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"Postflashover Fires" — An Overview of the Research at the National Research Council of Canada (NRCC) 1970 - 1985

by T.Z. Harmathy

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RÉSUMÉ

Étude du modèle du CNRC pour feux de compartiment en pleine combustion. Les mathématiques en jeu sont fort simples, mais le modèle permet une simulation assez détaillée du processus d'incendie. Il fournit une explication des conclusions voulant que le "contrôle de la ventilation" soit lié au mécanisme de la pyrolyse et ne résulte pas de la rareté de l'air dans le compartiment en feu, et que la rétroaction thermique joue un rôle secondaire dans la combustion (pyrolyse) des produits cellullosiques. L'originalité de ce modèle tient aussi à l'introduction du concept de charge calorifique normalisée. Il s'agit d'une quantité scalaire qui dépend de la chaleur globale absorbée par l'enveloppe du compartiment au cours de l'incendie, et est pratiquement indépendante de l'évolution des températures pendant l'incendie. Le modèle a été vérifié expérimentalement. Il permet de décrire de façon simple la combustion de charge calorifique normalisée. Il permet de convertir les résultats expérimentaux en résistance au feu. Il permet de prévoir la façon à ce que l'incendie se développe. C'est un modèle des incendies de plastique.

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T. Z. Harmathy

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Postflashover Fires — An Overview of the Research at the National Research Council of Canada (NRCC), 1970–1985

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(Paper submitted September 1985, accepted April 1986)

The NRCC model of fully developed compartment fires is discussed. Although the mathematics involved is quite simple, it allows a rather comprehensive simulation of the fire process. The model offers an explanation for the findings that 'ventilation control' is related to the pyrolysis mechanism and is not a result of scarcity of air in the fire compartment, and that thermal feedback is of secondary importance in the "burning" (pyrolysis) of cellulosic fuels. Another feature of the model is the introduction of the normalized heat load concept. The normalized heat load is a scalar quantity that depends on the total heat absorbed by the compartment boundaries during the fire incident, and is practically independent of the temperature history of the fire. A simple explicit formula has been proposed and proved experimentally to describe the normalized heat load for real-world fires with fair accuracy. The normalized heat load concept offers a simple means for converting fire severities into fire resistance requirements, and makes it possible to design buildings for prescribed levels of structural fire safety. The potential of fires to spread by convection and the expected characteristics of fires of noncharring plastics are also discussed.

RESearch ON POSTFLASHOVER FIRES looks back on a history of almost 30 years. Fourteen models of postflashover fires were reviewed in a recent paper;¹ there are probably a few more models not dealt with in the review. These models have, no doubt, been developed with the implicit purpose of checking the validity of various fire safety requirements prescribed by building regulations. Unfortunately, the work has not yet brought the result researchers have hoped for. Fire safety measures, especially fire resistance requirements, are still laid down by the writers of building codes on the basis of their experience and judgment. Their reluc-

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Key Words: Postflashover fires, pyrolysis, cellulosic fuels, stoichiometry, char, reactor model, burning regimes, modeling, normalized heat load, thermal absorptivity.

tance to take advantage of the postflashover fire models is understandable to some extent. The contradictory concepts reflected by the existing models have not inspired much confidence. There has also been a lack of convincing evidence to demonstrate their accuracy.

By the end of the 1970s, research on postflashover fires lost momentum as increasing numbers of researchers turned their attention to the challenging area of preflashover fires. Realizing that some important aspects of postflashover fires still awaited clarification, the NRCC researchers kept up their work on postflashover fires. The following is a summary of the more important results of that work.

Theories of postflashover fires have so far been based mainly on room-burn experiments conducted with cellulosic fuels. Consequently, the validity of some of the conclusions to be discussed is, in a strict sense, restricted to those (by no means uncommon) fires which are fueled mainly by cellulotics.

A CRITIQUE OF THE WELL STIRRED REACTOR MODEL

Postflashover fires are usually modeled (see the references listed under 1.6-1 in Table 1 of Reference 1) on the assumption that the conditions in a room containing a fully developed fire are fairly uniform; the burning compartment is, for all intents, equivalent to a well stirred reactor. This way of modeling has no doubt been instrumental in the evolution of the popular concept regarding ventilation- and fuel-bed-controlled fires. According to this concept, the fuel burns at a rate allowed by the combustion stoichiometry: the higher the rate of entry of air into the compartment, the faster the rate of burning. There is, however, an upper limit to the rate of burning, which is determined by the quantity of fuel and the nature of the fuel bed.

The stoichiometric air requirement for burning cellulotics can be estimated from an approximate formula for a typical wood:²

Assuming that all the carbon oxidizes into CO_2 , the stoichiometric air requirement is 5.11 kg air/kg fuel. If a more realistic reaction scheme is assumed, in which 40 percent of the carbon oxidizes into CO rather than CO_2 , the air requirement is 4.13 kg air/kg fuel.

It is known from hundreds of experiments that in a poorly ventilated compartment the rate of "burning" of the fuel (or more accurately: its rate of mass loss), is proportional to the rate of inflow of air. Their ratio is approximately

$$\frac{\text{rate of inflow of air}}{\text{rate of mass loss of fuel}} \approx 5.4$$

In other words, the rate of inflow of air is higher than that needed for the

stoichiometric oxidation of the fuel within the confines of the compartment. Yet, observations indicate that external flaming through broken windows or open doors is a common occurrence rather than an exception. That there is enough air and yet, that the combustion process is not completed inside the compartment, is an indication that the mixing of the gaseous pyrolysis products and the air is not perfect. This conclusion clearly contradicts the well stirred reactor model.

The well stirred reactor model can be objected to on another ground. It can be shown that some aspects of the burning* of cellulosic fuel cannot be explained unless the existence of distinct zones in the compartment is postulated.

What is loosely referred to as "rate of burning" is in fact rate of mass loss, which is an approximate measure of the rate of pyrolysis. In this light, the experimental finding concerning the rate of air flow and the rate of burning should be correctly stated as follows: in poorly ventilated fires, the rate of pyrolysis of the fuel is controlled by the air flow rate. But the air flow rate can control pyrolysis, a heat activated process, only in a roundabout way: by controlling a reaction that evolves heat which, in turn, controls pyrolysis. There is only one reaction that fits into this scheme: the oxidation of char, the solid product of the pyrolysis.

The following model has been suggested to explain the approximate proportionality between the rate of pyrolysis and the air flow rate.^{2,4,5} In poorly ventilated fires, the air entering the compartment can feed the oxidation of char only in a relatively narrow zone near the ventilation opening (window, door) before the bulk of its oxygen content is depleted. As the pyrolysis of fuel progresses, this zone, while remaining roughly unchanged in size, gradually moves toward the end of the compartment opposite to the opening. In this way the rate of mass loss by the fuel remains roughly constant as long as undecomposed fuel is present in the compartment.

According to this model, there is a zone of intense pyrolysis in the fuel bed, s2 (see Figure 1), and above that a flame zone, g2, where the volatile products of the pyrolysis intermix with air and combust. Next to these zones toward the opening are zone s1, where the oxidation of the collapsed fuel network, consisting mainly of char, takes place, and the gaseous zone, g1, occupied mainly by fresh air. Behind the zone of intense pyrolysis lies zone s3 where, under the effect of mainly radiant heat feedback, the fuel network pyrolyzes at a slow rate, and above that the gaseous zone, g3, consisting of a mixture of volatile pyrolysis products and combustion gases.

This model again contradicts the concept of the well stirred reactor. The question is now whether this model can be supported on experimental grounds. One should, of course, keep in mind that a model is only a practical tool for making a complex phenomenon accessible to mathematical treatment.

* It has been suggested³ that the word "burning" be used as an umbrella term, covering the three types of reactions that go on simultaneously in a compartment on fire: pyrolysis, gas-phase combustion, and char-oxidation.

Figure 1. Modeling of fully developed compartment fires: char forming fuels.

Figure 2. Effect of ventilation on rate of burning (change of mass) of cribs.

Two experimental findings support the model. The first concerns the claim that it is the presence of oxidizing char that makes the pyrolysis of char forming fuels respond to the flow rate of air. Figure 2 shows the rate of mass loss of piles of five different fuels as a function of air flow rate through the pile.⁶ Three of the fuels are char forming: pine and two phenolic plastics; two, PMMA and polylyte, are noncharring. Clearly, the rate of mass loss for char forming fuels increases almost in proportion to the air flow rate, while that for noncharring fuels is practically independent of it.

The second finding concerns the claim that in poorly ventilated fires a zone of intense pyrolysis exists, which moves slowly from the ventilation opening toward the opposite side of the compartment. Figure 3 shows temperatures recorded by thermocouples embedded in a pile of cellulosic fuel in a small scale room-burn experiment.⁷ A shift of the temperature levels indicates the movement of the zone of intense pyrolysis. Similar observations have been made in full scale room-burn experiments.

THE ROLE OF THERMAL FEEDBACK

The fact that the oxidizing surface char plays a significant part in the burning process for cellulose has a further implication. Clearly, if the fuel draws much of the heat needed for its pyrolysis from the oxidizing char, the rate of pyrolysis is not expected to depend strongly on thermal feedback to

Figure 3. Illustration of zonal pyrolysis in ventilation-controlled fires of cellulose in a small scale room-burn experiment. Thermocouples: 1 — near window, 2 — middle of room, 3 — near far side of room.

the fuel from the fire gases and the hot compartment boundaries.⁸ This expectation has been confirmed by experiments.

If the thermal feedback to the fuel were important, the pyrolysis would, because of the predominance of radiation in the feedback, depend on some higher power of the temperature in the burning compartment. Yet, hundreds of experiments suggest that (at low ventilation) the relationship between the rate of mass loss, R (kg s^{-1}), and the air flow rate, U_a (kg s^{-1}), is of the form

$$R \approx \text{const} \times U_a$$

not

$$R = f(U_a, \text{temperature})$$

For noncharring fuels, such as many plastics, there is no such "built-in" heat source (char oxidation), and therefore their pyrolysis relies entirely on thermal feedback. For such fuels the temperature level in the burning compartment is an important factor in the fire process.

THE TWO REGIMES OF "BURNING"

Returning to the claim of zonal burning, it is assumed that the size of the zone of intense pyrolysis is approximately proportional to the rate of inflow of air. One can expect, therefore, that if the air flow rate were gradually increased, a point would be reached where the zone of intense pyrolysis extended over the entire fuel bed. Beyond that point, the pyrolysis (rate of mass loss by the fuel) would depend on the extent of char covered surfaces (and in turn on the total surface area of the fuel), and would remain independent of the air flow rate.

Based on this reasoning, a model was formulated for the rate of mass loss by the fuel.² With input from a multitude of experimental data, the rate of mass loss was expressed as a function of ventilation and of fuel surface area, so as to allow the separation of the ventilation-controlled and fuel-surface-controlled regimes of burning. In Figure 4 experimental values of the rate of mass loss, R , are plotted against the ventilation factor, Φ (kg s^{-1}); both variables are normalized by dividing them by the surface area of the fuel, A_f (m^2). As will be shown presently, the ventilation factor is closely related to the air flow rate, U_a . Φ is minimum if the inflow of air is not augmented by wind or draft. Φ_{min} is related to the ventilation opening (window, door),

$$\Phi_{min} = \rho_a A_v \sqrt{gh_v} \quad (1)$$

where ρ_a (kg m^{-3}) is the density of air entering the compartment, A_v (m^2) is the area of the ventilation opening, h_v (m) is the height of the ventilation opening, and g (m s^{-2}) is the gravitational constant.

Part of the ventilation opening is taken up by the outflow of fire gases (combustion and pyrolysis products), which in turn is related to the rate of

mass loss by the fuel. The following approximate relationship holds between the flow rate of air, the ventilation factor, and the rate of mass loss:⁵

$$U_a = 0.138 \Phi - 0.53 R \quad (2)$$

The considerable spread of the points in Figure 4 is due to two factors: (1) Many of the experimental results used were not reported in a way suitable for obtaining all the information needed for the plot; judgment often had to be used to fill the gaps; (2) to keep the model simple, the less important variables were deliberately disregarded, such as the porosity of the fuel bed,⁹ and the temperature of the fire gases which, as discussed, is of secondary importance in the pyrolysis of cellulosic fuel.

The full line curve in Figure 4 represents the "best fit." It can be described by the following equations:

$$\text{if } \frac{\Phi}{A_f} < 0.263 \quad R = 0.0236 \Phi \quad (3a)$$

Figure 4. Correlation of experimental data concerning rate of consumption of cellulosic fuel in fully developed compartment fires.

$$\text{if } \frac{\Phi}{A_f} \geq 0.263 \quad R = 0.0062A_f \quad (3b)$$

The first of these equations describes the rate of mass loss by the fuel in the ventilation controlled regime; the second, in the fuel surface controlled regime.

The surface of the fuel, A_f , can be expressed with the aid of the specific surface area, φ ($\text{m}^2 \text{kg}^{-1}$), and the total mass of the fuel, G (kg),

$$A_f = \varphi G \quad (4)$$

The fuel mass is usually calculated from statistical data on the specific fire load, L (kg m^{-2}), and the floor area of the compartment, A_f (m^2),

$$G = LA_f \quad (5)$$

Table 1 gives some information on the specific fire load in various occupancies, based on Swedish data.¹⁰ In the Table, $\bar{L}$ (kg m^{-2}) is the mean for L , and σ_L (kg m^{-2}) is the standard deviation.

It can be seen that the existence of ventilation controlled and fuel surface controlled regimes has nothing to do with combustion stoichiometry; only fires involving char forming fuels can exhibit such dual behavior.

THE MODELING OF POSTFLASHOVER FIRES

Modeling implies scrutinizing the input variables, keeping those that seem to be the most important, and disregarding the rest. In the NRCC model,^{2,11,12} there are six input variables:

A_f	floor area of compartment, m^2
A_t	total area of compartment boundaries, m^2
h_c	height of compartment, m
$\sqrt{k_{qc}}$	thermal absorptivity (thermal inertia) of compartment boundaries, $\text{J m}^{-2}\text{s}^{-1/2}\text{K}^{-1}$
Φ	ventilation factor, kg s^{-1}
L	specific fire load (mass of fuel per unit floor area), kg m^{-2} .

TABLE 1. Information on fire load.¹⁰

Occupancy	Specific Fire Load kg m^{-2}	
	$\bar{L}$ Mean	σ_L Standard Deviation
Dwelling	30.1	4.4
Office	24.8	8.6
School	17.5	5.1
Hospital	25.1	7.8
Hotel	14.6	4.2

The specific surface area of the fuel is not included in this list. It has been taken as $0.13 \text{ m}^2 \text{ kg}^{-1}$, a quantity that applies to a wide range of conventional furniture.

Values for the first three variables are available from the building plans; the fourth is known from the planned use of building materials. The last two, ventilation factor and specific fire load, are random variables. In laboratory experiments these two are controlled (the first at its minimum value; see Equation 1) so that the output (process) variables that characterize the fire can be derived without resorting to statistical methods. An analysis of the problem^{7,11} will reveal six process variables:

- U_a rate of inflow of air, kg s^{-1}
- R rate of mass loss of fuel, kg s^{-1}
- Q rate of evolution of heat from the fuel within the compartment, W
- T_g average temperature of gases in zones g_2 and g_3 , K
- q average heat flux penetrating the compartment boundaries, $W \text{ m}^{-2}$
- τ duration of the fully developed fire, s .

Of these process variables, the first five are, in general, time dependent. To define them as functions of time, one needs six equations. These equations can then be solved by employing a numerical follow-up scheme. Three energy balances can be written: one over the ventilation opening, another over the fuel surface, and a third over the surface of the compartment. The other three equations are: one for the rate of inflow of air (Equation 2), a second for the rate of evolution of heat from the fuel, and a third for the duration of a fully developed fire. In general, these six equations are coupled.

The NRCC has used a simple technique for solving the problem.^{2,5,11} First, based on the argument that the first three of the six process variables, U_a , R , and Q , are not strongly time dependent, and that the nature of the problem does not require an accuracy higher than what can be obtained by deriving average values for the next two process variables, T_g and q , the need for a follow-up scheme has been eliminated. Second, drawing on the approximation expressed by Equations 3a and 3b, R has been decoupled from all process variables except U_a . Finally, a factor, δ (to be defined empirically), has been introduced to quantify that part of the fuel energy which is released inside the compartment boundaries. With these simplifications the problem has collapsed into the simultaneous solution of two equations, one for $\bar{q}$ ($W \text{ m}^{-2}$), the surface- and time-averaged heat flux penetrating the compartment boundaries, and another for $\bar{T}_g$ (K), the space and time averaged temperature of fire gases.

THE NORMALIZED HEAT LOAD CONCEPT

Of the six process variables, two are of particular interest: $\bar{q}$ and τ . The product $\bar{q}\tau$ is the total heat absorbed, per unit area, by the compartment boundaries during the fire incident. This product has been referred to as "heat load."¹³ Of special interest is a parameter referred to as "normalized heat load," H' ($s^{1/2}K$). It is defined as

$$H' = \frac{\bar{q}\tau}{\sqrt{k\rho c}} \quad (6)$$

where the prime is used as a reminder that H relates to real-world compartment fires, as opposed to standard test fires.

The more general definition of the normalized heat load is¹⁴

$$H = \frac{1}{\sqrt{k\rho c}} \int_0^\tau q dt \quad (7)$$

where t (s) is time. The normalized heat load is a scalar quantity which, as will be seen, is excellently suited for characterizing the severity of fires.

For two decades the term "fire severity" has been used without a clear-cut definition, with the tacit understanding that it has something to do with the destructive potential of fires. The way fire resistance tests are conducted has no doubt been instrumental in the widespread practice of regarding the temperature history of a fire as the descriptor of its severity. The temperature of the fire gases in an enclosure used to be looked upon as the embodiment of the fire itself, and the boundaries of the enclosure merely as passive participants in the fire, responding to the temperature conditions imposed upon them. This simplistic view can be objected to on two grounds. First, a fire is more than just the presence of hot gases in an enclosure; it is a process involving a strong and complex interaction between the fire gases and the enclosure boundaries. Second, since the thermal attack of the fire

Figure 5. Dimensionless presentation of the maximum temperatures attained at a point in a semi-infinite solid.

Figure 6. The influence of the nonconstancy of the penetration heat flux on the maximum temperature attained at a point in a semi-infinite solid.

gases on the enclosure boundaries is mainly by radiative heat transfer (which depends on the fourth power of the temperature of fire gases), the first power temperature history of the fire gases has no direct relevance to the destructive potential of the fire.

To understand the significance of fire severity, the following problem was studied:¹³ a semi-infinite solid is penetrated by a heat flux $\bar{q}$ for a period τ ; then the solid is allowed to cool. Using dimensional analysis, a large number of computer generated results were correlated by an equation of the following form:

$$\frac{\bar{q}\tau}{\rho c a (T_m - T_o)} = f \left(\frac{a}{\chi^{1/2} \tau^{1/2}}, h \frac{\tau^{1/2}}{\sqrt{k \rho c}} \right) \quad (8)$$

where a (m) is a distance in the solid from the surface, T_m (K) is the maximum temperature reached at that distance, T_o (K) is the initial temperature, ρ (kg m^{-3}) is the density of the solid, c ($\text{J kg}^{-1} \text{K}^{-1}$) is the specific heat of the solid, χ ($\text{m}^2 \text{s}^{-1}$) is the thermal diffusivity of the solid, and h ($\text{W m}^{-2} \text{K}^{-1}$) is the heat transfer coefficient of cooling after heat penetration has ceased.

A plot of Equation 8 is shown in Figure 5. Apparently, at a distance corresponding to a Fourier number between 0.8 and 1.2, the maximum temperature reached is practically independent of the rate of cooling, as characterized by the group $h(\tau^{1/2}/\sqrt{k\rho c})$. Picturing now the solid as the boundary of an enclosure, one will see that this distance into the solid corresponds roughly to the location of the important load bearing components (e.g., steel reinforcing bars in concrete).

The curves in Figure 5 were derived for the case of constant heat flux into the solid. What happens if the heat flux is variable? To answer this question, some further studies were conducted with heat fluxes varying linearly, as illustrated in Figure 6. The results indicate that, within reasonable limits, the heat flux history has only a minor effect on the maximum temperatures reached at any point inside the solid. This finding implies that the harm from the exposure of a solid to high temperatures depends mainly on the total heat absorbed by it, and is practically independent of the temperature history of the medium that imparts heat to the solid.

Figures 5 and 6 show that for Fourier numbers between 0.8 and 1.2

$$\frac{\bar{q}\tau}{\rho c a (T_m - T_o)} \approx 2.3 \quad (9)$$

This equation can be rearranged into the following form:

$$(H) = \frac{\bar{q}\tau}{\sqrt{k\rho c}} = 2.3 \frac{a}{\chi^{1/2}} (T_m - T_o) \quad (10)$$

where the expression on the left side is the normalized heat load.

Equation 10 suggests that the maximum temperature rise at a critical distance below the enclosure boundary depends on two factors: the normalized heat load and the thermal diffusivity of the boundary. For a given type of boundary (i.e., for a given value of κ) the maximum temperature rise will depend on H only. The normalized heat load can, therefore, be looked upon as a measure of the detrimental effect of various fires on one given type of enclosure boundary. With this interpretation, it can be used for comparing the performance of an enclosure boundary in two different fires; e.g., in a real world fire and a test fire.[‡]

The theorem of uniformity of the normalized heat load¹⁵ states that for an enclosure bounded by different materials, the normalized heat load is approximately the same for each boundary and that, in addition, it is equal to the normalized heat load for the enclosure as a whole. This theorem is expressed by the following equations:

$$\frac{1}{\sqrt{k_Q c}} \int_0^\tau q dt \approx \frac{1}{\sqrt{k_1 \varrho_1 c_1}} \int_0^\tau q_1 dt \approx \frac{1}{\sqrt{k_2 \varrho_2 c_2}} \int_0^\tau q_2 dt = \dots \quad (11)$$

where

$$\sqrt{k_Q c} = \frac{1}{A_i} (A_1 \sqrt{k_1 \varrho_1 c_1} + A_2 \sqrt{k_2 \varrho_2 c_2} + \dots) \quad (12)$$

and

$$A_i = A_1 + A_2 + \dots \quad (13)$$

and the numerical subscripts indicate values related to the various boundary surfaces.

Since the normalized heat load concept has been based on studies of heat penetration into semi-infinite solids with constant thermal properties, some researchers have questioned its general applicability. Recent studies (not yet reported) have offered the following conclusions:

1. The normalized heat load is a measure of the maximum temperature rise at some critical depth in a construction, regardless of the surface geometry of the construction. The actual value of the maximum temperature rise is, of course, dependent on the surface geometry. For example, it is higher at a location below a rectangular corner than at another location below a plane surface. One is reminded, however, that the utility of the normalized heat load is in facilitating the comparison of the performance of a given construction in various fires; it is, *per se*, not a failure criterion.

2. Under practical circumstances the finite thickness of the enclosure boundaries has hardly any effect on the value of the normalized heat load.

3. The most common use of the normalized heat load is to relate the performance of a given construction in a real world fire to its performance in a

‡ A cautionary note: the normalized heat load concept cannot be applied to compartment boundaries containing important load bearing components in, or in the immediate vicinity of, the enclosure surface. For example, it cannot be applied to enclosures built with unprotected steel.

standard test fire. It is a fair assumption that some factors not taken into consideration in quantifying H in the two fires; e.g., the nonconstancy of the thermal properties or the presence of moisture, will lead to errors of similar magnitude in two fires characterized by equal values of H .

NORMALIZED HEAT LOAD IN REAL WORLD FIRES

The model outlined in the section "Modeling of Postflashover Fires" was used to develop information on the normalized heat load in real world fires. With the aid of the iterative technique mentioned briefly in that section and described in detail in Reference 11, a large number of solutions were generated and used to find, by trial and error, an explicit expression for H' . That expression is as follows:¹²

$$H' = 10^6 \frac{(11.0 \delta + 1.6) A_f L}{A_f \sqrt{k_{qc}} + 935 \sqrt{\Phi} A_f L} \quad (14)$$

where

$$\delta = \begin{cases} 0.79 \sqrt{h_d^3 / \Phi} \\ 1 \end{cases} \quad \text{whichever is less} \quad (15)$$

The meaning of δ in these equations is not clear-cut. For poorly ventilated fires it represents roughly that part of the fuel energy which is released inside the compartment.² For well ventilated fires it is merely a correction factor which seems to improve the accuracy of Equation 14.

Seventeen full scale room burn experiments have so far been conducted¹⁶ with the purpose of obtaining answers to two questions: (1) does the NRCC model predict reasonably well the normalized heat load imposed on the room boundaries by real world fires? and (2) does the theorem of uniformity of normalized heat load hold under practical circumstances?

The experimental compartment was a cinder block room. One wall was fitted with either a door or a window. The walls, ceiling, and floor were lined with panels of selected materials. The dimensions of the room were: floor, 2.4 m by 3.6 m; height, 2.6 m. The total surface area was about 48 m².

Eight wooden cribs were used for fuel. Made from pine sticks, 4 by 4 by 84 cm, they were supported on bricks 10 cm above the floor. To drive the fire to the fully developed stage as quickly as possible, all the cribs were ignited

TABLE 2. Thermal absorptivities of lining materials used in room burn experiments.

Material	$\sqrt{k_{qc}}$ (J m ⁻² s ^{-1/2} K ⁻¹)
Normal weight concrete	2192
Brick	1260
Gypsum wallboard	742
Insulating firebrick	425
Fiberfrax board	131
Wood	436

simultaneously. The quantity of fuel per unit floor area was chosen to be representative of that found either in hotel rooms, 15 kg m^{-2} , or in offices, 27 kg m^{-2} .

Brick, insulating firebrick, fiberfrax board, gypsum wallboard, and normal weight concrete were used as lining panels. These materials were selected to span the whole range of thermal absorptivities likely to be encountered in buildings. Their thermal absorptivities are listed in Table 2. Brick, insulating firebrick, and fiberfrax board were used most frequently on account of their excellent thermal stabilities.

A summary of the experiments and the experimental results is shown in Table 3. In each experiment at least two of the five materials were present as lining. In runs A-D the floor and ceiling were lined with fiberfrax and the walls with brick; in E-J, the floor and ceiling were lined with fiberfrax, the two long walls with brick, and the two short walls with insulating firebrick; in K-M, the floor and ceiling were lined with fiberfrax and the walls with insulating firebrick; and in N-Q, the ceiling was lined with fiberfrax, the floor with normal weight concrete, and the walls with gypsum wallboard. The overall thermal absorptivity of the experimental room is also given in Table 3. These values were calculated from Equation 12, assuming that half the floor area was lined with wood.

To determine the normalized heat load, nine thermocouples were installed approximately one Fourier number deep into each of the walls, the ceiling, and the floor.

The ventilation opening was chosen to be roughly 9, 12, or 16 percent of the compartment floor area. In two cases, experiments G and J, the opening was a doorway; in all other cases it was a window.

The fire load, the overall thermal absorptivity of the boundaries, and the size of the window were varied from test to test, as shown. The last four col-

TABLE 3. Information on room burn experiments.

Experiment	L kg m^{-2}	Φ kg s^{-1}	$\sqrt{k\rho c}$ $\text{J m}^{-2}\text{s}^{-1/2}\text{K}^{-1}$	Normalized Heat Load ($10^4\text{s}^{1/2}\text{K}$)			
				Theoretical	Brick	Experimental Firebrick	Fiberfrax
A	15	3.56	868	2.76	2.97	—	3.41
B	27	3.56	868	4.44	4.52	—	4.71
C	27	5.49	866	4.09	4.31	—	4.83
D	15	5.49	866	2.53	2.91	—	3.39
E	15	3.43	666	3.27	3.48	3.34	3.87
F	15	5.25	667	3.05	3.27	3.30	3.75
G	15	7.93	668	2.61	3.00	2.97	3.47
H	27	3.43	666	5.26	4.99	4.88	5.17
I	27	5.25	667	4.69	4.68	4.66	—
J	27	7.93	668	4.15	4.29	4.30	—
K	15	3.43	344	4.60	—	4.14	—
L	27	5.25	344	5.92	—	5.38	—
M	15	5.25	344	4.12	—	3.77	—
N	27	3.31	725	5.17	—	5.13	—
O	27	3.31	725	5.17	—	5.17	—
P	27	3.31	725	5.17	—	5.34	—
Q	27	3.31	725	4.17	—	4.01	—

umns in Table 3 give the values of the normalized heat load, both as calculated from Equation 14, and as calculated from the maximum temperature rise in three different lining materials using Equation 10. Clearly, there is fair agreement between the theoretical and the experimental values. The agreement was even better if, instead of Equation 14, the iterative technique was used to calculate the six process variables and, in turn, the normalized heat load.

The experimental values of the normalized heat load with respect to brick, insulating firebrick, and fiberfrax panels, are compared in Figures 7, 8 and 9 with theoretical values. The agreement is within ± 10 percent for

Figure 7. Comparison of measured normalized heat loads and model predictions for brick. Points falling between the dashed lines represent an agreement better than $\pm 10\%$ between experiment and theory.

Figure 8. Comparison of measured normalized heat loads and model predictions for insulating firebrick. Points falling between the dashed lines represent an agreement better than $\pm 10\%$ between experiment and theory.

Figure 9. Comparison of measured normalized heat loads and model predictions for fiberfrax board. Points falling between the dashed lines represent an agreement better than $\pm 20\%$ between experiment and theory.

Figure 10. Comparison of measured normalized heat loads absorbed by different lining materials. Fiberfrax compared with insulating firebrick (Δ), fiberfrax with brick ($\square$), insulating firebrick with brick ($\circ$).

both brick and insulating firebrick, despite the large differences in their thermal properties. For fiberfrax board the agreement is somewhat poorer, ± 20 percent.

All but one of the experiments described in Table 3 represented ventilation controlled fires. Seven more experiments have been carried out since the completion of the first series, with conditions set to produce fuel surface controlled fires. The results of these latter experiments (not yet reported) have revealed that the predictive capability of the NRCC model is somewhat lower under such conditions.

The information presented in Figure 10 proves the approximate validity of the theorem of uniformity of normalized heat loads. In this figure normalized heat loads, as determined from the maximum temperature rise, are compared for three pairs of lining materials: fiberfrax and insulating firebrick, fiberfrax and brick, and insulating firebrick and brick. Values for the material with the lower thermal absorptivities are plotted along the y-axis.

There are some signs, however, that the accuracy yielded by the NRCC model, using either the iterative technique or the approximate expression, Equation 14, is somewhat fortuitous. The problem is the δ factor. δ was originally introduced to quantify the fraction of energy released by combustion of the pyrolysis gases inside the compartment. On the reasoning that this fraction is governed by the rate of entrainment of air into the pyrolysis gases, two sets of semi-empirical equations² were suggested for δ .^{*} Results of a series of experiments performed in Britain^{17,18} seemed to support the validity of these equations. Unfortunately, the room burn experiments reported in Table 3 did not fully bear out their validity. Flaming above the ventilation opening was observed in all experiments, even though the semi-empirical equations yielded, in some cases, a value of 1, suggesting that the combustion of the combustion gases was completed inside the room.

At this time, inadequate understanding of the energy release outside the fire compartment is the weakest point of all models of postflashover fires.

NORMALIZED HEAT LOAD IN STANDARD FIRE TESTS

Since the temperature history of standard test fires is defined and unique, one would expect that the normalized heat load for standard test fires, H'' (two primes are used as a reminder that H relates to standard test fires) will depend on a single input variable: the duration of the test fire, τ'' (h). The curves in Figure 11 show that the thermal absorptivity of the test specimen has also some effect on the normalized heat load. (In addition, there are some other factors of lesser importance; they are discussed in Reference 15.) Since the curves represent specimens with a wide range of thermal absorptivities, from 86 (mineral wool) to $2200 \text{ J m}^{-2}\text{s}^{-1/2}\text{K}^{-1}$ (normal

^{*} Equation 15 represents one of these sets. It was originally recommended for ventilation controlled fires only. When used in connection with Equation 14, δ becomes basically an empirical correction factor.

weight concrete), one may conclude that the effect of thermal absorptivity on H'' is not very significant, and that representing the normalized heat load versus testing time relation with a single curve is, therefore, an acceptable approximation. Curve 1 in Figure 12 is such a curve. It was calculated on the assumption that the test is performed in a high efficiency furnace.

For real world test furnaces the curve will follow a flatter course. Curve 2 in the same figure represents the normalized heat load versus testing time relation for the NRCC floor test furnace. That curve can be approximated by the following equation:

$$\tau'' = 0.11 + 0.16 \times 10^{-4} H'' + 0.13 \times 10^{-9} (H'')^2 \quad (16)$$

where τ'' is in hours.

DESIGNING FOR A PRESCRIBED LEVEL OF STRUCTURAL FIRESAFETY

The fire resistance of a compartment boundary is quantified by τ'' , the duration of a test fire successfully withstood by a specimen of that compartment boundary. Clearly, the boundaries for a compartment subject to a normalized heat load H' should be selected (from among fire tested constructions) in such a way as to satisfy the condition $H'' \geq H'$. It appears, therefore, that finding the fire resistance requirement is as simple as substituting H' (real world fire) for H'' (test fire) in Equation 16. In reality,

Figure 11. The H'' versus τ'' relation for standard fire tests conducted in high-efficiency fire test furnaces.

the problem is more complex. As previously mentioned, in Equation 14 the specific fire load, L , and the ventilation parameter, Φ , are random variables, and consequently H' is also random. Furthermore, due to differences in furnace construction, heating fuels, workmanship of test specimens, and loading and restraining conditions, the fire resistances yielded by standard fire tests (as characterized by the duration of test fire, τ'') are subject to some uncertainties.

With respect to the effect of ventilation, a glance at Equation 14 will reveal that H' decreases monotonically with an increase in the ventilation factor, Φ . Since fire safety decisions must be made on the assumption of reasonably adverse conditions, it would seem logical to select Φ_{min} (Equation 1) as the design value for Φ . This selection is supported by the finding that H' at Φ_{min} is rarely more than twice that which may arise at very large values of Φ , such as those at draft augmented ventilation.

The other random variable is the specific fire load, L . Although it is possible to find an extreme value for L (assuming that the compartment is filled up from floor to ceiling with combustibles), using this value (about 900 kg m^{-2}) in the design would hardly be justified. According to statistical data (see Table 1), the mean specific fire load in a building is usually between 15 and 30 kg m^{-2} , and rarely exceeds 100 kg m^{-2} . Clearly, the design should be based on probabilistic considerations.

Regarding the randomness of the results of standard fire tests, τ'' , a recent ASTM study¹⁹ indicated that the coefficient of variation for τ'' is

Figure 12. Correlations between H'' and τ'' for standard fire tests. Curve 1 — high-efficiency furnace; Curve 2 — floor test furnace in DBR/NRCC laboratory (estimated).

somewhere between 0.02 and 0.15, depending on the type of construction.

The task, then, is to assign a design value to the fire resistance requirement, τ_d (h), so as to ensure that in a real world fire the probability of failure of the compartment boundaries will remain below a specified level. The following design formulas have been derived,²⁰ using the technique referred to as second-moment analysis:^{21,22}

$$\tau_d = 0.11 + 0.16 \times 10^{-4} \bar{H}'' + 0.13 \times 10^{-9} (\bar{H}'')^2 \quad (17)$$

where

$$\bar{H}'' = \bar{H}' \exp \left[\beta \sqrt{\left(\frac{\sigma_{H''}}{\bar{H}''} \right)^2 + \left(\frac{\sigma_{H'}}{\bar{H}'} \right)^2} \right] \quad (18)$$

and in the latter equation,

$$\frac{\sigma_{H'}}{\bar{H}'} = \frac{\sigma_L}{\bar{L}} \frac{A_i \sqrt{k \rho c} + 467.5 \sqrt{\Phi_{min} \bar{L} A_F}}{A_i \sqrt{k \rho c} + 935 \sqrt{\Phi_{min} \bar{L} A_F}} \quad (19)$$

and

$$\frac{\sigma_{H''}}{\bar{H}''} = 0.9 \frac{\sigma_{\tau''}}{\bar{\tau}''} \quad (20)$$

where the σ 's are standard deviations for the variables shown in the

Figure 13. β as function of P_f .

subscripts, the bars over the symbols indicate mean values, and β is a factor (dimensionless), the value of which depends on the specified allowable failure probability, P_f (dimensionless), as shown in Figure 13. $\sigma_r/\bar{\tau}$ is the coefficient of variation for the results of standard fire tests and, as mentioned before, usually $0.02 < \sigma_r/\bar{\tau} < 0.15$.

In keeping with the earlier recommendation, the minimum value of the ventilation factor, Φ_{min} , is used as the design value in Equation 19. Figure 14 shows the effect of increasing ventilation on the fire resistance requirement for a particular construction. For a 5 percent allowable failure probability, the fire resistance requirement for this construction was calculated as 1.23 h. The figure suggests that drafts or windy conditions may lower the fire resistance requirement by as much as 50 percent.

CONVECTIVE SPREAD POTENTIAL

So far, the potential of fires of cellulose to spread by destruction has been discussed, together with the design procedure to counter such a spread by using fire resistant compartment boundaries. It is well known, however, that fires rarely spread by successive destruction of the compartment boundaries. The most common mechanism of spread is convection: the advance of flames and hot gases through windows, open or burned out doors, or any other opening. Irrespective of the nature of the surface lining materials in the spaces adjoining the fire compartment, fuels and conditions conducive to massive combustion of pyrolysis gases outside the compart-

Figure 14. The effect of ventilation on the fire resistance requirement at a given failure probability.

ment boundaries present a very real danger from the point of view of fire spread.

A factor has been introduced to characterize the convective spread potential of fires. It is denoted by μ and defined as the ratio:

$$\mu = \frac{\text{rate of evolution of heat outside fire compartment}}{\text{total rate of heat evolution from fuel}} \quad (21)$$

Once the process variables for the fire are known (from calculations using the iterative technique, as described in Reference 11), μ can be evaluated. For cellulosic fuel, an approximate expression for μ is

$$\mu = 0.873 (1 - \delta) \quad (22)$$

Although the potential of fires to spread by convection can be quantified by the μ -factor, there is as yet no established method on how to use its value in fire safety design. An acceptable approach would be to require some extra measures of countering the convective spread potential whenever μ exceeds some specified limit, say 0.4. A number of such measures have been described in Reference 23.

FIRE INVOLVING NONCELLULOSIC FUELS

For reasons discussed in the section "The Role of Thermal Feedback," char forming plastics can be expected to behave on the whole similarly to

Figure 15. Modeling of fully developed compartment fires: pool fires of liquids and quasi-liquids.

cellulosics. One can expect, therefore, that (1) the rate of mass loss by the fuel will not strongly depend on the temperature level in the fire compartment, (2) in calculating the process variables, the rate of mass loss may be decoupled from all the process variables except the flow rate of air, and (3) the burning of the fuel will be ventilation controlled or fuel surface controlled, depending on the level of ventilation.

The development of a mathematical model for noncharring plastics seems to be a challenging problem. Since the pyrolysis of such materials relies on thermal feedback from the hot gases and compartment boundaries, the rate of mass loss by the fuel can be expected to depend on that part of the fuel surface which sees the hot gases and compartment boundaries, rather than, (as with cellulosics) on the total fuel surface.

A model has been suggested⁵ for a very simple case: the burning of a plastic fuel in a pool configuration on the floor of a compartment. Figure 15 shows the main features of the model. Again, the existence of six zones has been postulated which are somewhat similar to those discussed in connection with the model for cellulosics (see Figure 1). The main difference is that, although zone s1 is still the zone of intense pyrolysis, the pyrolysis of the fuel in zone s2, under the effect of (mainly) radiant feedback from the hot gases and compartment boundaries, is not insignificant. The result is a very high rate of production of pyrolysis gases, especially if the pyrolysis temperature of the fuel is relatively low.

Since the entry of air into the fire compartment is restricted by the ventilation opening, a large portion of the pyrolysis products will, by necessity, leave the compartment uncombusted. Although, from the point of view of the heat load on the boundaries of the fire compartment, it is advantageous to have a major portion of the pyrolysis gases burn outside the compartment, outside burning is clearly extremely dangerous and undesirable from the point of view of convective fire spread.

The model developed for pool burning⁵ was used to generate information on the spread potential of noncharring plastics, as compared with cellulosics. The following conclusions have been drawn:

1. From the point of view of potential for destructive spread, fires of cellulosics are usually more dangerous than fires of noncharring plastics. From the point of view of potential for convective spread, the opposite is true.
2. For both types of fuel, the destructive spread potential decreases with increasing ventilation.
3. The convective spread potential of fires decreases with increasing ventilation if the fire load consists of noncharring plastics, and increases with increasing ventilation if the fire load consists of cellulosics.

Whether these conclusions are justifiable will be seen when the results of a series of room burn experiments using noncharring plastics, now in the planning stage, become available.

NOMENCLATURE

a	distance in the solid from the surface, m
A	area, surface area, m^2
c	specific heat, $J\ kg^{-1}K^{-1}$
g	gravitational acceleration, $\sim 9.8\ m\ s^{-2}$
G	total mass of fuel, kg
h	(with subscript) height, m
h	(without subscript) heat transfer coefficient, $Wm^{-2}K^{-1}$
H	normalized heat load, $s^{1/2}K$
H'	normalized heat load for real world fires, $s^{1/2}K$
$\bar{H}'$	mean of H', $s^{1/2}K$
H''	normalized heat load for test fires, $s^{1/2}K$
$\bar{H}''$	mean of H'', $s^{1/2}K$
k	thermal conductivity, $W\ m^{-1}K^{-1}$
$\sqrt{k\rho c}$	thermal absorptivity (thermal inertia), $J\ m^{-2}s^{-1/2}K^{-1}$
$\underline{L}$	specific fire load, $kg\ m^{-2}$
$\bar{L}$	mean of L, $kg\ m^{-2}$
q	heat flux penetrating the surface, $W\ m^{-2}$
$\bar{q}$	time and space averaged value of q, $W\ m^{-2}$
Q	rate of heat evolution from the fuel within the compartment, W
R	rate of mass loss (rate of "burning"), $kg\ s^{-1}$
t	time, s
T	temperature, K
$\bar{T}$	time and space averaged value of T, K
U	mass flow rate, $kg\ s^{-1}$

GREEK LETTERS

δ	(loosely) fraction of energy of gaseous pyrolysis products released within the compartment, dimensionless
κ	thermal diffusivity, m^2s^{-1}
μ	factor characterizing the convective spread potential of fires, dimensionless
ρ	density, $kg\ m^{-3}$
σ	standard deviation, various dimensions
τ	duration of fully developed fire, duration of exposure to a heat flux $\bar{q}$, s
τ''	duration of test fire (= fire resistance time), h
$\bar{\tau}''$	mean of τ'' , h
φ	specific surface of fuel, m^2kg^{-1}
Φ	ventilation factor, $kg\ s^{-1}$

SUBSCRIPTS

a	of air entering the compartment
c	of the compartment
d	design
f	of the fuel
F	of the floor
H'	pertaining to H'
H''	pertaining to H''
g	of the fire gases in zones g2 and g3
L	pertaining to L
m	maximum
min	minimum
i	initial
t	total for the compartment
v	of the ventilation opening
'	pertaining to τ''

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