditions in the aircraft cabin. The time available for evacuation depends on wind conditions, the size and strength of the fire, the geometry of the cabin and its openings in relation to the fire, and the materials comprising the seats and lining materials of the cabin, as well as miscellaneous items brought aboard by the passengers. Ideally, these parameters should be identified and formulated into a mathematical model which could calculate an escape time. Although a considerable amount of effort has been put into the development of mathematical models for room fires [1][1], and to a lesser extent for aircraft cabin fires [2,3,4], such an analytical model is still a few years away. At the present time, we must rely on full-scale tests to provide the needed information. These are costly to run and are particularly difficult in the case of ambient wind conditions which are difficult to achieve under controlled conditions in a test bay large enough to house the fuselage of a wide-body aircraft. Tests outside are at the mercy of the short-time variability of the winds. It is natural to inquire whether reduced-scale models can be used to reduce the test to manageable size where wind velocities can be controlled and many tests can be economically run to explore the above variables.

There are two reasons to run reduced-scale tests. The first is to predict what will happen in full-scale, and the second is to provide an experimental basis for developing and checking out the analytical models.

The most exact form of physical modeling is pressure modeling [5]. If all of the lengths are scaled as the negative 2/3 power of the pressure, both the Reynolds number and the Froude number are preserved. In that case the Navier Stokes equations take on the same dimensionless form for all scales. If radiation can be neglected or can be assumed to be proportional to the burning rate and if the fuel can be assumed to be a simple evaporating solid such as Polymethylmethacrylate, the heat and mass transfer will also scale properly. However the presence of radiation and char forming solids in real room fire situations has caused some problems with pressure modeling. A pressure chamber required to model a compartment opening into a large air supply plenum would be expensive.

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

Furthermore the requirement that the thickness of the lining material be reduced by the negative 2/3 power of the pressure would make it costly to test compartments with composite lining materials such as laminates and honeycombs which would have to be specially fabricated for the test. Pressure modeling will not be discussed further in this report, but the reader is referred to the work of Alpert et al. [6] for more information.

Atmospheric modeling though less exact provides a less expensive alternative. A large fire cannot be scaled down exactly at atmospheric pressure because it is impossible to maintain both Reynolds and Froude numbers constant. The atmospheric models attempt to reproduce the same temperature in the model as in the full scale compartment by requiring that the total rate of heat production, the total rate of heat transport by convection through the doorway, and the total heat loss to the compartment lining materials all be reduced in the same proportion.

Waterman at the Illinois Institute of Technology Research Institute (IITRI) [7] examined the use of atmospheric modeling of the fire buildup in rooms with upholstered chairs and couches and with noncombustible walls and ceiling. These rooms had an opening in one wall. In order to have the burning items or their gas simulators occupy the same relative area and have the same burning rate per unit area in the model as in the prototype, it was necessary to require that the total rate of heat release be proportional to the square of the scale factor (S^2). Since the ventilation is proportional to $wh^{3/2}$, where w is the width of the doorway and h is the doorway height, Kawagoe [8], it is necessary that $wh^{3/2} \sim S^2$ so that the air flow is reduced in the same proportion as the heat release rate. Based on the testing of various alternatives Waterman chose to make $w \sim S$ so that $h \sim S^{3/2}$. Because of the relatively small spacing above the door it was necessary to make $H \sim h \sim S^{3/2}$ where H is the height of the room. The following scaling rules were then applied in the study of flashover conditions at IITRI.

$$L \sim W \sim S$$
$$H \sim S^{3/2}$$
$$w \sim S$$
$$h \sim S^{3/2}$$
$$\text{fuel supply} \sim S^2$$

where L, W, and H are the length, width, and height of the room while w and h are the width and height of the opening.

While scale factors as small as one-eighth worked satisfactorily, it was recommended that a lower limit of 0.6 m (2 ft) be used for the height of the model room. The comparison of the temperatures, heat fluxes, and gas concentrations between the full-scale and model tests was impressive using only propane gas burners as the fuel source. This type of scaling would be expected to work satisfactorily for combustible furnishings but not for combustible walls, since the wall area is proportional to the scale to the 5/3 power rather than the square of the scale factor as specified for the fuel input.

The reduced-scale modeling at the Factory Mutual Research Corporation (FMRC) by Heskestad at atmospheric pressure has used geometrical modeling of both the room and doorway dimensions [9]. Since the air flow rate is proportional to $wh^{3/2}$ or $S^{5/2}$ where S is the scale factor, the total heat release rate, $\dot{Q}$, is made proportional to $S^{5/2}$. By making $\dot{Q} \sim S^{5/2}$ the flame height is properly scaled. The geometrical modeling insures that the Froude number, u^2/gL, remains constant since $u \sim \sqrt{h} \sim \sqrt{s}$. However, since the area of the burning items must be proportional to $S^{5/2}$, this scaling procedure is not applicable to burning wall and ceiling surfaces which are proportional S^2.

The comparisons between the reduced- and full-scale room fire tests for combustible linings described in this report are based on a set of scaling principles, originally rejected at IITRI for furniture fires, but used at National Bureau Standards (NBS) because they have been found to be the best suited when combustible walls are involved. These scaling rules will be described in the next section. The strengths and weaknesses of this scaling procedure are brought out in the report. The emphasis will be on NBS studies of fire development in compartments.

2. SCALING RULES

The quarter-scale model room used in these experiments was based on a set of scaling rules which attempt to produce the same average gas temperature in the upper part of the room as the full-scale test [10]. These rules are derived from a quasi-equilibrium energy balance where the rate of heat production, $\dot{Q}_B$, by a burner located in the room plus the rate of heat production of the walls and ceiling is equal to the heat losses through the lining materials of the room and that radiated through the doorway plus the heat carried out of the doorway by the hot gases. This balance can be written

$$\dot{Q}_B + \bar{\dot{q}}''_w A_w + \bar{\dot{q}}''_c A_c = \lambda(T-T_\infty) + \rho C_p \dot{V} (T-T_\infty) , \qquad (2\text{-}1)$$

where $\bar{\dot{q}}''_w$ and $\bar{\dot{q}}''_c$ are the average heat release rates of the wall and ceiling and A_w and A_c are the respective areas of involvement; T and T_∞ are the absolute temperatures of the air in the upper part of the room and the ambient air, respectively; λ is the total rate of heat loss by conduction through the walls and ceiling and by radiation to the lower part of the room or through the doorway divided by the temperature rise, $T-T_\infty$; and ρ, C_p, and $\dot{V}$ are the density, heat capacity, and volumetric flow rate respectively of the exhaust air. The temperature rise is, therefore, given by

$$T-T_\infty = \frac{\dot{Q}_B + \bar{\dot{q}}''_w A_w + \bar{\dot{q}}''_c A_c}{\rho C_p \dot{V} + \lambda} . \qquad (2\text{-}2)$$

The scaling rules for the model are obtained by dividing the terms in the numerator and denominator of the right-hand side by the floor area, A_{fl}, so that

$$T-T_\infty = \frac{\left(\frac{\dot{Q}_B}{A_{fl}}\right) + \bar{\dot{q}}''_w \left(\frac{A_w}{A_{fl}}\right) + \bar{\dot{q}}''_c \left(\frac{A_c}{A_{fl}}\right)}{\rho C_p \left(\frac{\dot{V}}{A_{fl}}\right) + \left(\frac{\lambda}{A_{fl}}\right)} \quad . \tag{2-3}$$

By preserving the ratios in the brackets as the scale is reduced, the temperature rise, and, hence, the severity of the fire, remains the same. $(\dot{Q}_B/A_{fl})$ is maintained constant by keeping the gas flow to the burner proportional to the floor area, $\left(\frac{A_w}{A_{fl}}\right)$ and $\left(\frac{A_c}{A_{fl}}\right)$ are kept constant by using geometric scaling, and λ/A_{fl} also is kept constant by geometric scaling if the fires are similar and the heat transfer coefficients are the same in full- and reduced-scale.

Since $\dot{V} \propto wh^{3/2}$, where w is the width of the doorway and h is its height, $\dot{V}/A_{fl}$ can be maintained constant by making h proportional to the scale factor and making w proportional to the square root of the scale factor. Actually $\dot{V} \sim wh_1^{3/2}$, where h_1 is the distance from the top of the doorway down to the neutral pressure plane. However, if scaling is successful, h_1 is the same fraction of h in the two scales so that the above relationship holds. The wall above the doorway traps the hot combustion products from the fire and is critical to the phenomena taking place in the room, so that this height was chosen to be scaled geometrically. For quarter-scale modeling, the doorway width then is half of its full-scale value while the other dimensions are only one quarter, except for the thickness of the wall and ceiling materials. The wall and ceiling materials must be of the same thickness as in the prototype, to insure that the heat losses per unit area remain the same for the same interior air temperature. (Small differences in heat losses are expected since the convective heat transfer coefficient will be less in the model due to lower air velocities.) This is of great practical value since materials are tested in the thicknesses available in the marketplace, and composites do not pose an additional fabrication problem. This is summarized by the following set of scaling rules.

All Dimensions $\sim$ Scale Except

o Doorway Width $\sim$ $(\text{Scale})^{1/2}$

o Wall and Ceiling Thickness Same as Full Scale

Fuel Supply Rate $\sim$ Floor Area

Air Supply Rate $\sim$ Floor Area

Time Same as Full-Scale

However, the following problems are encountered with the scaling:

1. since the lateral flame spread rate does not change with scale, the area covered by flame is relatively too large in the model;

2. the flame heights are observed to be too high in the model;

3. the convective heat transfer coefficient is too low in the model since air velocities are proportional to the square root of the scale;

4. radiation from the upper part of the room is scale-dependent when the hot air layer is semi-transparent and a vertical temperature gradient exists within the layer;

5. flame radiation is quite scale dependent;

6. the increased size of the doorway opening required to scale the volumetric air flow rate permits slightly greater heat losses and slightly less heat release from combustible walls; and,

7. during the final approach to flashover, the volume expansion of the upper layer may account for a significant fraction of the outflow. This rate of expansion is proportional to the cube of the scale factor, whereas the buoyancy driven flow is proportional to the square of the scale factor according to the above rules.

Nevertheless, in many tests at NBS with different interior finish materials from low density foam plastics to high density cellulosics, both the maximum temperatures reached and the times to flashover* and flame out of the doorway for the full-scale and the model rooms have been similar, but usually somewhat longer times and lower temperatures have been observed in the model. Duplication of the full-scale tests have been reasonably good which encourages the study of the fire development phenomena on a more economical scale. Presently, some empirical adjustments to the scaling rules are being examined in an effort to achieve a higher level of agreement.

3. COMPARISON OF THE RESULTS OF FULL- AND REDUCED-SCALE ROOM FIRE TESTS WITH COMBUSTIBLE LINING MATERIALS

3.1 Room Corner Fire Tests at NBS

The first test at NBS of the modeling procedure described in section 2 was of a room corner test in which the wall in the rear corner of the full-scale room was formed by two 1.2 x 2.4 m (4 x 8 ft) panels of lauan plywood. These panels were 6.4 mm (11/64 in) thick. The remainder of the 2.9 x 3.1 x 2.4 m (9.5 x 10.5 x 7.9 ft) room was lined with 16 mm (5/8 in) thick Type X gypsum wallboard. There was a 0.90 m (35 in) wide by 2.0 m (80 in) high open doorway and a 6.4 kg (14 lb) wood crib in one rear corner as an ignition source. The full-scale test was part of a series [11] in which the wall and ceiling materials and the ignition source size were varied in order to compare the performance of materials in a room fire with their performance in various laboratory fire tests. This series of tests is reviewed in reference [12] as part of the correlation of ASTM E 84 with full-scale room fire tests. The quarter-scale model used a natural gas diffusion burner whose linear

*Flashover is defined here as the time at which the incident heat flux on the center of the floor reaches 20 kW/m^2 which is sufficient to ignite light combustibles.

dimensions were one-quarter of those of the wood crib and whose gas flow was adjusted to release heat at one-sixteenth of the average heat release rate of the wood crib. The effective heat of combustion of the wood during the burning phase was taken to be 15 MJ/kg (6500 Btu/lb). Figures 1 and 2 give a comparison of the air temperature and doorway velocities between the two scales. These figures give a good indication of the rough agreement that can be expected with this type of scaling.

3.2 Navy Berthing Compartment Fire Tests at NBS

Reduced-scale modeling was next applied to a Navy berthing compartment [13]. The prototype compartment fires were those conducted under the Navy's habitability program of making berthing spaces more comfortable for the crew without an accompanying significant increase in the fire risk potential.

The series of thirteen full-scale tests shown in Table 1 was conducted in the 3.1 x 3.1 x 2.1 m (10 x 10 x 7 ft) burnout room with a 0.7 x 1.9 m (27 x 75 in) doorway. The door was either closed, open, or partly open during each test. The contents of the compartment consisted of a three-man bunk with bedding and a three-man locker stuffed with cotton waste. The bedding included a neoprene mattress, cotton sheets, a wool blanket, and a chicken feather pillow. The lining materials and ventilation conditions were varied from test to test. Ignition was by 800 ml of ethyl alcohol in the middle of the lower bunk. The bedding materials were in considerable disarray in order to promote a rapid growth of the fire and thus produce the worst conditions.

Ten of these tests were duplicated in the small-scale model using a scaled-down bunk and locker. The bedding was reduced in area by a factor of 16. Scaling requires that the thicknesses stay the same. However, the 76 mm (3-in) thick neoprene mattress could barely be squeezed along with the other bedding into the space between the tiers of the bunk. This would have constricted the airflow by an unacceptable amount. A compromise was made by using a 25 mm (1-in) thick mattress in the model, feeling that it would be satisfactory at least in the early part of the test.

Table 1 shows the range of conditions covered by the tests. The first four have progressively increased ventilation. In all but the fourth test, the bunk had closed ends and back. In that test, the back and ends were removed allowing easy flow of air across the bunk. The standard set of materials, which included a high melting temperature polyamide carpet, vinyl coated aluminum bulkhead panels, and a low density acoustical tile overhead, was used on these first four tests. Then the lining materials were varied one at a time. A partial doorway opening and curtains over the bunk openings were also included as variations in the subsequent tests.

Figure 3 shows the temperature history of the upper air as determined by a thermocouple 25 mm (1-in) down from the center of the ceiling in the first three tests with the model. In the first two tests, the door was closed. The forced ventilation was on in the second test. Neither led to temperatures above 200°C. On the third test with the door open, the air temperature reached nearly 500°C. This temperature would probably have been maintained for a while or even increased if the full mattress thickness could have been used. Except for a localized bunk fire, this was not very spectacular.

Figure 4 shows the same three tests in the full-scale room. The first two tests again exhibited temperatures less than 200°C. The third test barely reached 500°C in the initial stage, but it maintained the high temperature long enough to ignite the cotton waste in the locker in the upper part of the compartment, causing a very severe fire reaching a final maximum temperature of 850°C.

Qualitatively, one might say that the model did not scale well for test 3 (a big fire in the burnout room--not much in the model). However, both temperatures approached 500°C, which is a danger point. One became fully involved; the other did not. It could have been deduced from the model test result that the potential for flashover was there. This is all that might be expected of the model.

Table 2 provides a comparison of the maximum gas temperatures 25 mm (1 in) below the center of the ceiling during the full- and reduced-scale tests. They are listed in order of increasing temperature in the full-scale test. They can be divided into three groups. When the temperature was less than 200°C in the full-scale test, it was less than 200°C in the model test. When the full-scale test temperature was between 200°C and 300°C the temperature in the model was also in the same range. However, when the temperatures were above 300°C there were large differences between the two scales. In two cases the temperature in the model was higher, and in the other two cases the temperature in the full-scale compartment was higher. These variations may be largely due to the complexity of the furnishing array where minor differences can determine whether or not the fire will progress to the next stage. Some statistical variation of the results would also be expected if these tests were repeated on the same scale. It is significant that in all four tests in the last group the temperatures in the model were above 450°C indicating that critical conditions were being approached. This suggests that a repeat of test 8 could lead to flashover.

In these tests, the reduced spacing between the tiers of the bunk in the model due to the thickness of the bedding materials brought the flames closer to the tier above thus increasing radiation feedback. This was counteracted to some extent by the restriction in airflow and by the earlier burnout of the mattress which had a smaller thickness than that required by the scaling rules. Thus, there are some limitations in the ability of the model to predict the full scale fire performance of complex furnishing arrays.

3.3 Room Fire Tests at Underwriters Laboratories, Inc.

In 1974 an extensive series of tests with over 30 different types of materials were run in the tunnel, wall, corner, and room configurations at the Underwriters Laboratories, Inc. (UL) [14] in order to examine the relationship between the ASTM E 84 flame spread classification (FSC) and the performance of the interior finish material in a variety of building enclosure geometries. Only five materials were run in the 2.4 x 3.6 x 2.4 m (8 x 12 x 8 ft) room with a 9.1 kg (20 lb) wood crib. Although the FSC correlated with the corner tests it did not do as well with the room tests as seen in Table 3. Materials S and A caused room flameover much sooner than the plywood even though they had a very low FSC. Neither the ASTM E 84 tunnel test, the corner tests, the ASTM E 162 flame spread test [15], or the NBS [16] and OSU [17] heat release calorimeters correlated with the room fire tests. The correlation between the full scale room fire tests and some previously unpublished quarter-scale room fire tests also run at UL using a gas burner were satisfactory for the four materials tested on the reduced scale. One reason that the times were shorter in the model was that the burner was turned on to its maximum gas flow at the beginning of the test while the wood crib required a considerable fire buildup time.

3.4 Mobile Home Fire Tests at NBS

A series of fire tests was conducted for the Department of Housing and Urban Development in the bedroom of a single-wide mobile home and in a quarter-scale model of that bedroom [18]. The objective of the test was to evaluate the relationship between fire buildup in the reduced-scale and full-scale enclosures and thus determine the feasibility of using a reduced-scale model test to assess the potential contribution of particular combinations of interior finish materials to fire growth in a mobile home.

The model tests simulated the phenomena of fire growth and flashover. However, the time to flashover occurred later in the model than in the full-scale bedroom. Flashover was taken as the time at which the level of radiation at the center of the floor reached 20 kW/m^2 which is just sufficient to ignite light combustibles. Several modifications to the model were examined, but while showing decided improvements, none adequately corrected the time delay to flashover. These modifications included (1) the use of propane instead of methane as the burner fuel to increase the radiation, (2) the use of a triangular burner designed to increase the flame contact with the walls, and (3) the use of a lower and wider door opening to increase the thickness of the gas layer above the door while maintaining $A\sqrt{H}$ the same. In spite of some long delays in flashover time, the rise in upper air temperature during the early part of the test was indicative of flashover conditions in the full-scale test.

The temperature history plots in Figure 5 illustrate the relatively rapid rise in the upper air temperature in the model up to around 500°C and the rather long delay time prior to flashover.

3.5 Room Fire Tests of Navy Hull Insulation at NBS

In the development of a screening test for the fire performance of hull insulation for the Navy [19], Lee and Breese found that strict conformance with the scaling rules did not result in flashover in the model for two PVC acrylonitrile butadiene closed cell foams even though flashover was observed for the corresponding full-scale room fire tests. In order to provide a more severe fire exposure in the model, the area of the wall above the door was increased to 1.4 and 1.8 times its scaled value. This resulted in lowering the doorway height to 0.93 and 0.86 times its scaled value. The width was adjusted in each case to maintain the scaled value of $wh^{3/2}$. The results of a number of these tests are summarized in Table 4. The lowering of the doorway height increases the area of the wall exposed to the high temperature gas layer and also affects the heat transfer to the surface. The amount by which it was lowered was determined empirically. No theoretical basis has yet been established. Even with these adjustments, the model tests still resulted in times to flashover as much as twice as long as their full-scale counterparts. The temperatures were somewhat higher in the full-scale room when the fire buildup was large and somewhat higher in the model when the fire buildup was low.

3.6 Room Fire Tests at NBS Involving Low Density Cellular Plastic Foams

A series of room fire tests was conducted at NBS with fiberglass, a 65 percent mineral and 35 percent cellulosic fiber insulating board, and five rigid cellular plastics covering a large range of FSC in a 2.9 x 3.2 m (9.5 x 10 ft) room 2.4 m (8 ft) high with a 0.74 m (29-in) wide and 1.9 m (76-in) high doorway [20]. These materials fully lined the test room during a cooperative research program with the National Research Council of Canada (NRCC) on the ASTM E 84 tunnel test. The ignition source was a natural gas diffusion burner located in a rear corner of the room and having a net heat release rate of 79 kW, which is equivalent to the burner in the ASTM E 84 tunnel. The time to flashover in the room is compared with that in the quarter scale model in Table 5. Data on the ASTM E 84 flame spread classification (FSC) and the peak heat release rate measured in the NBS heat release rate calorimeter [17] at an external radiant exposure of 20 kW/m^2 are included. Plywood (see section 3.7) and rigid PVC nitrile rubber foam (see section 3.5) are added to the table in order to extend the range of materials. Although they were not part of this test series, the test room and the gas flow rates to the burner are about the same for these materials. It is noted that the times to flashover in the model correlate better than the FSC or the heat release rates with the full-scale behavior. The ranking of the materials in the quarter-scale room fire tests is identical to that for the full-scale tests with flashover times on the order of 50 percent greater than for the full-scale counterparts. The fire buildup for low density plastic foam material D in the full scale room rapidly reached a peak and died back within 60 s. It finally went to flashover at 368 s when the corner above the burner reignited. The model followed the initial buildup and decay but did not reignite.

3.7 Room Fire Tests at NBS Involving Plywood

Two unreported demonstration tests were then run a week apart in November 1976 of a 3.1 x 3.1 x 2.4 m (10 x 10 x 8 ft) room fully lined with 4 mm (5/32 in) thick plywood using a natural gas burner in the rear corner with a net heat release rate of 79 kW (4500 Btu/min). Each full-scale test was followed by a quarter scale model test on the same day. As seen in Table 6, the times at which the flames first extended beyond the doorway were remarkably close between the model and prototype run on the same day, even though there were differences between the full-scale tests run on different days. A comparison of the temperature histories in the upper part of the room for the two scales is shown in Figure 6 for test 2.

3.8 Reduced Scale Room Corner Fire Tests at Upjohn

The Upjohn Company [21] has also utilized this scaling procedure to model the 2.4 m x 3.7 m (8 x 12 ft) height ICBO [22] room fire test. They used a scale factor of one-third rather than one-fourth. A premixed flame of propane and air was programmed to provide a similar heat release rate and flame height history as the 13.6 kg (30 lb) crib used in the full-scale tests at the Underwriters Laboratories. However, the gas flow rate to the burner was reported to be approximately 12 percent higher than the scaled value. This was apparently done to make the model test slightly more severe. No reduction in $wh^{3/2}$ was made to account for the air introduced by the premixed burner. Three cellular plastic foams: a 25 mm (1 in) foil-faced polyisocyanurate, a 25 mm (1 in) foil-faced polyurethane, and a 51 mm (2 in) sprayed-on polyurethane, were compared between full- and one-third-scale tests. The gas temperature rise at 13 mm (0.5 in) and 25 mm (1 in) below the center of the ceiling in the model and full-scale tests, respectively, agreed within $\pm$ 12 percent which they considered acceptable for a screening test. In the test with an asbestos cement board blank, the temperatures were as much as 30 percent higher in the model. The temperature histories are shown in Figures 7-10.

3.9 Room Fire Research Tests at NBS Using Fiberglass Insulation

In order to investigate the observed differences between the two scales and the effect of the lintel height in the model, Lee conducted a set of full- and quarter-scale room fire tests with the walls and ceiling lined with fiberglass [23]. The room was 3.1 x 3.1 x 2.3 m (10 x 10 x 7.5 ft) high with a 0.73 x 1.93 m (29 x 76 in) doorway. Three different doorway heights were used in the model. The burner gas flow rate was varied to represent different degrees of fire buildup. Full- and quarter-scale comparisons were made with the burner in the center of the room, against the rear wall, and in the corner. The test results summarized here are limited to the corner location for the burner and to two heat release rates by the burner. In full-scale test P2, the heat release rate was 140 kW, and the flames were restricted to the region around the corner where the burner was located. In the reduced-scale counterparts, the flame extended somewhat beyond that but did not reach the thermocouple tree at the center of the room. In full-scale tests P12, the heat release rate was 460 kW, and the flames covered the ceiling but did not extend beyond the doorway. In the corresponding model tests, they extended beyond it. The thermocouples at the center of the ceiling and 102 mm (4 in) below it were outside of the flame zone in the P2 series of tests and were in the flames in the P12 series. The temperature versus time of the center of the ceiling and of the air 102 mm (4 in) below it for full-scale test P2 and at 25 mm for its reduced-scale counterparts are shown in Figures 11 and 12. MI refers to the scaled doorway height, MII to 93 percent scaled, and MIII to 86 percent scaled. The best agreement is obtained for strict adherence to the scaling rules. The vertical temperature profiles for the center of the room and the center of the doorway are shown in Figures 13 and 14 for the same tests. Again, the agreement was best for strict scaling. Figure 15 compares the doorway velocities for the same tests. In Figure 16, the velocity in the model (M2I) is multiplied by 2, which is the reciprocal of the square root of the scale factor, and compared with the velocity in the full-scale test. The maximum scaled inflow velocity was about 20 percent higher than the full-scale value, but the outflow velocities were quite close. Table 7 shows the heat fluxes incident at the center of the ceiling and the temperature of the hot gas layer 25 mm (1 in) below it. In neither full-scale test P2 nor its quarter-scale counterparts does the flame from the burner reach the thermocouple. However, the relatively larger flames in the models may be responsible for their larger heat fluxes at the center of the ceiling. In the case of full-scale test P12, the flame covers the entire ceiling as it does for its quarter-scale counterparts M12I, M12II, and M12III. In this case, the heat flux to the surface is considerably higher than for the models. As seen in column 6, the heat flux to the surface divided by the black body radiation based on the gas temperature below it is relatively constant (0.6 ± 0.06) where there is flame impingement. This would suggest that the increased heat transfer in the full-scale case is related to the increase in temperature rather than a dramatic increase in the effective emmissivity. These ratios are higher for the nonflame impingement case presumably due to a higher ratio of convection to radiation. No attempt was made to separate the convective and radiative components of the heat flux in these measurements.

4. ANALYSIS

The data from the fiberglass lined rooms in section 3.9 suggest a possible explanation for the reduction in severity of the quarter-scale test. Because of the relatively longer flame lengths and larger flame areas in the model, the heat release rate per unit area of the flame is smaller. Therefore, the temperature and the rate of heat loss per unit area of the flame would be lower and there would be a lower rate of heat transfer to the surface. This reduction in heat transfer would result in a lower mass loss rate or heat release rate per unit area of the specimen material. The lower heat release rate per unit area of the specimen may be compensated for to some extent in the early growth of the fire by the larger scaled burning area.

However once the flames cover the ceiling in the full scale room, they will be extending beyond the doorway of the model for the same scaled heat input and this extension will not be contributing to the temperature rise in the room.

A simplified analysis will be presented here to qualitatively account for the dependence of the flame height and the flame temperature on scale. The full-scale test P12 will first be idealized by assuming that the 460 kW is released from a line burner at the 3.0 m long base of the rear wall in order to simulate one dimensional flame spread.

The flame length is calculated by the following model suggested by Quintiere [24]. The velocity distribution u(y) in a turbulent boundary layer on a flat plate with forced flow is given by [25].

$$\frac{u}{u_\infty} = \left(\frac{y}{\delta}\right)^{1/7} \tag{4-1}$$

where u_∞ is the free stream velocity and y is the distance from the surface. The boundary layer thickness is given by

$$\delta = 0.366\ x \left(\frac{u_\infty x}{\nu}\right)^{-1/5} \tag{4-2}$$

where x is the distance along the surface and ν is the kinematic viscosity. The mass of air flowing in a unit width of the boundary layer is given by

$$\dot{m}'_a = \rho_\infty \int_0^\delta u dy = \rho_\infty\ u_\infty \int_0^\delta \left(\frac{y}{\delta}\right)^{1/7} dy = \frac{7}{8}\ \rho_\infty\ u_\infty\ \delta\ . \tag{4-3}$$

The heat released per unit width of the flame up to the distance x assuming complete consumption of the oxygen is given by

$$\dot{Q}' = \left(\frac{H_c}{r}\right) y_{O_2} \dot{m}'_a = E\ m.c._{O_2} \dot{m}'_a = E\ y_{O_2} \frac{7}{8} \rho_\infty u_\infty\ (0.366)\ x \left(\frac{u_\infty x}{\nu}\right)^{-1/5}$$

$$\dot{Q}' = 0.32\ \rho_\infty u_\infty^{4/5}\ \nu^{1/5}\ x^{4/5}\ E\ y_{O_2} \quad , \tag{4-4}$$

where H_c is the heat of combustion of the fuel, r is the ratio of the mass of oxygen to the mass of fuel consumed in the reaction, y_{O_2} is the mass fraction of oxygen in the free stream, and E is the heat released per unit mass of oxygen consumed (13.1 MJ/kg [26]). Thus, the flame length is given by

$$x_f = (0.32\ \rho_\infty)^{-5/4}\ \nu^{-1/4} u_\infty^{-1}\ (E\ y_{O_2})^{-5/4}\ (\dot{Q}')^{5/4} \quad . \tag{4-5}$$

Taking the following values for the properties,

$$E = 13.1 \times 10^3\ \text{kJ/kg} \qquad \rho_\infty = 1.2\ \text{kg/m}^3 \qquad \nu = 20 \times 10^{-6}\ \text{m}^2/\text{s} \qquad y_{O_2} = 0.233\ ,$$

$$x_f = 0.00218\ (\dot{Q}')^{5/4}/u_\infty \quad . \tag{4-6}$$

The flame length for the idealized model has the following scale dependence,

$$x_f \sim \frac{(\dot{Q}')^{5/4}}{u_\infty} \sim \frac{\left(\frac{\dot{Q}}{W}\right)^{5/4}}{u_\infty} \sim S^{3/4}. \tag{4-7}$$

This expression is derived by assuming that the total heat release rate is proportional to the square of the scale, the width proportional to the scale, and the velocity proportional to the square root of the scale. Hence the flame from the burner in the model will be too high by a factor of $\frac{S^{3/4}}{S} = S^{-1/4}$ or 1.41 for 1/4 scale.

Next assume that the temperature, T, of the flame is given by

$$\frac{T_{ad} - T}{T_{ad} - T_\infty} = \frac{\dot{q}''_L}{\dot{q}''_f} \quad , \tag{4-8}$$

where the drop in temperature from its adiabatic value, T_{ad}, divided by the increase in the adiabatic temperature over the free stream temperature, T_∞, is equal to the ratio of the rate of heat losses, $\dot{q}''_L$, to the rate of heat production, $\dot{q}''_f$, in the flame. It is assumed that the heat capacity is independent of temperature. Then

$$T - T_\infty = (T_{ad} - T_\infty)\left(1 - \frac{\dot{q}''_L}{\dot{q}''_f}\right) . \tag{4-9}$$

or letting $T^* = T - T_\infty$ and $T^*_{ad} = T_{ad} - T_\infty$

$$T^* = T^*_{ad}\left(1 - \frac{\dot{q}''_L}{\dot{q}''_f}\right) . \tag{4-10}$$

The adiabatic flame temperature for methane is 2236 K, and if the ambient temperature is taken to be 293 K, then T^*_{ad} = 1943 K. The area of the back wall and ceiling of the fiberglass lined room was 15.9 m^2 (171 ft^2) and the total heat release rate of the gas burner during test P12 was 460 kW yielding an average heat release rate per unit area of 29 kW/m^2 for the flame. The heat flux to the ceiling tabulated in Table 7 is really to a water-cooled heat flux gage. The heat flux to the ceiling surface would be much less due to the smaller temperature difference between the hot gas and the surface. In estimating the heat losses from the flame, the actual net heat flux into the fiberglass ceiling at 78 seconds can probably be neglected in comparison to the heat radiated into the lower part of the room. Likewise, the rate of heat absorption of the walls will be low so that a large portion of the radiation will reach the floor level. If the 13 kW/m^2 measured by a heat flux gage located at the center of the floor is taken as a lower limit for the losses, then 1092°C is the upper limit for the temperature according to equation (4-9). The temperature was 820°C measured with a 0.51 mm (20 mil) bare thermocouple 102 mm below the center of the ceiling. The temperature would be expected to vary some with distance along the ceiling.

Next we determine how this temperature might be expected to change with scale. The heat release rate from the burner $\dot{Q}_B$ is proportional to the area, A, of the room. To the extent that there is a similar involvement on the two scales, the total rate of heat release, $\dot{Q}$, will also be proportional to the area. (This assumption will be violated to some extent by differences in both the heat release rate per unit area of the surface and the area of the surface that is covered with flame at a given time. The assumption will be better early in the test before the involvement of the lining materials becomes large.) However, in the experiments discussed here, the total rate of heat release is from the burner and it is proportional to A on both scales so that

$$\dot{Q} \equiv \dot{Q}_B \sim A \sim S^2 \tag{4-11}$$

as assumed in equation (4-7).

The heat release rate per unit area of the flame is given by

$$\dot{q}''_f = \frac{\dot{Q}}{Wx_f} \sim S^{1/4} \tag{4-12}$$

Then the heat release rate per unit area of the flame on any scale is given by

$$\dot{q}''_f = S^{1/4}\,\dot{q}''_{f_1} , \tag{4-13}$$

where $\dot{q}''_{f_1}$ is the heat release rate per unit area of the flame in the full-scale test. Next make the approximation that the rate of heat loss per unit area from the flame is proportional to the temperature rise. That is

$$\dot{q}''_L = aT^* \quad . \tag{4-13}$$

Equation (4-10) can now be written

$$T^* = T^*_{ad} \, (1 - aT^*/\dot{q}''_{f_1} S^{1/4}) \quad . \tag{4-14}$$

For the full-scale test the temperature rise T^*_1 is given by

$$T^*_1 = T^*_{ad} \, (1 - aT^*_1/\dot{q}''_{f_1}) \quad , \tag{4-15}$$

Since $a/\dot{q}''_{f_1}$ is unknown but assumed to be constant between scales, it can be eliminated by combining equations (4-14) and (4-15) so that

$$\frac{T^*}{T^*_1} = \frac{S^{1/4}}{1 + (S^{1/4}-1)T^*_1/T^*_{ad}} \quad , \tag{4-16}$$

where $T^*_{ad} = 1943$, $S^{1/4} = 0.707$ for the quarter-scale test, and T^*_1 was observed to be 800 K. Then $T^*/T^*_1 = 0.80$ and T^* would be expected to be 643 K. The observed temperature increase for the model with strict scaling was reported to be 610 K.

Obviously, the oversimplification of the theory, the idealization of the experiment, and the uncorrected radiation errors in the temperature measurement do not permit confidence in the fortuitous agreement reported here. However, this reasoning does indicate and the experiments verify that one should expect a reduced effective flame temperature and a lower heat transfer rate to the pyrolyzing surfaces in the model and, thus, a less serious fire. Until the heat release rate in the compartment due to the combustible linings becomes comparable to the heat release rate by the burner, the average upper gas temperature developed in the model with strict scaling should be similar but the flame temperature and the heat transfer to the flame covered surface should be greater in the full-scale case.

5. PHYSICAL MODELING OF AIRCRAFT CABIN FIRES

In this discussion, two types of Froude scaling will be considered, both of which scale the aircraft fuselage geometrically. The type presently used at FAA [27] also scales the doorways geometrically and will be referred to as "geometrical scaling". The type used in the NBS compartment modeling will be referred to as "proportional scaling" in the sense that the induced air flow is proportional to the area of the compartment. This is accomplished by making the doorway width proportional to the square root of the scale and the height proportional to the scale.

In order to maintain the same temperature in the model as in the prototype, the heat produced, the heat absorbed by or conducted through the room linings, and the heat passing out of the doorway must all be reduced in the same proportion. For geometrical scaling, the air flow through the doorway is proportional to $wh^{3/2}$ and, thus, to $S^{5/2}$. The heat release rate or burning area and the heat losses through the lining materials must also be proportional to $S^{5/2}$ and, hence, the heat loss per unit area through the lining must be proportional to $S^{1/2}$. This would be accomplished if $K\rho_s C_{p_s} \sim S$ and $Z^2 \sim K/\rho_s C_{p_s}$ where Z is the thickness, K is the thermal conductivity, ρ_s is the density and C_{p_s} is the heat capacity of the room lining material. For proportional scaling, the air flow and the burning area are proportional to S^2, so that $K\rho_s C_{p_s}$ and Z should both be independent of scale. If combustible lining materials are involved, it is essential to have proportional scaling.

The fact that the radiation into the compartment from an external fire is proportional to the doorway area (wh) while the air flow is proportional to $wh^{3/2}$ will be referred to as the "doorway dilemma" since their ratio needs to be independent of scale for proper modeling in either of these cases.

Unless the "doorway dilemma" is counteracted, it will result in radiation levels--and, thus, temperatures--which are too high in either model. Note that both of these models obey Froude scaling only so long as the temperature profiles in the compartment are similar on both scales. In that case, the velocity is proportional to $\sqrt{h}$ and h is proportional to the scale.

External to the fuselage there is also a problem of maintaining the Froude number. For large pool fires, the heat release rate per unit area becomes independent of the diameter. Thus,

$$\dot{Q} \sim D^2 \sim S^2 \quad , \tag{5-1}$$

since the diameter of the pool, as well as the fuselage, is taken to be proportional to the scale (S) in these experiments. According to de Ris [28], the heat release rate ($\dot{Q}$) of the pool fire can be written as

$$\dot{Q} = \rho C_p T^* \left(\frac{\pi D^2}{4}\right) u \quad , \tag{5-2}$$

where ρ is the density of the fire plume gas, C_p is its heat capacity, T^* is its average temperature rise over ambient, D is the diameter of the fire, and u is the average upward velocity of its plume gas.

Thomas [29] gives the following expression for the flame height, ℓ,

$$\frac{\ell}{D} = f\left(\frac{\dot{Q}^2}{gD^5 B \Delta T}\right) \quad , \tag{5-3}$$

where B is the expansion coefficient of gases, 1/T; ΔT is the temperature rise of the flame just above the pool; and g is the acceleration due to gravity. For geometric similarity of the flame:

$$\frac{\dot{Q}^2}{g D^5 B \Delta T} = \text{constant} = \phi \quad . \tag{5-4}$$

Combining (5-2) and (5-4), the Froude number is given by

$$Fr = \frac{u^2}{gD} = \frac{16\phi B\Delta T}{\pi^2 \rho^2 C_p^2 T^{*2}} \quad , \tag{5-5}$$

which can be considered to be independent of scale. Furthermore, from (5-3) the flame height (ℓ) is proportional to scale:

$$\ell \sim D \sim S \quad . \tag{5-6}$$

However, this requires that

$$\dot{Q} \sim D^{5/2} \sim S^{5/2} \quad , \tag{5-7}$$

as seen from (5-4).

Because of the conflict between (5-1) and (5-7), it is impossible to maintain Froude scaling, as pointed out by Eklund [27], as long as the same liquid fuel is used for the model and the prototype. However, if a gaseous fuel were used in the model, it could be controlled so that $\dot{Q} \sim S^{5/2}$ and similarity of the flame and the Froude number could be maintained. If the emissivity of the gas flame in the model were less than that of the flame from the aviation fuel in the full scale fire, the radiation into the small scale compartment would be reduced. This would help to counteract the "doorway dilemma" discussed above. Hence, the fuel gas should be selected on the basis of its radiating properties.

The ambient wind must scale as

$$u_W \sim D^{1/2} \sim S^{1/2} , \qquad (5\text{-}8)$$

to satisfy Froude scaling.

In light of the above discussion, the following set of scaling rules are recommended.

(1) The fuselage and fuel source dimensions should be proportional to the scale except for the thickness of the fuselage, which should be independent of scale.

(2) The doorway heights should be proportional to scale.

(3) The doorway widths should be proportional to the square root of the scale.

(4) The rate of heat release ($\dot{Q}$) of the simulated pool fire should be proportional to the scale to the 5/2 power.

(5) The emissivity of the flame should be proportional to the square root of the scale.

(6) The wind velocity should be proportional to the square root of the scale.

The above set of scaling rules apply to the whole system. If the impact of the lining materials on the fire growth is to be singled out for evaluation, then the problem can be uncoupled by replacing the burning seats with a gas burner adjusted to produce the same scaled rate of heat output as in a full scale cabin fire. That rate could be determined from mass loss rate measurements in the full scale fire along with effective heat of combustion measurements of the seats in a large scale calorimeter such as the NBS furniture calorimeter [30]. Proportional scaling should be used.

6. CONCLUSIONS

Until mathematical modeling becomes sufficiently advanced to make reliable predictions based on fire property data, the reduced-scale physical model probably represents the best indication of full-scale behavior of any laboratory test method. However, the model is generally less severe in terms of maximum temperatures and times to flashover than the full scale test. Thus, important conclusions found from the model tests need to be verified by a limited amount of full-scale testing. The reduced-scale model can also be helpful in the development, verification, and refinement of the mathematical models which should be able to predict on small- as well as full-scale.

Proportional scaling (i.e., the area of the doorway times the square root of its height is proportional to the area of the enclosure) is required when the interior surfaces are involved in combustion. It also is necessary in order to take heat conduction losses through the lining materials properly into account unless the lining materials can be changed with scale as required by geometrical scaling. The burning of interior items, such as seats, can be done with geometrical scaling, provided the burning areas are adjusted to be proportional to $S^{5/2}$.

The proportionally scaled quarter-scale room fire test has always been somewhat less severe than its full-scale counterpart, and a plausible explanation is presented in this report. Provided the fire is not ventilation limited, the geometrically scaled test would be expected to be somewhat more severe than the proportionally scaled test because of the reduction in air flow. Neither type of scaling can be counted on to exactly duplicate a full-scale test.

7. ACKNOWLEDGMENT

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Table 1. Compartment Fire Tests

Model * Test	Full-Scale Test	Lining Variable	Door	Air ** Supply	Bunk	Bunk Location
M1	1	Standard Set	Closed	Off	Closed	Side
M2	2	Standard Set	Closed	On	Closed	Side
M3	3A	Standard Set	Open	At bunks	Closed	Side
M3≡	3B	Standard Set	Open	On	Closed	Side
	4A	Standard Set	Open	On	Open	Side
M4≡	4B	Standard Set	Open	On	Open	Side
M5	5	High Density Acoustical Panels on Overhead	Open	On	Closed	Side
M6	6	Melamine Coated Panels on Bulkhead	Open	On	Closed	Side
M7	7	Wool Carpet and Pad on Deck	Open	On	Closed	Side
M8	8	Wool Carpet and Pad on Deck	1/2 Open	Off	Closed	Back
M9	9***	Wool Carpet and Pad on Deck and Curtains Over Bunk Openings	1/2 Open	Off	Closed	Back
	10***	Acrylic Carpet and Pad on Deck and Curtains Over Bunk Openings	1/2 Open	Off	Closed	Back
	11	80% Wool-20% Pheno-Formaldehyde Carpet and Pad on Deck and Curtains Over Bunk Openings	1/2 Open	Off	Closed	Back

*Used one inch bunk mattresses.
**Mainly from ceiling vent unless otherwise specified.
***Ignition along length of bottom bunks. All other tests had localized ignitions on the bottom bunk.

Standard Set Of Linings

Low Density Acoustical Panels on Overhead
Fibrous glass Insulation on Two Bulkheads
Vinyl Coated Panels on Two Bulkheads
High Temp Polyamide Carpet Bonded to Steel Deck

Table 2

Comparison of the Maximum Upper Air Temperature in the Full and Quarter Scale Navy Berthing Compartment Tests

Test	Scale	Temperature (°C)
2	Full	172
	Quarter	168
1	Full	191
	Quarter	107
---	---	---
6	Full	200
	Quarter	209
3B	Full	230
	Quarter	202
5	Full	232
	Quarter	265
7	Full	244
	Quarter	250
---	---	---
8	Full	354
	Quarter	706
4B	Full	570
	Quarter	875
9	Full	800
	Quarter	531
3A	Full	850
	Quarter	463

Table 3

Comparison of Room Fire Tests at UL with Quarter Scale Model and Other Laboratory Tests

Code	Flameover in (2.4x3.6m) Room (s)	FSC	Flameover in Open Corner (s)	E 162	Heat Release Rate OSU at $30 kw/m^2$	Heat Release Rate NBS at $30 kW/m^2$	Heat Release Rate NBS at $60 kW/m^2$	Flameover in 1/4-Scale Room (s)
S	80	22	∞	375	NA	31	51	15
A	100	23	∞	3	21	8.1	32	NA
G	∞	23	∞	NA	114	< 3	56	∞
R	∞	27	∞	3	NA	< 3	16	∞
H	260	178	285	NA	> 190	120	130	150

S = Polyisocyanurate Foam (37 kg/m^2) (2.3 PCF)

A = Polyisocyanurate Foam (30 kg/m^3) (1.9 PCF)

R = Same as "A", but Foil-Faced

H = Plywood (6.4 mm) (0.25 in)

G = Fire Retardant Treated Plywood (12.7 mm) (0.5 in)

Table 4

Summary of Full- and Quarter-Scale Compartment Fire Tests of Navy Hull Insulation

Test	Scale	Doorway Height	Insulation	Coating	Gas burner ignition source (kW)	Time to flame out doorway (s)	Max.* upper air temp. T_1 (°C)	Max.** doorway temp. T_2 (°C)	Ignition time of flashover indicator (s)
FS-1	Full	Full size	C2	2 Coats A-207	62	27	774	572	30
6	1/4	Scaled	C2	2 Coats A-207	Scaled 62	< 45	647	-	> 45
23	1/4	0.93 Scaled	C2	2 Coats A-207	Scaled 62	54	683	598	64
FS-2	Full	Full size	C2	None	62	32	829	630	30
4	1/4	Scaled	C2	None	Scaled 62	-	216	215	∞
22	1/4	0.93 Scaled	C2	None	Scaled 62	38	707	585	43
33	1/4	0.86 Scaled	C2	None	Scaled 62	44	604	451	47
FS-3	Full	Full size	B2	None	62	42	830	695	46
10	1/4	Scaled	B2	None	Scaled 62	∞	221	196	∞
24	1/4	0.93 Scaled	B2	None	Scaled 62	∞	410	288	∞
34	1/4	0.86 Scaled	B2	None	Scaled 62	88	646	500	97
FS-4	Full	Full size	B2	2 Coats 0-987	94	∞	362	297	∞
44	1/4	0.93 Scaled	B2	2 Coats 0-987	Scaled 94	∞	398	288	∞
51	1/4	0.86 Scaled	B2	2 Coats 0-987	Scaled 94	∞	427	311	∞
FS-5	Full	Full size	B2	2 Topcoats 0-987 3 Topcoats A-207	94	∞	357	287	∞
46	1/4	0.93 Scaled	B2	2 Topcoats 0-987 3 Coats A-207	Scaled 94	∞	404	324	∞

*Measured at 25 mm below center of overhead
**Measured at 25 mm below top of doorway

Table 5

Comparison of Times to Flashover in Full- and Quarter-Scale Room Fire Tests with Laboratory Fire Tests in the NBS/NRCC Cooperative Program

Material Description	Material Code	Time to Flashover in Room (s)	Time to Flashover in 1/4-scale Model (s)	FSC	Maximum Heat Release Rate at 20 kW/m^2 (kW/m^2)
Rigid Polyurethane Foam	A	13	18	500	140
Rigid Polyurethane Foam	B	14	19	250	140
Rigid Polyurethane Foam	C	18	32	30	47
Rigid PVC Nitrile Foam	B-2	42	97	31 31 28	100
Plywood	H	156	185	178	120
Polyisocyanurate Foam	D	368	∞*	20	13
Foil Faced Polyisocyanurate Foam	E	∞	∞	20	7
65% Mineral 35% Cellulose Fiberboard	F	∞	∞	15	28
Fiber Glass	G	∞	∞	15	< 3

*When the gas flow to the burner was increased by 50%, flashover occurred in 53 s.

Table 6

Comparison of Times to Flashover and Flameover in Full and Quarter Scale Fire Tests of Rooms Fully Lined with Plywood

TEST	FLAME OUT DOORWAY (sec)	FLASHOVER (sec)
Full-Scale #1	190	*
Model #1	188	*
Full-Scale #2	158	156
Model #2	158	185

*Fire was extinguished before flashover occurred.

Table 7

Comparison of Air Temperatures and Heat Fluxes in the Full- and Quarter-Scale Room Fire Tests with the Room Fully Lined with Fiberglass

Test*	Scaled Heat Release Rate of Burner (kW)	Air Temperature (°C)**	Heat Flux at Center of Ceiling $\dot{q}''_c$ (kW/m^2)	Time to Flashover (s)***	$\frac{\dot{q}''_c}{\sigma T^4}$****
P2	140	374	5.2	∞	0.52
M2I	140	365	11.0	∞	1.20
M2II	140	428	12.0	∞	0.92
M2III	140	413	11.0	∞	0.92
P12	460	820	51.0	78	0.64
M12I	460	630	25.0	138	0.68
M12II	460	719	33.0	102	0.61
M12III	460	721	31.0	90	0.57

*P = full-scale test
MI = model test with doorway height 0.25 x full-scale
MII = model test with doorway height 0.23 x full-scale
MIII = model test with doorway height 0.22 x full-scale

**Measured 25 mm below center of ceiling at time of flashover if it occurred; otherwise at 240°C.

***Time at which heat flux to floor reached 20 kW/m^2.

****Ratio of heat flux to the center of ceiling to the black body radiation at the temperature in column 3.

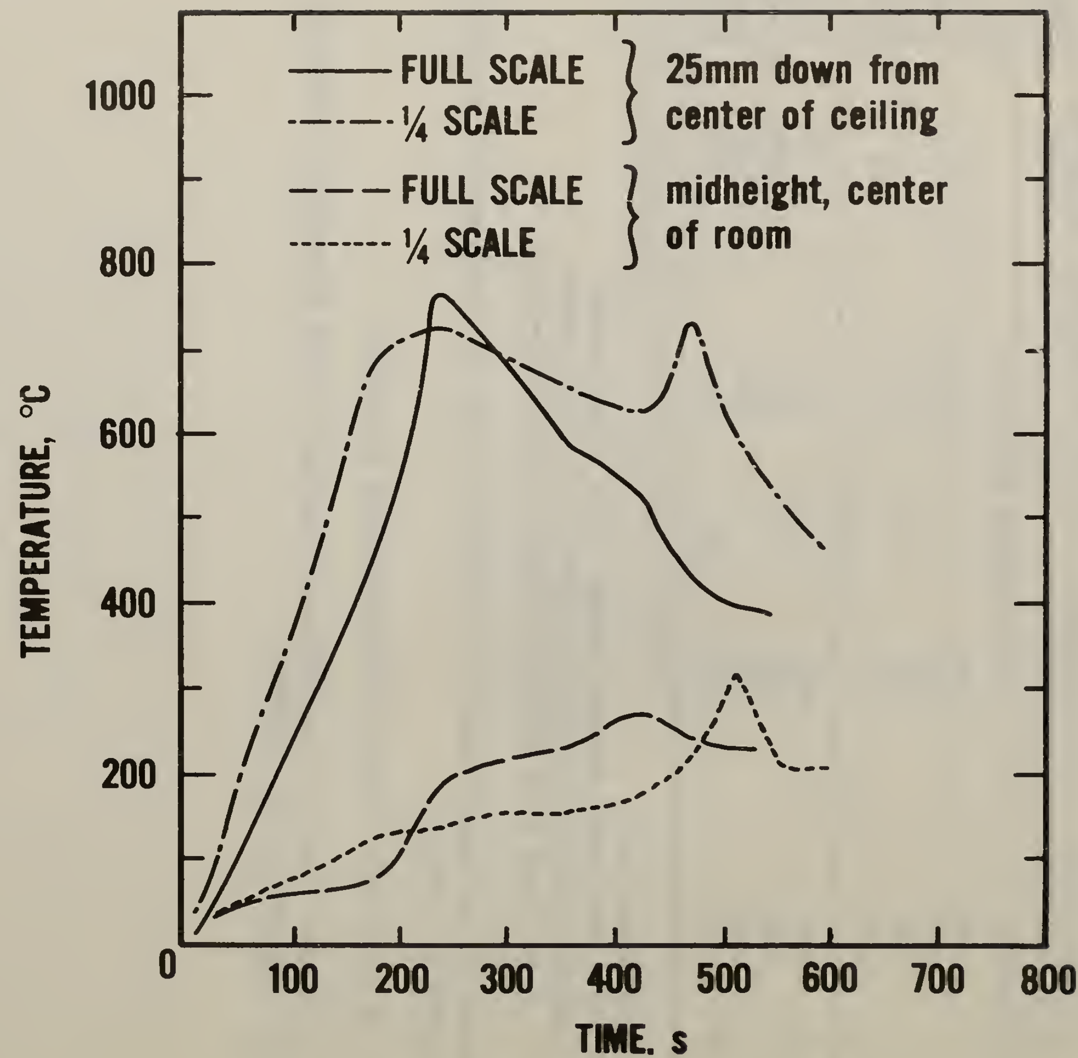

Figure 1. Comparison of Air Temperature Histories in Full- and Quarter-Scale Room Corner Tests with Lauan Plywood Walls and Gypsum Board Ceiling

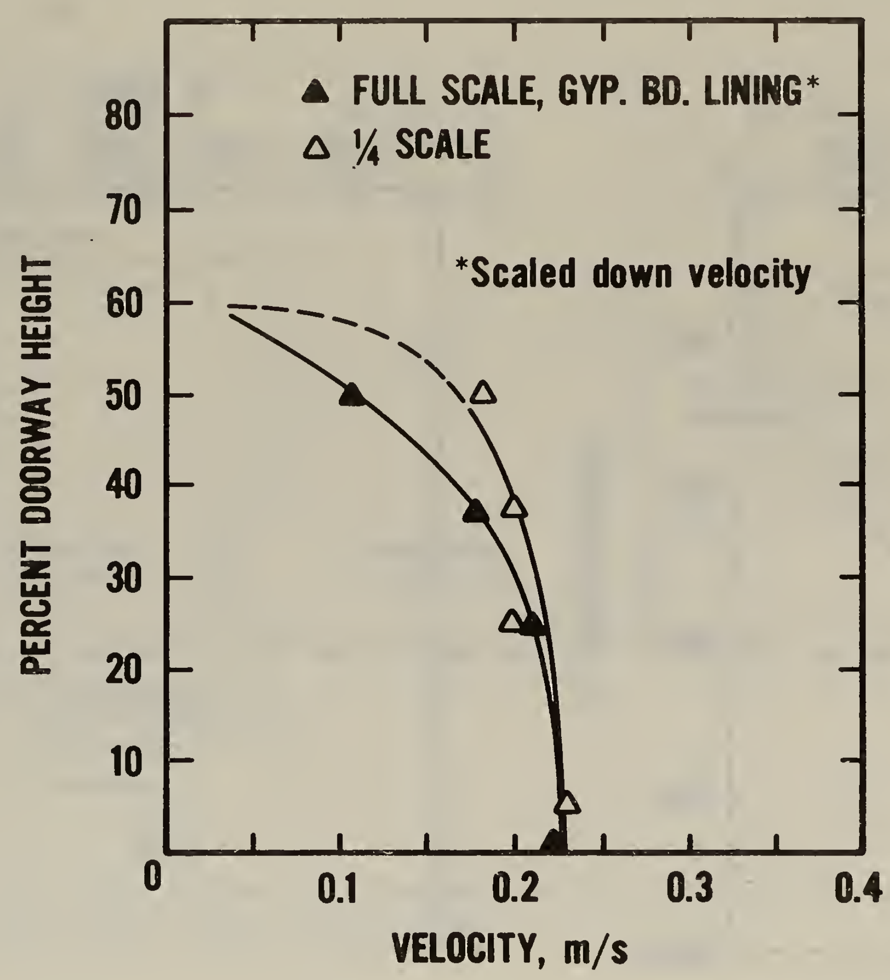

Figure 2. Comparison of Air Inflow Velocity Profiles in Full- and Quarter-Scale Room Corner Tests with Lauan Plywood Walls and Gypsum Board Ceiling

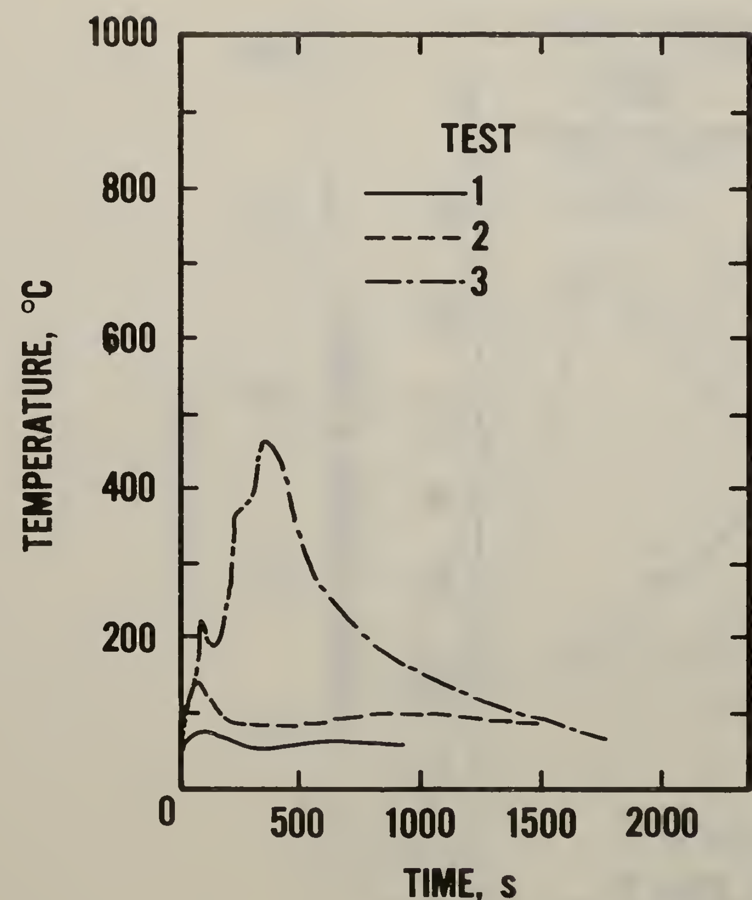

Figure 3. Upper Air Temperature History in Full-Scale Navy Berthing Compartment

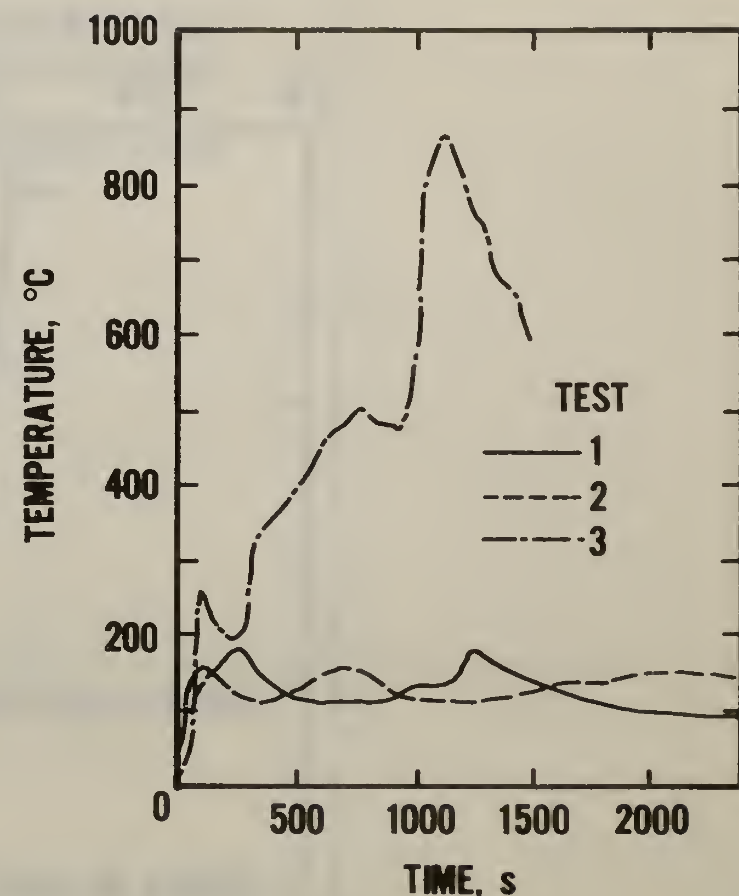

Figure 4. Upper Air Temperature History in Quarter-Scale Navy Berthing Compartment

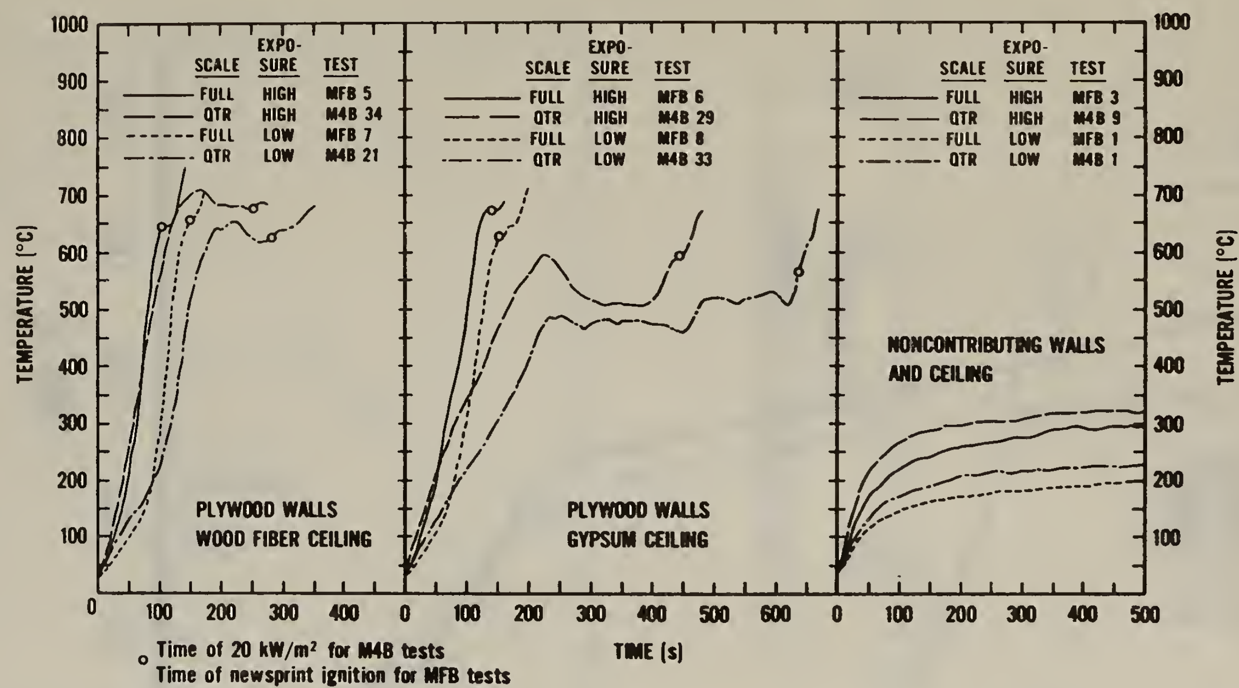

Figure 5. Comparison of Upper Air Temperature Histories in Full-and Quarter-Scale Mobile Home Bedroom Tests

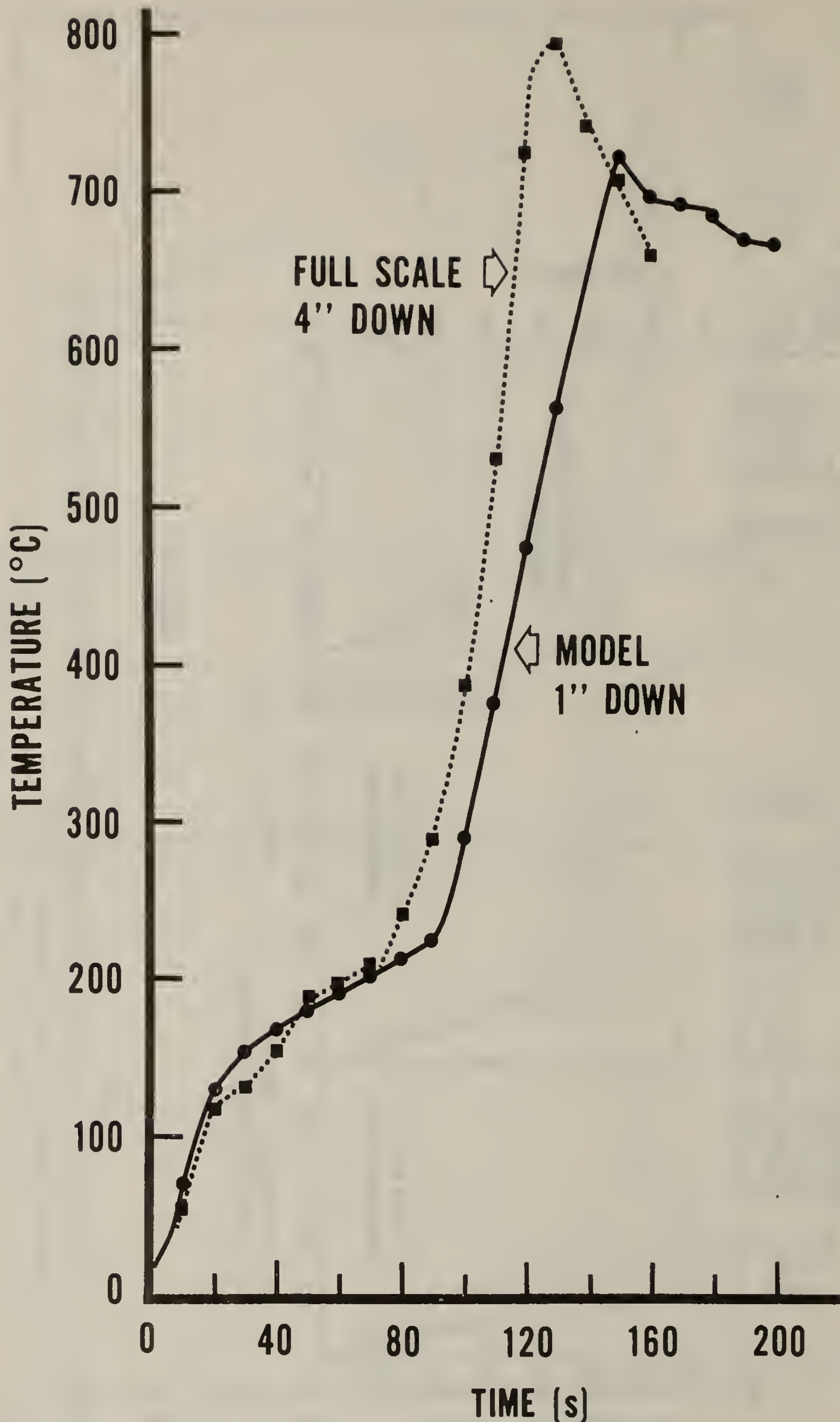

Figure 6. Comparison of Upper Air Temperature Histories in the Full- and Quarter-Scale Fire Tests of the Room Fully Lined with Plywood

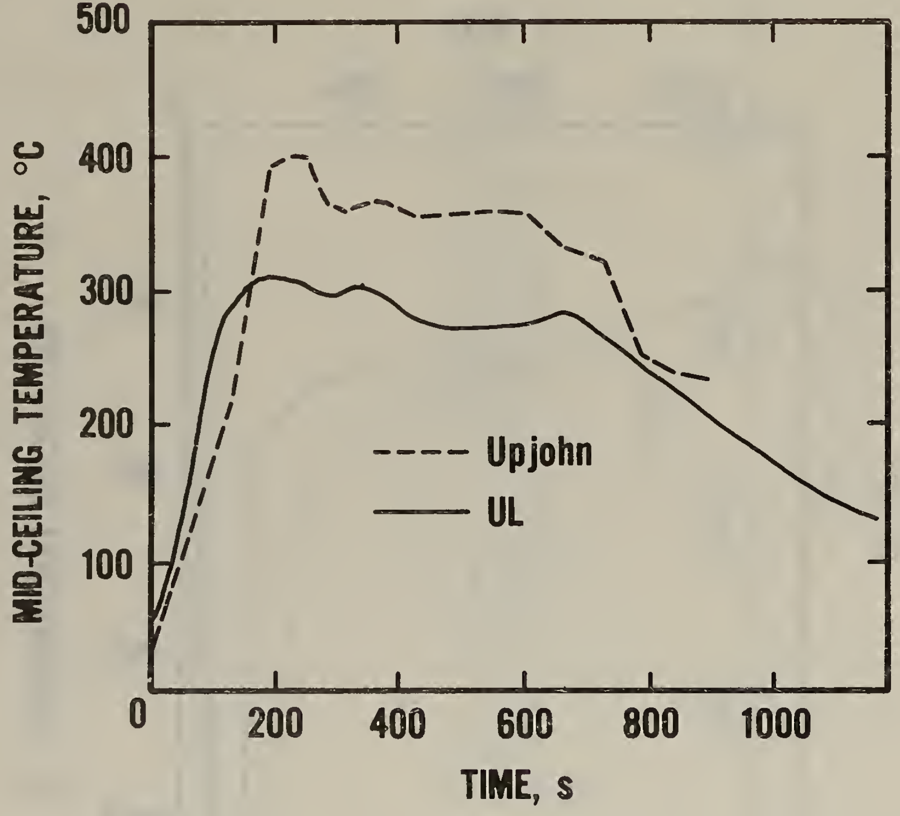

Figure 7. Comparison of Upper Air Temperature Histories in the Full- and Third-Scale Fire Tests of the Room Fully Lined with Asbestos Cement Board

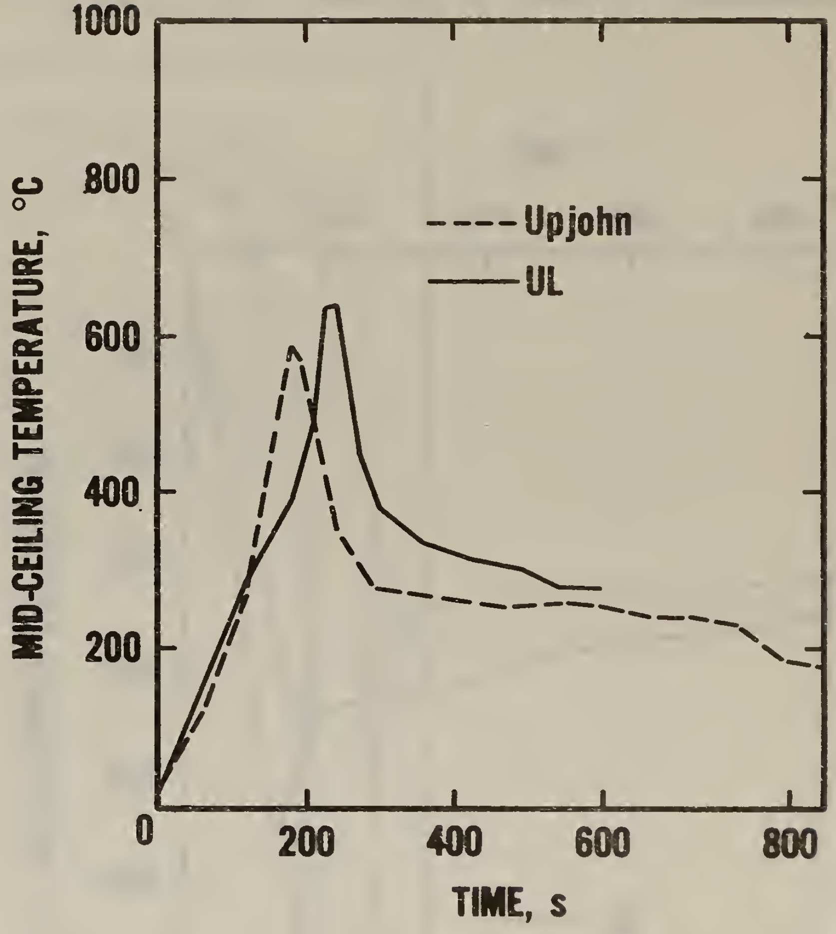

Figure 8. Comparison of Upper Air Temperature Histories in the Full- and Third-Scale Room Corner Tests of a Foil-Faced Polyurethane

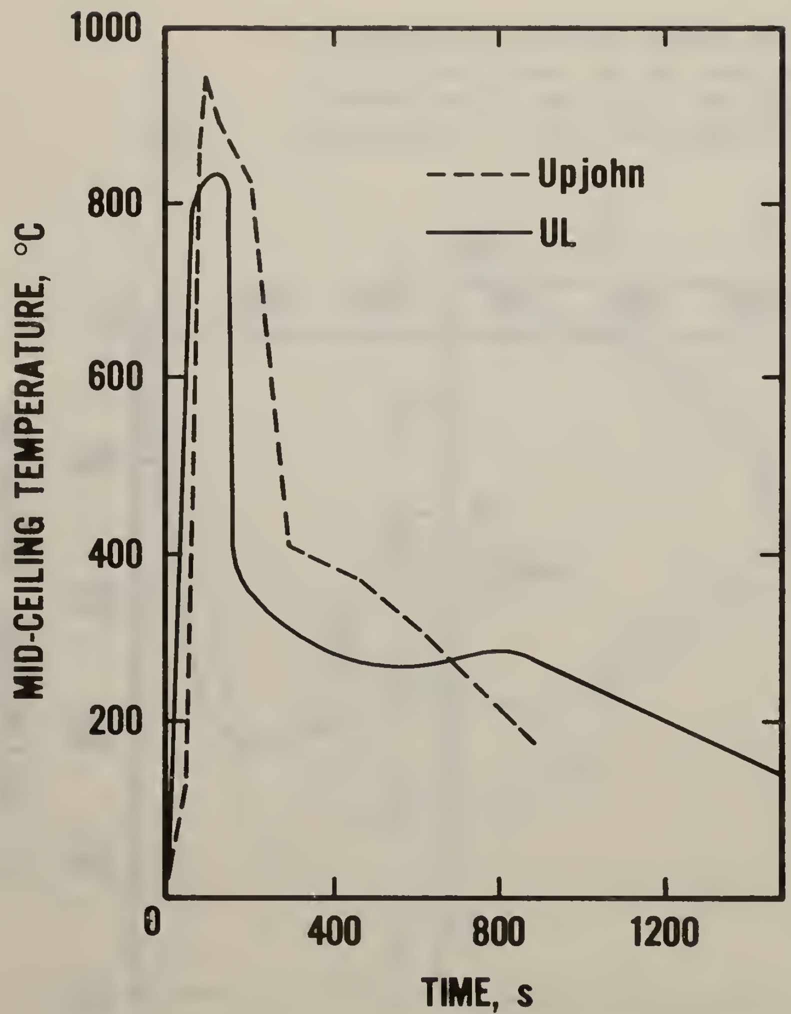

Figure 9. Comparison of Upper Air Temperature Histories in the Full- and Third-Scale Room Corner Tests of a Foil-Faced Polyisocyanurate

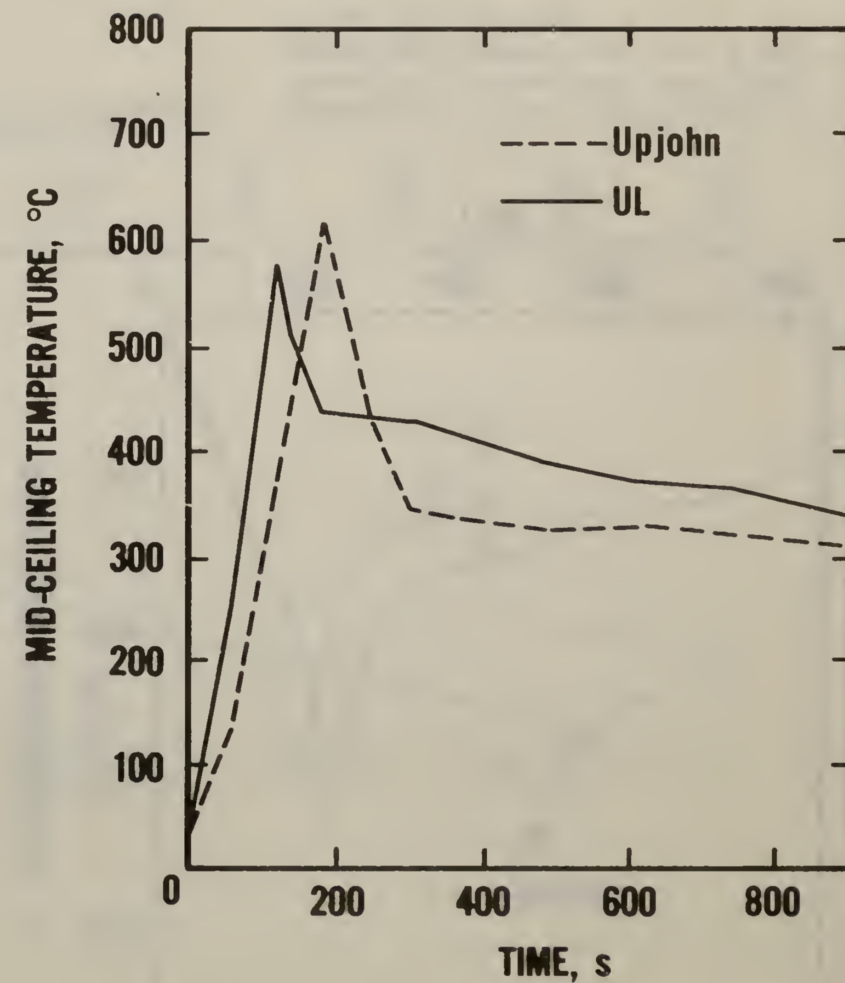

Figure 10. Comparison of Upper Air Temperature Histories in the Full- and Third-Scale Room Corner Tests of a Sprayed-On Polyurethane

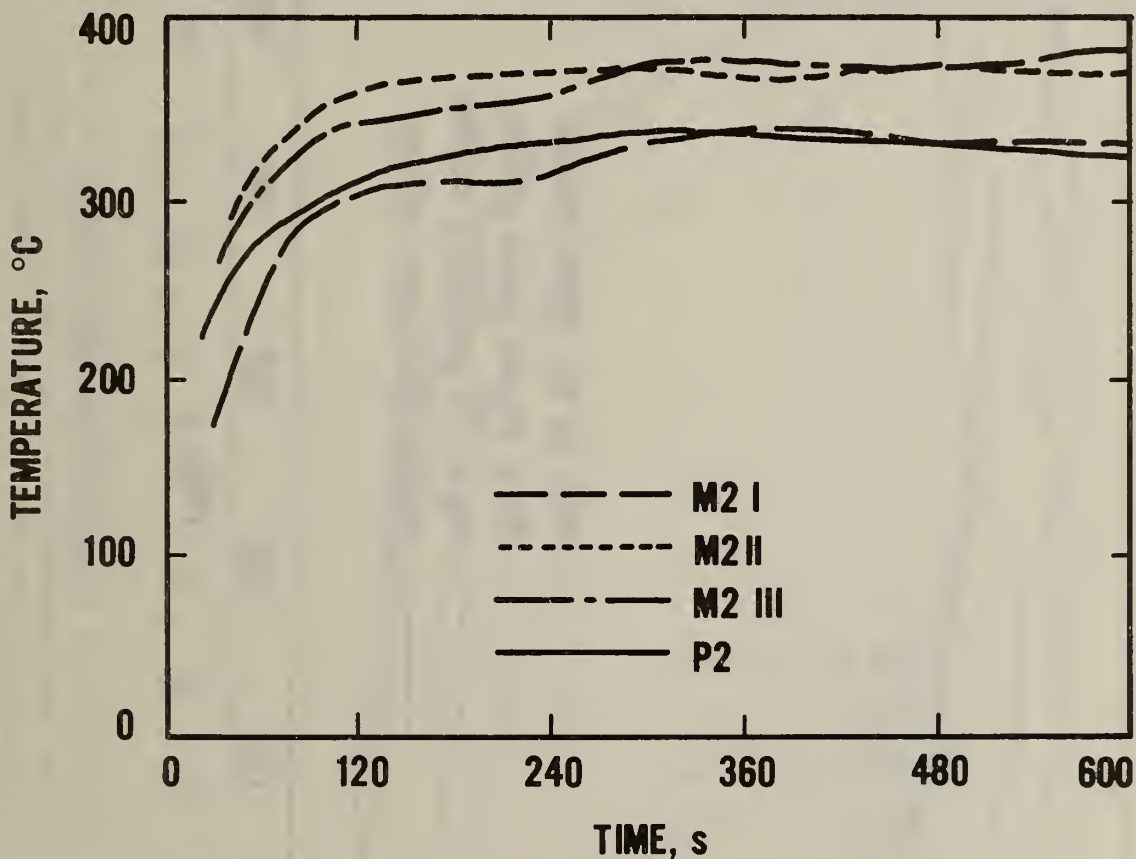

Figure 11. Comparison of Mid-Ceiling Temperature Histories for Full- and Quarter-Scale Fire Tests in the Room Lined with Fiberglass at a Scaled Burner Heat Release Rate of 140 kW

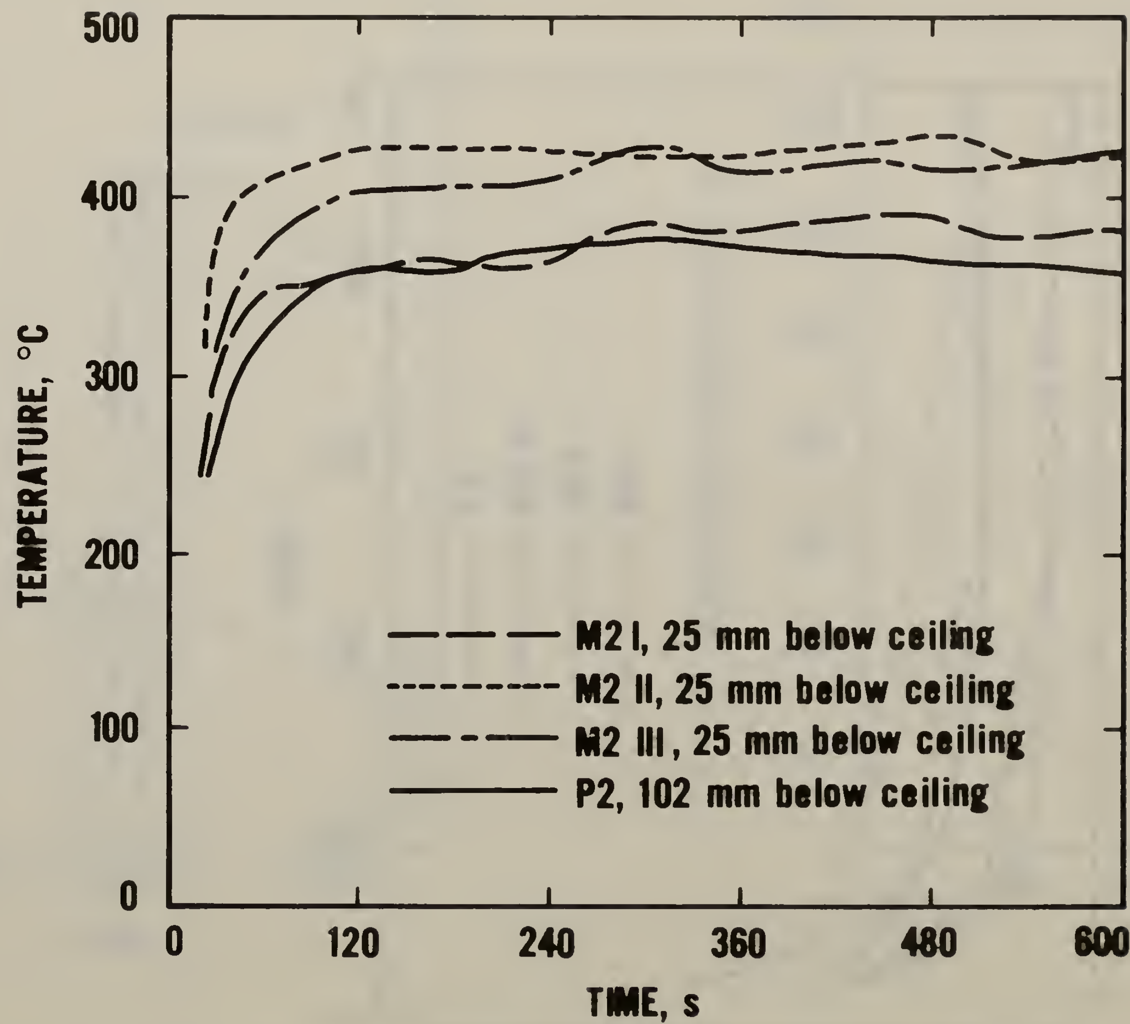

Figure 12. Comparison of Upper Air Temperature Histories for Full- and Quarter-Scale Fire Tests in the Room Lined with Fiberglass at a Scaled Burner Heat Release Rate of 140 kW

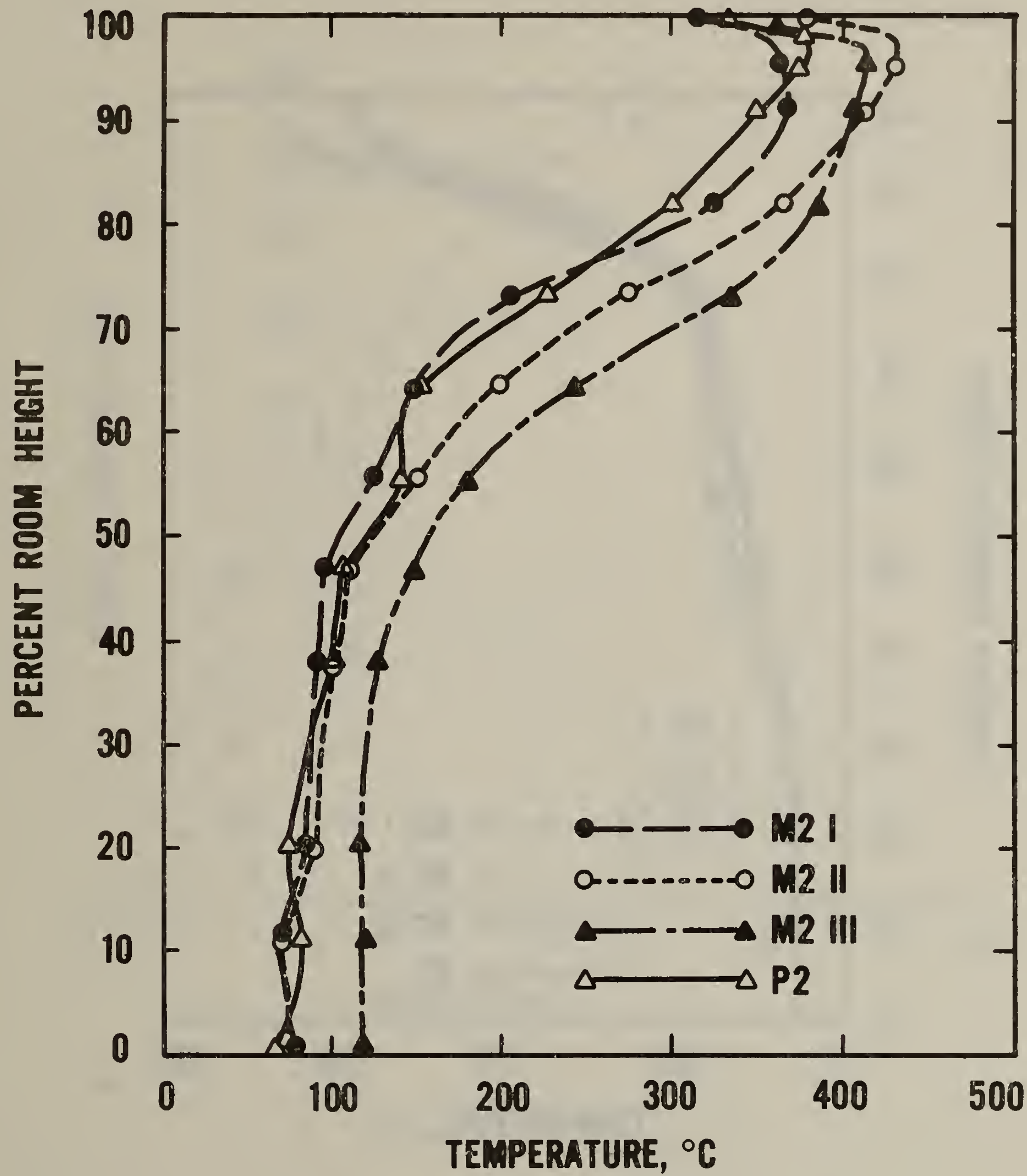

Figure 13. Comparison of Air Temperature Profiles in the Middle of the Full- and Quarter-Scale Rooms Lined with Fiberglass

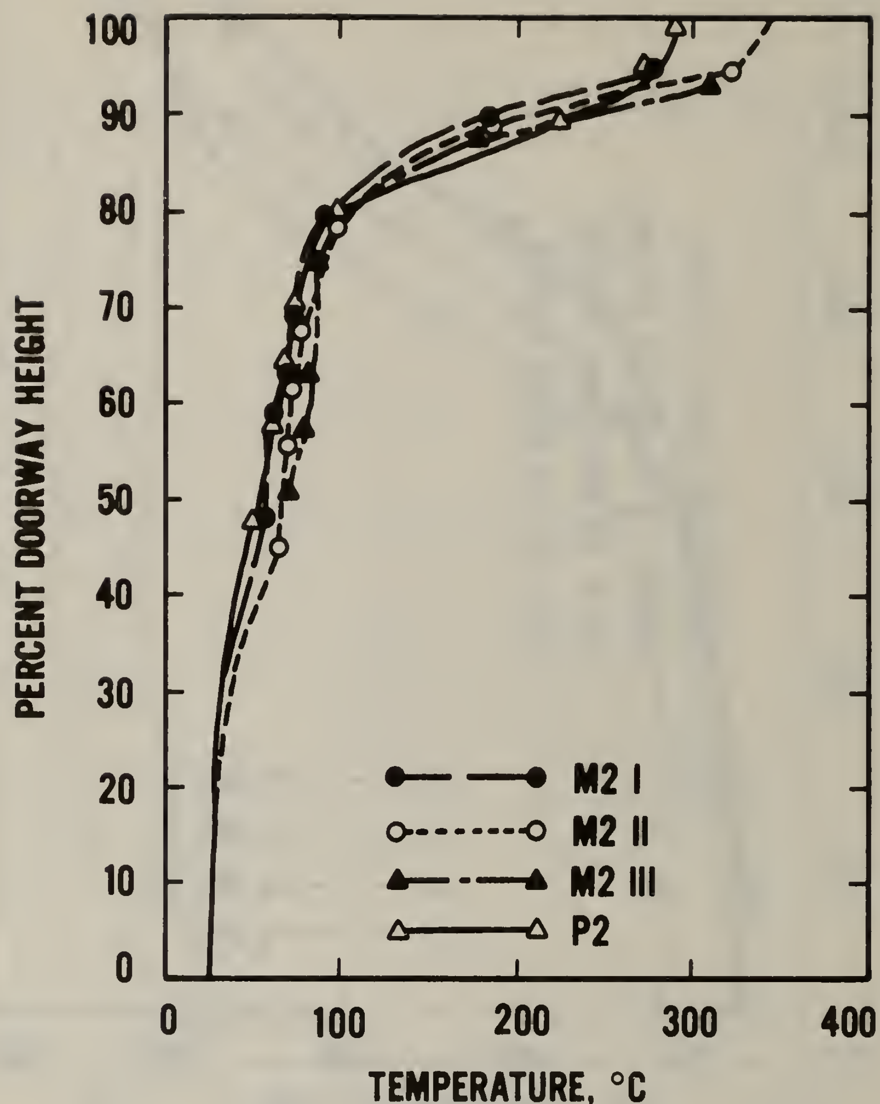

Figure 14. Comparison of Air Temperature Profiles in the Doorway of the Full- and Quarter-Scale Rooms Lined with Fiberglass

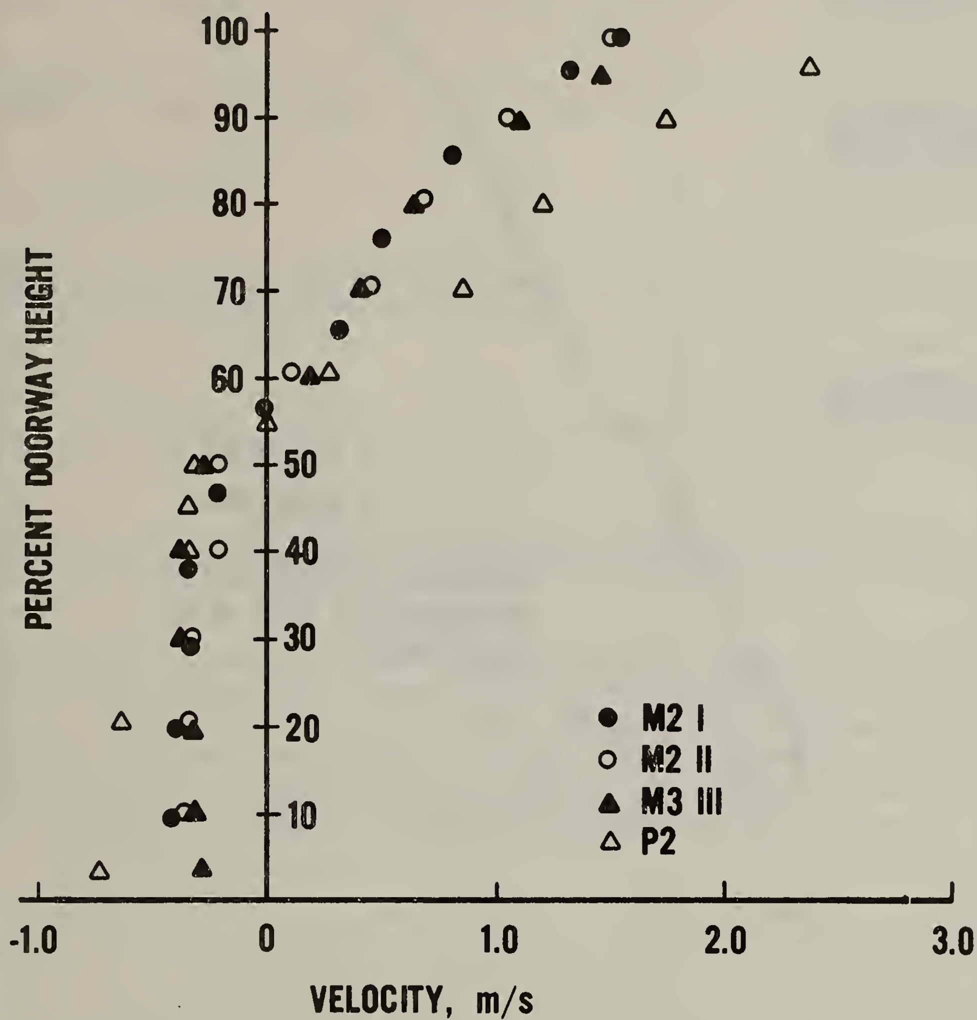

Figure 15. Comparison of Air Velocity Profiles in the Doorway of the Full- and Quarter-Scale Rooms Lined with Fiberglass

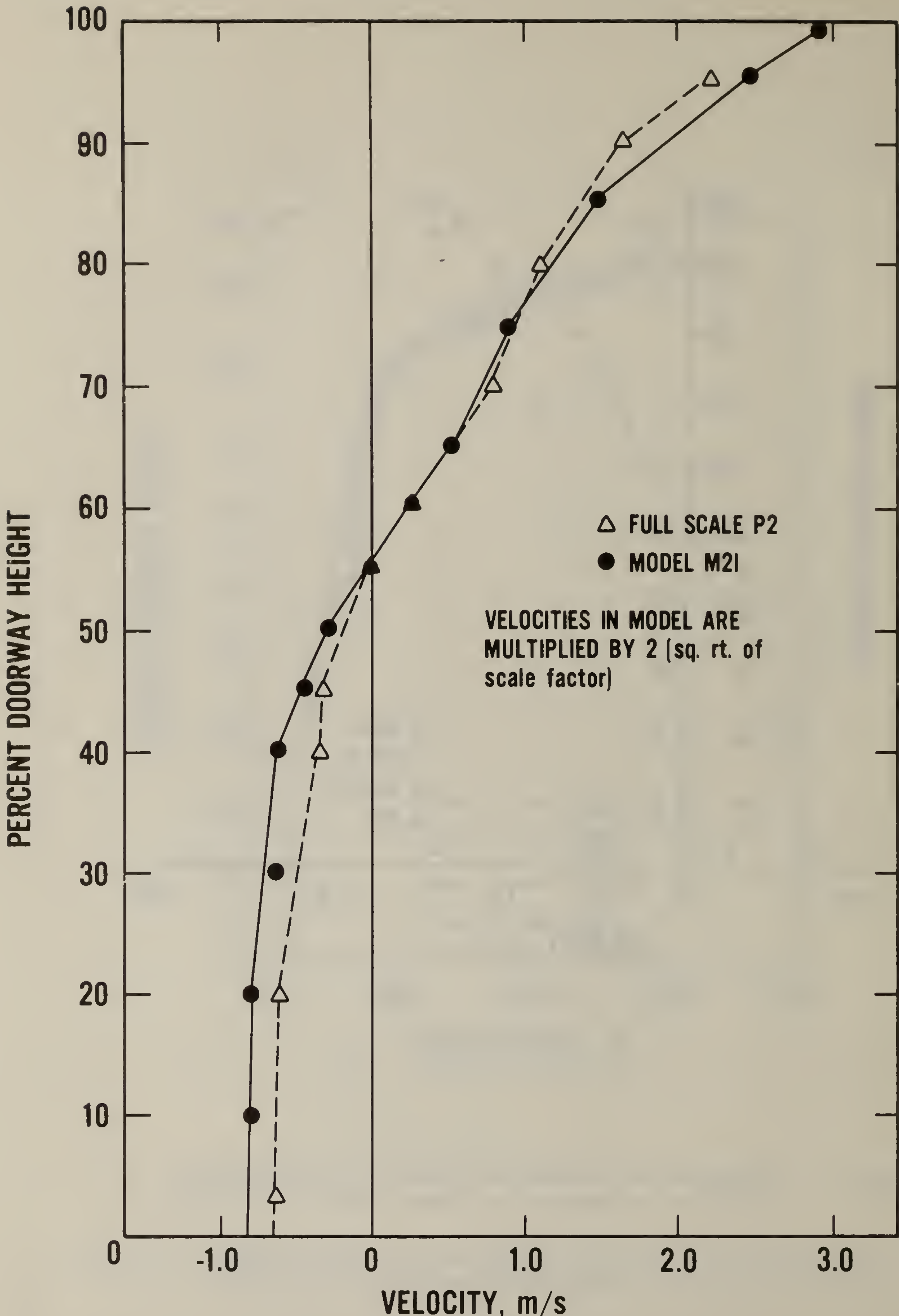

Figure 16. Comparison of Scaled Air Velocity Profiles in the Doorway of the Full- and Quarter-Scale Rooms Lined with Fiberglass

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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.)*

The temperatures, heat fluxes, air velocities, and times to flashover were compared between a number of previously reported full- and reduced-scale room fire tests. The model tests were usually similar but somewhat less severe than their full-scale counterparts. A simplified analysis is presented to account for the lower temperatures observed in the models. Some recommendations are made with regard to physical modeling of the aircraft postcrash fires.

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)*

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