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A discussion of an earlier paper by T. Z. Harmathy.

IN THE August and November 1972 issues of FIRE TECHNOLOGY appeared a two-part paper by T. Z. Harmathy titled "A New Look at Compartment Fires."[1] The paper gives a clear, concise and valuable summary of the present knowledge of the heat balance equation and other basic characteristics of fire development in a compartment. We feel, however, that a discussion in some detail of Harmathy's approach to the solution of heat balance is justified. Our reasons for writing this discussion are threefold.

PURPOSES OF PAPER

Due to an unfortunate communication delay, no reference is made in Reference 1 to a paper published by the authors in 1971, "Comments on the Rate of Burning and Rate of Gas Flow for Compartment Fires."[2] In the latter paper, some twenty model and full-scale experimental fires from laboratories in England, France, and Sweden were theoretically analyzed. One of the purposes of the investigation was to determine how the rate of burning and rate of gas flow are influenced by a transformation of fire development from the ventilation controlled to the fuel bed controlled regime. As the two papers[1,2] thus are related to the same problem area and, in fact, partly analyze the same experimental fires, a discussion of the two approaches presented might be of common interest.

Mention is made in Reference 1 to calculations (Part II, pages 343–344) made earlier by the authors.[3] The purpose of that work was to examine the possibility of computing the gas temperature-time curve of the complete process of fire development, the decay period included; and in that way, lay the basis for a theoretical design procedure for fire-exposed structural members. To this end, gas temperature-time curves were calculated for fire compartments with different fire loads, openings, and materials in the structures surrounding the fire compartment. These curves, which presuppose ventilation control, are intended to be used for design purposes. They are meant to give an upper bound of the fire severity for a specified fire compartment (fire load, openings, and thermal characteristics of floor, walls, and ceiling) and are not intended to be used, as they were in Reference 1, to describe individual fire developments in the fuel bed controlled regime.

In Sweden, a structural fire engineering design, based directly on the gas temperature-time curve for each specific case, is explicitly permitted by the building code. (See Reference 4, Chapter 4.1.) This means that the problem of ventilation control vs. fuel bed control is by no means only of academic interest. If the beneficial consequence of fires with a low ratio of fire load to ventilation openings having lower maximum rate of burning could be used throughout, the economic savings would be substantial. However, to make it possible for the practicing engineer to use this differentiated approach, all the design curves in the manuals published[5] are based on the assumption that fire development is ventilation controlled for all ratios of fire load to ventilation openings. Evidently, the reasons for choosing this alternative have to be expounded. In this connection, it is important to study the reliability of the quantities used by Harmathy to characterize fire severity, $\bar{q}_E$, τ, and T_g, as design parameters.

In the first part of this paper, a brief description is given of the simulation technique developed and employed by the authors.[2,3] A term by term comparison is made between this technique and the approach made by Harmathy.[1] After that the reasons why, at present, the ventilation controlled fire process has to be chosen as a basis for structural design purposes are explained. Some examples are presented to illustrate the influence on fire severity of variations in the rate of burning and the rate of air inflow for a given compartment with a given fire load. The last section is devoted to a discussion of the relevance of the fire severity parameters $\bar{q}_E$, τ, and T_g, introduced in Reference 1. The paper ends with a summary.

THEORETICAL SIMULATION OF NATURAL FIRES IN COMPARTMENTS

The physical basis for the theoretical simulation of compartment fires is the heat balance equation

$$q_t = q_g + q_E + q_R \tag{1}$$

where q_t = rate of heat release by combustion inside the compartment; q_g = rate of heat loss by convection in the openings; q_E = rate of heat loss through bounding walls, floor, and ceiling; and q_R = rate of heat loss by radiation through the openings. Equation 1 expresses momentary energy conservation; and if expressions for all the terms are known, the gas temperature at any given instant can be obtained. Of the four quantities, the right-hand terms are more or less approximately given by known heat transfer laws. For q_t, however, no general physical relation exists. The complexity of the combustion process restricts our advance, presimulation knowledge of the rate of heat release to a few general basic assumptions. The first is that the energy balance of the total fire process requires q_t to obey the following relation:

$$\int_{o}^{\infty} q_t \bullet d_t = G_o \bullet H_f \tag{2}$$

where G_o = initial weight of fuel, and H_f = effective net heat value of the fuel. The second, commonly accepted and experimentally verified assumption regards the maximum rate of burning R_{max}.

Somewhat idealized, a compartment fire can be characterized as ventilation controlled or fuel bed controlled, depending on what factors are determining the maximum rate of burning. In the ventilation controlled regime, this rate is proportional to the ventilation factor, $A_W \bullet h^{\frac{1}{2}}$, irrespective of fire load density and properties of the fuel bed. For a given compartment, ventilation control implies an upper bound for the rate of energy release.

In the fuel bed controlled regime, however, the rate of burning will be determined by a number of parameters. The most significant of which is the exposed surface area of the fuel, A_f; but specific fuel bed properties, such as average thickness, spatial orientation, and porosity also play an important role. Thus, in this regime, the maximum rate of energy release can vary from almost zero up to the value corresponding to ventilation control.

The consequences of this are illustrated in Figure 1, where two possible $q_t - t$ curves are shown for a given combination of fire load and ventilation opening. Assuming that the maximum level of Curve 1 is determined by the rate of air supply, it follows that this curve corresponds to a ventila-

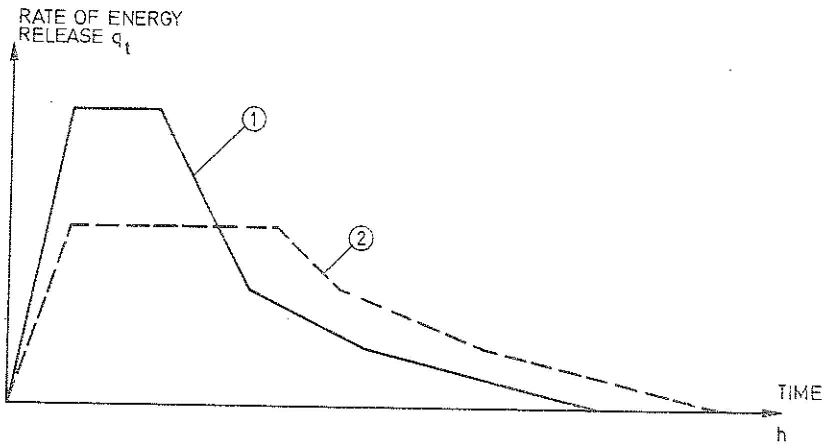

Figure 1. Rate of energy release rate as a function of time for a ventilation controlled (Curve 1) and a fuel bed controlled fire (Curve 2). Ventilation factor, $A_w \bullet h^{\frac{1}{2}}$, and fire load are the same in both cases.

tion controlled fire process. If the geometric properties of the fire load display are changed in such a way that the process becomes fuel bed controlled (See Curve 2.), the maximum level of q_t will be lower but the duration longer, so that Equation 2 is still satisfied. The change from Curve 1 to Curve 2 could be produced by an increase in the average thickness of the fuel, for example. Obviously, one of the main purposes for an analysis of compartment fires must be to investigate which q_t curve is valid for the individual combination of ventilation opening, fire load density, and fuel bed display.

To examine the reliability of the theoretical model and to determine the corresponding $q_t - t$ curves, a number of experimental full-scale and model fires were numerically simulated.[2,3] The experimental basis was taken from tests made in England, France, Japan, and Sweden.

For each test, the $q_t - t$ curve was chosen on trial and then adjusted until full agreement was obtained between theoretical and experimental temperature-time curves. Figure 2a shows the results from such analyses for Tests, P, D, N, and L in the full-scale tests performed by the Joint Fire Research Organization (JFRO) in the United Kingdom[6,7] (cf[1], Figure 10, page 344). Figure 2b shows the proportions between the terms in Equation 1 for Test D. In Reference 2, more results of this kind can be found.

Generally speaking, this analysis showed that the applied model is accurate and relevant for the simulation of natural fires. In particular, the following conclusions could be drawn from the analysis.

• In the majority of cases, it was possible to find a time distribution of rate of energy release, q_t, such that the theoretical and experimental temperature-time curves agreed simultaneously as Equation 2 was satis-

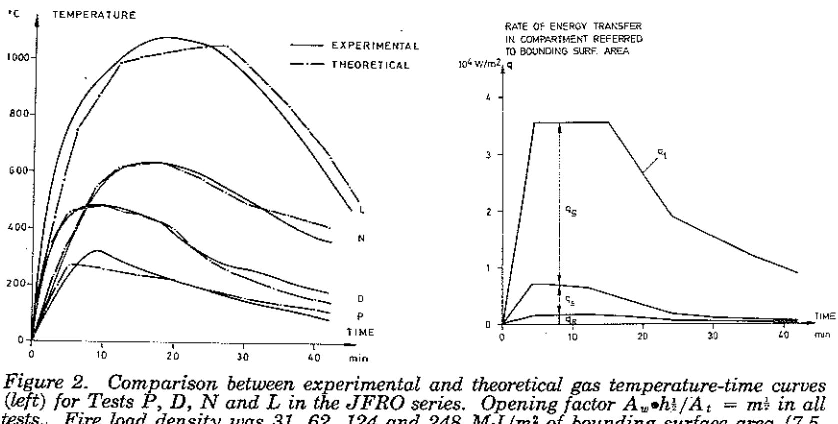

Figure 2. Comparison between experimental and theoretical gas temperature-time curves (left) for Tests P, D, N and L in the JFRO series. Opening factor $A_w \bullet h^{\frac{1}{2}}/A_t = m^{\frac{1}{2}}$ in all tests. Fire load density was 31, 62, 124 and 248 MJ/m^2 of bounding surface area (7.5, 15, 30 and 60 kg/m^2 of floor area) respectively. The time curve of the different terms in the heat balance equation for Test D are shown at the right.

fied. Provided that the terms of the heat balance equation are accurate, this means that the combustion is complete and takes place solely within the compartment.

• For some tests with large openings — for instance the JFRO tests with half the front wall open — the given amount of energy appeared to be too small to give agreement between the test results and theory. Our conclusion was that the gas flow model used for calculating the convective heat loss, q_g in Equation 1 (cf Reference 7, page 12), was not correct for large openings. According to Thomas et al.,[8] that model results in too high values for the convective gas flow in the openings, and when the flow is reduced by a factor of 0.8 (0.6 in some model-scale tests) agreement was achieved for these tests also.

• For a few tests with very small openings and high fire load densities, the given energy was excessive. The theoretical temperatures appeared to be too high whatever q_t − t curve was taken. This indicates that combustion in these cases was incomplete or, in part, took place outside the compartment.

• In all cases, the ratio between q_t and the measured rate of weight loss was in the range of 9 to 13 $MJ.kg^{-1}$ when averaged over the period of maximum fire intensity, defined as the period during which the weight of the combustible material decreased from 80 percent to 30 percent of the initial weight. The smaller values referred to the tests with very small openings mentioned in the preceeding paragraph, indicating more incomplete combustion inside the compartment.

• For reasons that will be explained in detail later, it was concluded that a structural fire engineering design at present has to be made under the assumption that the fire process is ventilation controlled.

A COMPARISON BETWEEN APPROACHES USED IN REFERENCES 1 AND 2

Here we will compare our approach to the problem of determining the behaviour of compartment fires and that of Harmathy. In both cases, Equation 1 forms the basis of calculation. The way of solving the heat balance equation, however, differs widely. Harmathy assumes that the rate of burning, R, can be determined in advance and is constant over the period of the fully developed fire, that the pre-flashover and decay periods of the fire process can be neglected, and that the boundary elements of the compartment can be treated as semi-infinite solids. This makes it possible for him to derive expressions for the "effective heat flux," $\bar{q}_E$, defined as the heat flux available for penetration into floor, ceiling, and walls averaged over the period of the fully developed fire, and for the average gas temperature T_g, averaging applying to both space and time coordinates. Our method has been to simulate the behavior of natural fires in a step by step process, taking into account the temperature dependence of relevant parameters and constantly checking with the experimental evidence available.

Unfortunately, the experimental data available are insufficient to confirm Equation 1 term by term at every instant. This means that even if good agreement with the measured time-temperature curve is achieved in a test, it does not necessarily mean that all the terms are correctly estimated. However, since the simulation technique has been successfully applied for more than 100 tests under the most varying conditions, its reliability must be regarded as fully demonstrated.[2,3]

As the JFRO tests[6,7] have been previously analyzed theoretically by Harmathy[1] and the authors[2], they are a natural experimental basis for a comparison.

If we try to make an analogue term by term in Equation 1, Table 1 compares the results of our calculations with Harmathy's for four tests — V, M, L, and D. These tests have been selected among ventilation controlled (V) and fuel bed controlled tests (D), as well as in the boundary region between the two regimes (M and L). Harmathy's results are taken from Reference 1, and refer to averages over the "primary burning period," the duration τ of which is defined as

$$\tau = 0.936\ G_o/\overline{R} \tag{3}$$

where $\overline{R}$ = rate of burning during primary burning.

In our calculations, which were made stepwise with regard to time, the proportions between the different terms in the heat balance equation are continuously changing during time, as shown in Figure 2. For the sake of

TABLE 1. *Comparison Between Heat Balance Calculations made by the Authors and Harmathy*[1]

Heat balance terms referred to A_t		*Test*							
		V		*M*		*L*		*D*	
		H	*A*	*H*	*A*	*H*	*A*	*H*	*A*
Total heat (q_t)	$\times 10^4$ W/m²	4.97	4.35	9.72	6.20	19.45	10.1	4.86	3.56
	%	100	100	100	100	100	100	100	100
Convective heat loss (q_c)	$\times 10^4$ W/m²	2.68	2.37	4.17	3.72	7.97	6.85	3.84	2.88
	%	53.8	54.4	42.9	60.0	41.0	67.7	78.9	80.9
Radiant heat loss (q_R)	$\times 10^4$ W/m²	0.15	0.49	0.14	0.51	0.25	1.50	0.04	0.16
	%	3.1	11.3	1.5	8.1	1.3	14.8	0.9	4.5
Conductive heat loss (q_E)	$\times 10^4$ W/m²	2.13	1.49	2.96	1.98	2.78	1.72	0.98	0.52
	%	42.8	34.2	30.5	31.9	14.3	17.0	20.2	14.6
Heat loss due to outside burning (q_B)	$\times 10^4$ W/m²	0.02	–	2.44	–	8.45	–	0	–
	%	0.3	–	25.1	–	43.4	–	0	–
Fire load	kg	1744		872		1744		436	
Initial energy content referred to A_t	MJ/m²	250		125		250		62	
Window area	m²	2.8		5.6		11.2		11.2	
Opening factor	m$^{\frac{1}{2}}$	0.03		0.06		0.12		0.12	

H = Harmathy; A = Authors; A_t = 126 m²; h = 1.83 m.

comparison, our values in Table 1 are taken as averages over that period of time when the measured weight of the fuel decreases from 80 to 30 percent of the initial value.

When comparing the two alternative calculations, we can conclude that the sum of the two terms q_B and q_R is almost the same value in both calculations, while the term q_E is notably smaller according to our calculations. A more detailed discussion of the term q_E will follow in the section dealing with fire severity.

The greatest difference between our calculations and Harmathy's appears in the term q_t, which is the total heat produced per unit time by combustion. In Harmathy's calculation, this term was determined as the rate of weight loss during primary burning, $\overline{R}$, multiplied by an effective heat value, ΔH. During the period of primary burning, ΔH was given the value 17.9 $MJ.kg^{-1}$, which is very close to the calorific value of the wood. The average values of q_t arrived at in our calculations correspond to heat values within the range of 9 to 13 $MJ.kg^{-1}$, when referred to actually measured rates of weight loss. Accordingly, the average values of q_t were considerably larger in Harmathy's calculations, as can be seen in Table 1.

Despite this marked difference, Harmathy also gets temperatures that, in most cases, are in fairly good agreement with the experimental results as far as the period of maximum burning is concerned. This is due to the overestimation of the term q_E and, more important, due to the assumption that, in certain cases, a considerable portion of the energy is released outside the compartment.

In our opinion, however the energy evolved outside the compartment is negligible in most cases, and values of q_t used by Harmathy are too high.

A consequence of Harmathy's assumption for q_t is that 88 percent of the total energy stored in the fuel is consumed during the time τ given in Equation 3, which means that only 12 percent is left for the rest of the fire. From our simulations, it was evident that much more energy is required after that time if agreement between experiment and theory is to be obtained. In those particular tests where Harmathy assumes burning outside the compartment, our calculations indicated that, within the uncertainties inherent in the simulation model, all the energy had been released inside the compartment. This general trend is confirmed in the more than 100 tests simulated to date.

CHOICE OF q_t—t CURVE FOR STRUCTURAL DESIGN PURPOSES

As has been shown in the preceeding sections, the theoretical simulation model as presented in References 2 and 3 makes it possible to determine the time-gas temperature curve as well as the different terms in the heat balance equation. The computations can be made with a relatively high degree of precision and have been validated both for the ventilation and fuel bed controlled processes. In the first case, the maximum rate of energy release is given by the air inflow, making the specification of the $q_t - t$ curve of the complete process a relatively simple task. For fuel bed controlled experimental fires, the corresponding $q_t - t$ curves have to be found by a trial and error method as a rule. Any attempt to systematize the $q_t - t$ curve in this regime has to stem from the fact that the primary influence must be the size of the fuel surface area. This approach was taken in the work reported in References 1 and 2. For wood crib fires, the average thickness of the fuel can be described by the hydraulic radius, r, which is the ratio V_f/A_f between volume and surface area. By plotting $\bar{R}/A_w h^{\frac{1}{2}}$ against $G_o/(r\, A_w\, h^{\frac{1}{2}})$ for a number of full-scale tests, we obtained the diagram shown in Figure 3.[2] The inclined line denotes the fuel bed controlled regime, and the horizontal line, the ventilation controlled regime. The scatter is considerable, but the two different kinds of behavior can be recognized. An estimated value of the transition point would be

$$G_o/(_r\, A_w\, h^{\frac{1}{2}}) \approx 17{,}000 \text{ kg.m}^{-7/2} \tag{4}$$

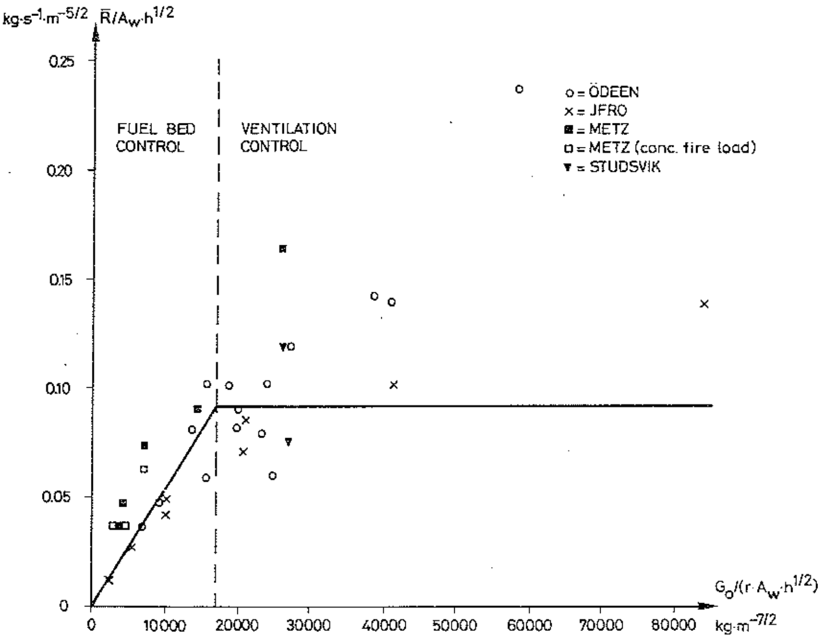

Figure 3. Relation between $R/A_w h^{\frac{1}{2}}$ and $G_o/(r\ A_w h^{\frac{1}{2}})$ for full-scale fire tests performed at various research laboratories. For the tests denoted by o, the rate of air supply was regulated by fans, which resulted in combustion characteristics partly differing from those predicted by theory. (See Reference 3, pages 48–54.)

In Reference 1, the transition point was defined by the parameter ϕA_f of which $G_o/r\,A_w\,h^{\frac{1}{2}}$) in principle is the inverse. The critical value was chosen as

$$\phi/A_f = 0.26 \text{ kg.m}^{-2}.\text{s}^{-1} \tag{5}$$

The corresponding value of ϕ/A_f found from Equation 4 would be approximately 0.13 kg.m^{-2}.s^{-1}. From Figure 3 and the analogous figure in Ref-ference 1, a value of the average rate of burning in the fuel bed controlled regime can be found. This procedure was unconditionally adopted for the approximate analysis made in Reference 1. The same method has been applied in the more precise simulation technique presented in Reference 9 to obtain the rate of energy release for those fire processes or parts of fire processes that are not ventilation controlled.

The question then arises, can the rate of burning derived in References 1 and 2 for the fuel bed controlled regime be used in a practical design procedure of fire-exposed structural members? This matter was given some consideration by the authors,[2] and the objections that were lodged against using this approach indiscriminately can be summarized as follows:

- For authentic fire loads of furniture, textiles, etc., the exposed fire surface area or hydraulic radius is usually difficult to define and cannot, for the moment, be measured with any reliability. In Reference 1, the value of the specific surface $\phi = A_f/G_o$ for furniture is stated to be in the area of $0.1 < \phi < 0.4$ m^2.kg^{-1}.
- In the fuel bed controlled regime, A_f is not the sole decisive factor for the rate of burning. The design of the fuel bed, meaning the geometric display of fuel, must have an influence. Compare the influence of the porosity factor for wood crib fires with identical surface areas. This effect probably accounts for some of the scatter in Figure 3.
- Swedish full-scale tests with furniture has indicated that the fire process for this type of fuel may be ventilation controlled for markedly low values of $G_o/A_w h^{\frac{1}{2}}$ compared with the same value for wood crib fires.

Even a cursory analysis leads to the conclusion that the transition from ventilation control to fuel bed control means a fire process of less severity. This effect, of which a numerical confirmation will be given in the next section, should ideally be taken into account in a practical design procedure. However, the uncertainties pointed out earlier strongly suggest that this cannot be done with our present level of knowledge as a basis. As far as we can see, such a design process would have to be based on extensive full-scale calibration tests for each potential fire load exposure geometry.

BURNING RATES, VENTILATION RATES, AND FIRE SEVERITY

The choice of the ventilation controlled fire process as the basis for a structural design of fire exposed building components has been mentioned earlier. The time-temperature curves obtained in this design case are shown in Figure 4a for the opening factor $A_w h^{\frac{1}{2}}/A_t = 0.08\ m^{\frac{1}{2}}$ and different fire load densities.

The predominant terms in the heat balance equation are q_t and q_g. A possible discrepancy between the actual time-temperature curve and the design curve thus, in general, is caused by the maximum rate of burning, R_{max}, and rate of gas outflow, U_g, having values divergent from the nominal ones. R and U_g are given by the following expressions:

$$R_{max} = C_R \bullet 0.092\, A_w \bullet h^{\frac{1}{2}} \tag{6}$$

$$U_g = C_u \bullet \kappa\, A_w \bullet h^{\frac{1}{2}} \tag{7}$$

where κ is a factor that varies slightly with the rate of burning and the compartment temperature. As long as $T_g > 300°$ C, κ in the horizontal pressure difference model can be considered to be constant. In the design case, the two dimensionless factors C_R and C_u are equal to 1. In practice, the following deviations are conceivable.

(1) $C_R > 1$, $C_u = 1$. For fire compartments with small ventilation areas ($A_w \bullet h^{\frac{1}{2}}/A_t < 0.04\ m^{\frac{1}{2}}$) and a high value of fuel surface area, the rate of burning measured as weight loss per unit time can be considerably higher than the ventilation controlled value. Air inflow can be shown to be

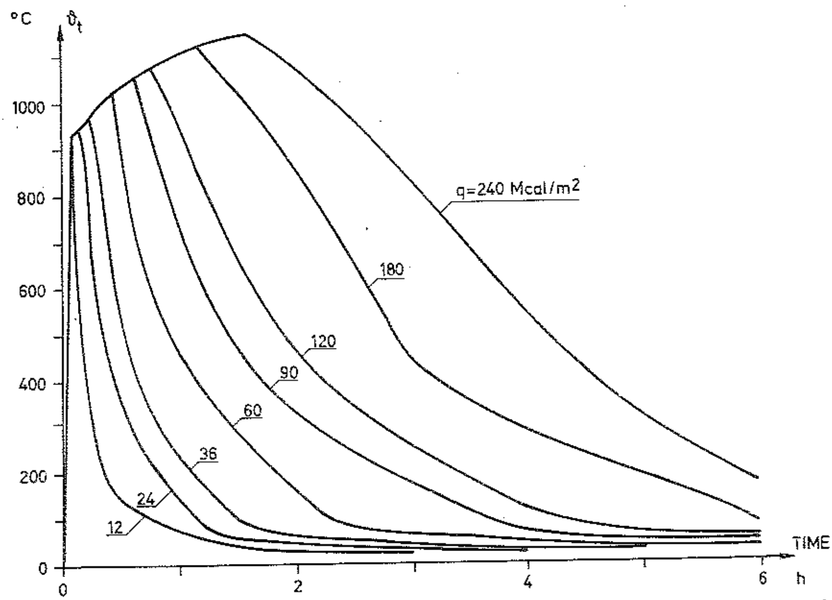

Figure 4a. Gas temperature-time curves of the complete process of fire $A_w \bullet h^{\frac{1}{2}}/A_t = 0.08\ m^{\frac{1}{2}}$. The bounding elements of the compartment are assumed to have $\lambda = 0.81\ W \bullet m^{-1} \bullet °C^{-1}$ and $C \bullet \lambda = 1.67 \bullet 10^6 J \bullet m^{-3} \bullet °C^{-1}$. Design case: $C_R = 1$, $C_u = 1$.

nearly independent of burning rate. This means insufficient air supply, incomplete combustion inside the compartment, and a lower effective heat value of the fuel.[2] The design procedure will give a fairly good approximation of the actual time-temperature curve and its impact on structural elements.

(2) $C_R<1$, $C_u = 1$. This corresponds to the fuel bed controlled case. An illustration will be given of what a change from ventilation to fuel bed control means to fire severity, in this case measured as the maximum steel temperature of an insulated steel column. Figure 4b gives the gas temperature-time curves for the same rate of air inflow and the same fire load densities as in Figure 4a with the value of R_{max} arbitrarily set to 50 percent of the ventilation controlled value — $C_R = 0.5$. The curves in Figures 4a and 4b have been used to compute the maximum steel temperature for a steel member with different degrees of insulation d_i/λ_i. The results are given in Figure 5a. It can be seen that, especially for uninsulated and lightly insulated structures, the assumption of ventilation control gives results considerably on the safe side.

(3) $C_R<1$, $C_u<1$, $C_R<C_u$. The example in Figure 5a is a rather extreme case. As discussed earlier in this paper and in Reference 2, the value of C_u will decrease for large openings. For $A_w \bullet h^{\frac{1}{2}}/A_t = 0.08\ m^{\frac{1}{2}}$, a value of $C_u = 0.8$ would be realistic. With this value of C_u constant throughout the fire process and with $C_R = 0.5$, the gas temperature-time curves are as shown in Figure 4c and the curves for maximum steel tem-

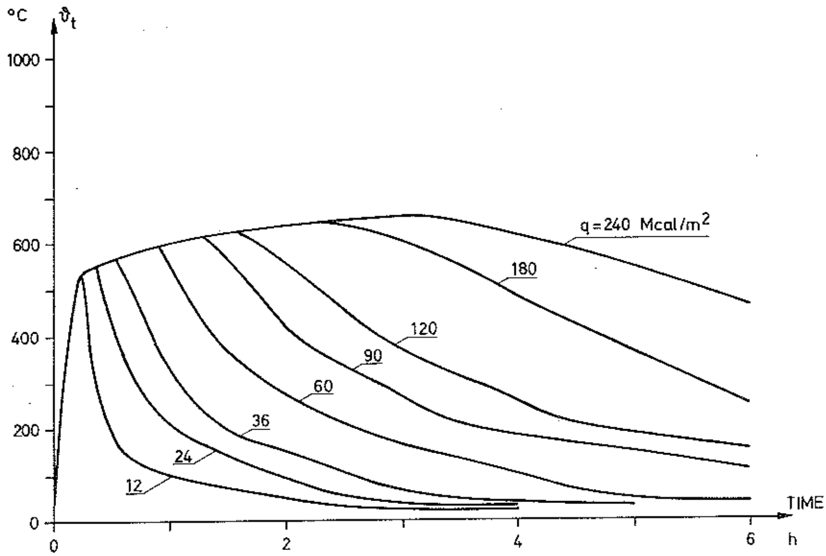

Figure 4b. Same as Figure 4a, but with $C_R = 0.5$ and $C_u = 1$.

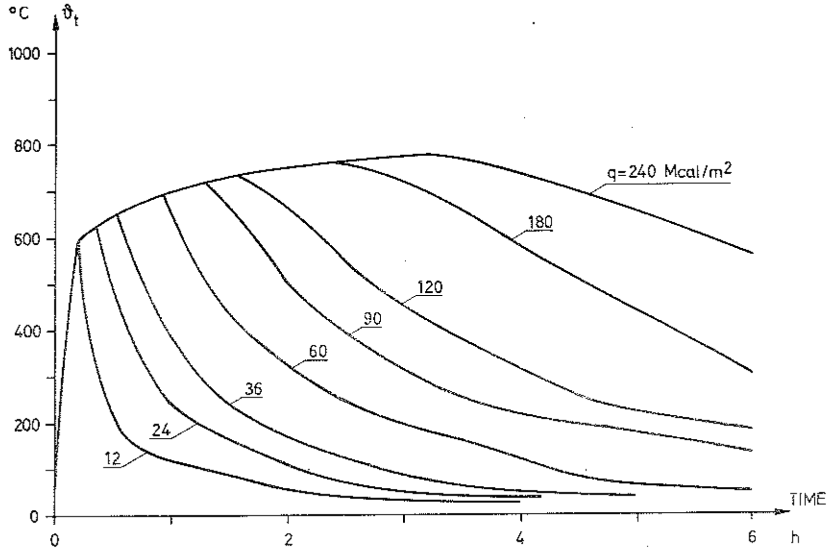

Figure 4c. Same as Figure 4a, but with C_R = 0.5 and C_v = 0.8.

perature are compared again with the design case in Figure 5b. The design curves still give results on the safe side, but for insulated structures, the difference is of minor importance.

(4) $C_R<1$, $C_v<1$, $C_R = C_v$. This is a combination that can give rise to a fire of slightly higher severity than the design case. It can happen when the rate of air inflow is reduced simultaneously as the fire is ventilation controlled. This case is identical with the design case for a smaller opening factor, which means a more severe fire. A measure of the increase in severity is given in Figure 6, which gives the resulting maximum steel temperature as a function of $A_w\bullet(h)^{1/2}/A_t$ for a fire-exposed steel member with A_i/V_s = 100 m^{-1} and insulated with 3 cm of mineral wool. The fire process is ventilation controlled with C_v = 1 (design case) and the fire load density equal to 210 MJ.m^{-2} (50 Mcal.m^{-2}) in all cases.

A simultaneous decrease of both air inflow and rate of burning with c:a 80 percent will at most increase the maximum steel temperature 50° C.

Harmathy[1] stressed the importance of designing the fire compartment in such a way that, whenever possible, the fire will be fuel bed controlled. In practice, this means as large ventilation openings as are feasible. As shown in Figure 6, this is wholly consistent with the differentiated Swedish approach to the fire engineering design problem in which the ventilation or "opening factor" value is the main parameter besides fire load density. Thus, the building code actively encourages the deliberate design of fire compartments advocated by Harmathy.

We have said before that the curve in Figure 6 represents an upper

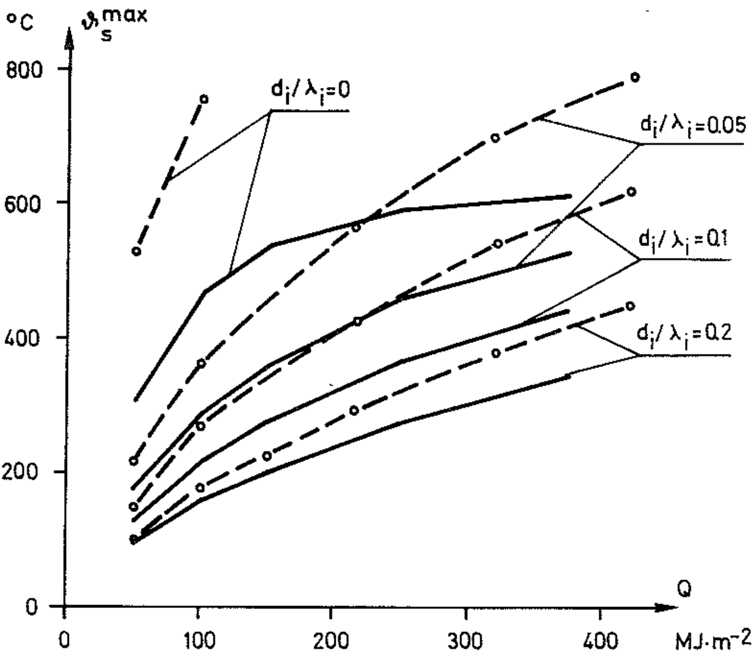

Figure 5a. Computed maximum steel temperature as a function of fire load density, Q_t for the degrees of insulation, d_i/λ_i = 0, 0.05. 0.1 and 0.2 $m^2 \bullet h \bullet {}^\circ C \bullet kcal^{-1}$ (0, 0.043, 0.086 and 0.172 $m^2 \bullet {}^\circ C \bullet W^{-1}$). Maximum steel temperature was obtained from the time-temperature curves in Figure 4a (dotted lines) and Figure 4b (solid lines).

limit. Expressed in a somewhat different way . . . the assumption of ventilation control can be seen as the introduction of an implicit safety factor in the design procedure. The importance of this extra load or safety factor should not be studied on the basis of deterministic concepts. Only a probabilistic analysis, taking into account the stochastic nature of relevant parameters, forms the proper, logical framework for such an evaluation. A first approach into this problem area has been taken.[10]

THE CONCEPT OF FIRE SEVERITY

The simulation technique presented in Reference 2 and 3 gives the time variation of the temperature fields in the constructions surrounding the fire compartment as an integrated part of the solution. Heat loss through the boundary structures is calculated by solving the partial differential equation for instationary heat flow, using temperature-dependent thermal parameters. A measure of the fire impact on floor, walls, and ceiling is thus immediately given. As the thermal response of load-bearing members, such as beams, columns, and frames of steel, as a rule considerably differs from that of the room-closing elements, a separate calculation must be made to determine the maximum steel temperature from exposure to fire.

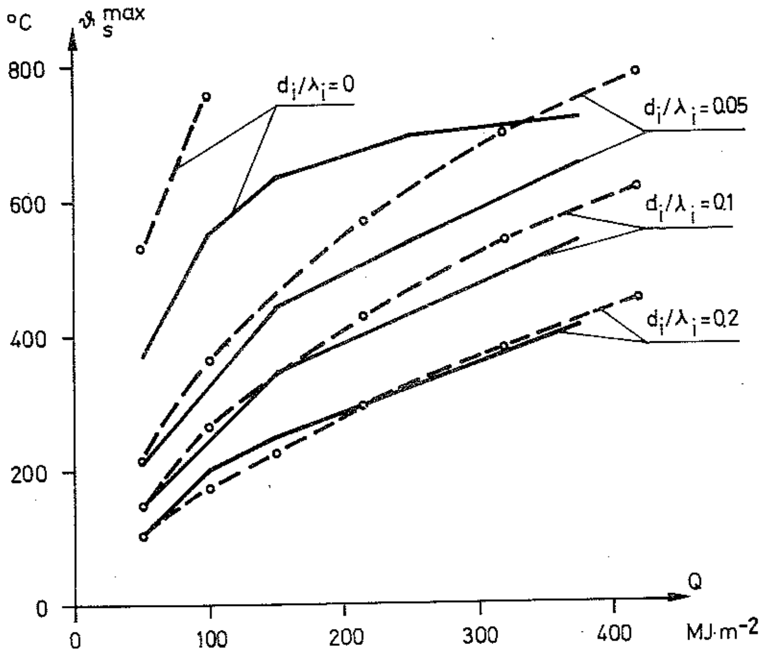

Figure 5b. Same as 5a except that maximum steel temperature was obtained in time-temperature curves in Figure 4a (dotted lines) and Figure 4c (solid lines).

The approximate method employed by Harmathy[1] results in three quantities characterizing the severity of each single fire — $\bar{q}_E$, τ, and T_g.

The importance of the new concept of fire severity introduced by these three parameters is stressed in Reference 1. An illustration of the relevance of $\bar{q}_E$, τ, and T_g to the structural fire design problem is, therefore, of interest. If we still restrict our analysis to the JFRO tests, data regarding heat flow into walls have been published for Test I only.[7] Figure 7 gives

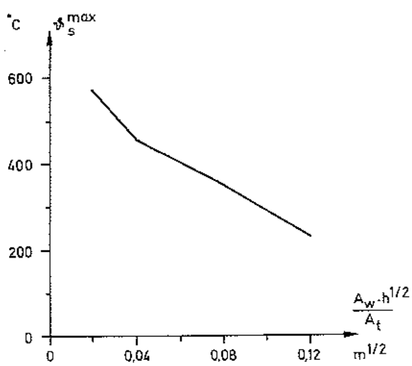

Figure 6. Computed maximum steel temperatures as a function of the opening factor, $A_w h^{1/2}/A_t$. Fire load density $Q = 210$ $MJ \cdot m^{-2}$ (50 Mcal m^{-2}) in all cases. Characteristics of the steel column $A_i/V = 150\ m^{-1}$; insulation was 3cm of mineral wool (Rockwool).

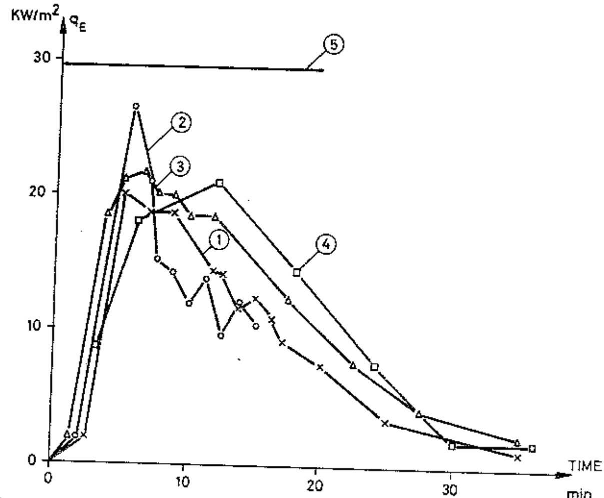

Figure 7. Heat flux into walls and ceiling for Test 1 in JFRO tests (Reference 7). 1 measured in rear wall. 2 measured in side wall. 3 measured in ceiling. 4 calculated in (2). 5 calculated in (1).

the time variation of the experimental heat flow into the walls as well as the corresponding curve obtained by the numerical simulations accounted for by the authors.[2] (Our curve is valid for Test M, but the two tests differ only in the geometrical arrangement of the wood cribs.) The agreement is good, indicating the reliability of the simulation technique. The value of $\bar{q}_E$ derived in Reference 1 for the same test is also shown. This latter quantity is only slightly larger than the measured maximum value, but as seen, the total amount of energy flowing into the bounding structures is grossly exaggerated by assuming this value during the entire fire duration.

The difference between Harmathy's $\bar{q}_E$ curve and ours illustrates the different degrees of accuracy obtained by the two approaches. The advantage with Harmathy's way of solving the heat balance equation by approximate methods is that the calculation can be made by hand. On the other hand, the amount of work needed to solve the parabolic non-steady state heat conductive equation in a correct way should not be exaggerated. It can be noted that most computer centers have one or more standard procedures for this problem in their program libraries. The inconvenience of designing a computer program must be weighed against the benefit of obtaining a complete picture of the temperature fields in the surrounding walls, floor, and ceiling during the entire fire process.

For the same Test I, the published experimental data comprises the time-temperature curve of a steel column exposed to fire. The dimensions of the column, which was insulated with 13 mm of mineral wool, are shown in Figure 8. Thermal conductivity is assumed to vary with temperature according to Figure 9, mainly in agreement with the same curve for the

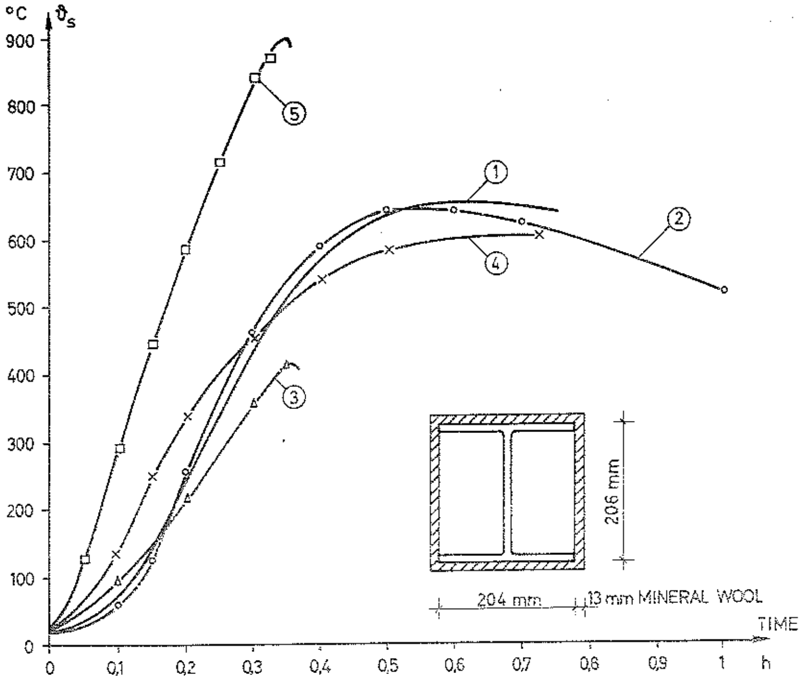

Figure 8. Average measured steel temperatures compared with steel temperature-time curves calculated according to four different methods for a protected steel column used in the JFRO Test 1. Weight of steel profile, 52 Kg•m⁻¹.

corresponding Swedish insulating material. In Figure 8, the following experimental and theoretical time curves for average steel temperature are given — (1) average measured values,[11] (2) computed with the experimental or simulated gas temperature-time curve as a basis, (3) computed with the $T_g - t$ given in Reference 1 as a basis, (4) computed with the measured $q_E - t$ curve (See Figure 7.) as a basis, and (5) computed with the q_E curve given by Harmathy[1] as a basis.

It is tempting to draw the conclusion from Figure 8 that a fire engineering design of uninsulated or insulated steel members, with the fire severity parameters $\bar{q}_E$, τ, and T_g defined by Harmathy[1] as a starting point, will produce results that are, in some cases at least, inconsistent and inadmissibly misleading. Use of the correct time-temperature curves for gas temperature or heat flow into surrounding structures will, on the other hand, produce results that are compatible and, as far as the thermal properties of the insulating material are known, reliable. Of course, the parameters $\bar{q}_E$, and T_g might be used for a general grading and categorizing of different fire compartment designs in order to obtain minimum fire damage for the enclosing structures. Their use in a differentiated structural design seems somewhat limited though.

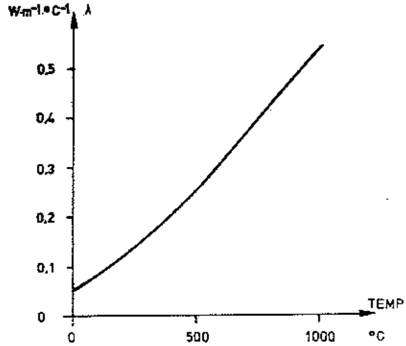

Figure 9. Relation between thermal conductivity and the temperature of mineral wool insulation used in calculations for Figure 8.

A further aspect of the T_g curves given in Reference 1 ought to be discussed. These curves omit the decay or cooling down period of the fire. There are numerous examples that a knowledge of the gas temperature during this period is essential for the determination of the maximum temperature occurring in a structure.[3] On page 347 of Reference 1, an analytical solution to the problem of maximum temperature in enclosing structures is presented, which presupposes constant heat flux across the boundary surface during a fully developed fire. As we have seen (Figure 7) this assumption is far from justified.

SUMMARY

This discussion was partly motivated by the two-part paper by T. Z. Harmathy, which appeared in the August and November 1972 issues of FIRE TECHNOLOGY.[1] To be specific, we wanted to supplement and give the proper basis for some of the material that appeared in his paper and which had been taken from one of our earlier papers. That done, we had one more incentive. In his excellent and well needed review of the theory and experimental practice of compartment fires, Harmathy also introduced a new approach to solving the heat balance equation. The essential novel feature is that some parts of the heat balance are approximated in such a way that a calculation by hand becomes possible. Obviously, the possibility of obtaining a solution without recourse to a computer is an advantage. However, the simplification necessary to achieve this objective means a corresponding inexactness in the determination of the critical parameters. This is demonstrated by explicit comparison between experimental results and the results obtained on one hand by Harmathy's method and on the other by a more realistic and accurate step by step numerical computer simulation of the complete fire process. The conclusions that can be drawn from this term by term checking reflects our strong belief that a more thorough understanding of the fire process and its impact on structural elements can only be achieved by a systematic use of a simulation technique into which all available knowledge is incorporated.

In Sweden, a theoretical approach to the design of fire exposed structures is permitted. Existing design diagrams presuppose a ventilation controlled natural fire process. Experimental evidence shows that, in many practical cases, the fire process will be regulated by other factors than the inflow of air into the compartment. Therefore, we discuss in some detail the effect of different combustion characteristics. We demonstrate that a design with fire load density and ventilation factor as fundamental parameters gives results that are on the safe side in almost all cases. Using this approach, the favorable influences of large ventilation openings on the load-bearing capacity of structural members is to an essential part taken into account.

NOMENCLATURE

A_f = exposed surface area of fuel, m^2
A_i = exposed area of steel insulation, $m^1.m^{-1}$
A_t = total bounding surface area, m^2
A_w = window area, m^2
C = specific heat, $J.kg^{-1}.°C^{-1}$
C_R = dimensionless coefficient defined in Equation 6
C_u = dimensionless coefficient defined in Equation 7
d_i = thickness of steel insulation, m
G_o = initial weight of fuel, kg
g = acceleration due to gravity, $m.s^{-2}$
H_f = effective heat value of the fuel, $J.kg^{-1}$
H = effective heat value during primary burning, $J.kg^{-1}$
h = window height, m
Q = initial energy content per unit bounding surface area, $MJ.m^{-2}$
q_E = rate of heat loss through bounding walls, floor, and ceiling, $J.m^{-2}.s^{-1}$
$\bar{q}_E$ = effective heat flux
q_g = rate of heat loss by convection in the openings, $J.m^{-2}.s^{-1}$
q_R = rate of heat loss by radiation through openings, $J.m^{-2}.s^{-1}$
q_t = rate of heat release by combustion inside the compartment, $J.m^{-2}.s^{-1}$
R = rate of weight loss. $kg.s^{-1}$
R_{max} = maximum rate of weight loss. $kg.s^{-1}$
$\bar{R}$ = rate of weight loss in the primary burning period, $kg.s^{-1}$
r = hydraulic radius = V_f/A_f, m
t = time
T_g = gas temperature, °C
U_g = rate of gas outflow, $kg.s^{-1}$
V_f = volume of fuel, m^3
V_s = volume of steel, $m^3.m^{-1}$
γ = bulk density, $kg.m^{-3}$
θ_s = steel temperature, °C

$\theta_{s_{max}}$ = maximum steel temperature, ° C
κ = factor defined in Equation 7,.$s^{-1}.m^{-5/2}$
λ = thermal conductivity, $W.m^{-1}.° C^{-1}$
λ_i = thermal conductivity of steel insulation, $W.m^{-1}.° C^{-1}$
ρ_a = density of air, $kg.m^{-3}$
τ = duration of primary burning period, s

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