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COMMENTS ON RATE OF GAS FLOW AND RATE OF BURNING FOR FIRES IN ENCLOSURES

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ϑ_g	Temperature of the combustions gases	^{o}C
φ	Coefficient in the equation $Q = \varphi \cdot A\sqrt{H}$	$kg \cdot s^{-1} \cdot m^{-5/2}$
ϕ	Porosity factor as defined by Gross	$cm^{1.1}$

PRINCIPAL NOTATIONS

A	Area of vertical openings in the enclosed space	m^2
A_t	Total bounding surface area of the enclosed space	m^2
$(A\cdot\sqrt{H})$	Air flow factor (Ventilation factor)	$m^{5/2}$
$(A\cdot\sqrt{H}/A_t)$	Opening factor	$m^{1/2}$
H	Height of vertical opening in the enclosed space	m
I_C	Heat energy released per unit time during combustion	$MJ\cdot s^{-1}$
I_L	Heat energy withdrawn per unit time from the enclosed space owing to the replacement of hot gases by cold air	$MJ\cdot s^{-1}$
I_R	Heat energy withdrawn per unit time from the enclosed space by radiation through openings in the enclosed space	$MJ\cdot s^{-1}$
I_W	Heat energy withdrawn per unit time from the enclosed space through wall, roof or ceiling, and floor structures	$MJ\cdot s^{-1}$
M	Quantity of combustible material	kg
Q	Rate of flow of the outgoing gases through a vertical opening in the enclosed space	$kg\cdot s^{-1}$
Q_a	Trial (assumed) value of Q	$kg\cdot s^{-1}$
R	Rate of combustion	kg of wood per unit time
W	Heat value of the fuel	$MJ\cdot kg^{-1}$
c_p	Specific heat of the combustion gases	$MJ\cdot kg^{-1}\cdot {}^oC^{-1}$
q	Fire load	$MJ\cdot m^{-2}$ of bounding surface area
r	Hydraulic radius	m
v_i	Rate of progress of charred layer of wooden sticks	$m\cdot s^{-1}$
ϑ	Temperature	^oC
ϑ_0	Temperature of the outside air	^oC

1. Introduction

In several countries a reassessment of existing methods of predicting the behaviour of fire-exposed structural members is at this time being made. This is a result of the intensified research carried out and in progress in the field of structural fire engineering. The reconsideration applies to the whole domain of structural fire engineering design, i.e. both to the process of fire development and to the fire behaviour of the structure.

A great many experimental fires, carefully executed, have demonstrated that a rational prediction of the fire resistance of a structural element only can be done by taking into consideration factors, hitherto not represented in the conventional design procedures of most countries. Fundamentally important in this design is the correct determination of the temperature-time curve of the fully-developed fire. The standard temperature-time curve used in most countries at best covers only a limited range of possible conditions regarding the detail characteristics of the fire load, the ventilation of the fire compartment and the thermal properties of the surrounding structures of this compartment. Moreover, usually no account is taken of the fact that the cooling down period of the fire must be included. Consequently, it is now being generally accepted that only the complete temperature-time curve of the actual fire can serve as a basis for an accurate analysis of the fire severity.

It has been suggested [1] that actual fires can be described by introducing a fictitious fire duration connected to the standard temperature-time curve. The fictitious fire duration is then determined by the condition that the maximum temperature in some reference structure shall be the same whether it is exposed to the actual temperatures or the standard curve temperatures. The advantage with such an approach is that results available from standard fire endurance tests can easily be utilized in design. However, Pettersson [2] has shown that the choice of reference structure notably affects the transition from an actual fire to an equivalent standard fire duration. The conclusion is that an appropriate fire engineering design must be based on realistic temperature-time curves.

In determining these curves, a systematic approach based on experiments is met with difficulties. The number of variables, some of them interacting, makes the number of necessary tests prohibitive. When in the early 1960's it was made clear that the solution of a simplified equation of energy or heat balance gives the variation of the gas temperature with the time, it meant that a new tool had been given the fire technologist. It is now possible to give a numerical simulation of the experimental fire process. Certain assumptions, concerning the different components in the energy balance, have to be made. An advantage with this theoretical analysis is that if the energy balance equation is solved and the solution compared with the measurements from the experimental fire an immediate check on the correctness of these assumptions is obtained. In this way the influence of each particular parameter can be studied and systematized.

These facts characterize the approach used by the authors in [3]. Some 30 full scale fires are analyzed. Based on results gained from these comparative calculations, temperature-time curves for the complete process of fire development are computed for a systematic variation of fire load, ventilation openings and type of structures bounding the fire compartment. As no definite assumptions could be made regarding fires controlled by other factors than the ventilation openings it was assumed throughout that the processes were ventilation-controlled. This was also justified by the fact that in almost every practical case the temperature-time curves computed in this way are on the safe side with regard to the corresponding fire behaviour and fire resistance of structures or structural elements.

The purpose of this report which is to be seen as a sequel or an appendix to [3] is mainly to discuss the behaviour of fires controlled by other factors than the ventilation of the compartment. Some pertinent questions such as how the individual terms in the energy balance equation are effected will be elucidated. This is most conveniently done with the help of comparative, theoretical, computer-aided calculations. We will further discuss under which conditions the fire process ceases to be ventilation controlled and to what accuracy this can be decided in advance. This will decide if the favourable effect of the fire not being controlled by ventilation can be used in a differentiated structural fire engineering design of load-bearing and separating structures.

2. Equation of heat balance

The equation, which describes the heat balance of a fire in an enclosed space can be written

$$I_C = I_L + I_W + I_R \qquad (2:1)$$

where

I_C = rate of heat release per unit time by combustion,

I_L = rate of heat loss per unit time by convection in the openings,

I_W = rate of heat loss per unit time though bounding walls, floor and ceiling,

I_R = rate of heat loss per unit time by radiation through the openings.

When expressions for all terms in the above equation are known it is possible to compute the temperature of the gases in the compartment.

Terms I_W and I_R

Two of these terms, I_W and I_R, present no particular problems. The instationary heat flow into surrounding structures is obtained by solving the general equation of non-stationary heat conduction in the one-dimensional case. If the relevant thermal properites are known the term I_W thus can be obtained with good accuracy. The radiation through the openings can be calculated from Stefan Boltzmann's law of radiation, assuming that the enclosed space behaves like a black body. This description is probably rough, but the term I_R usually is of relatively small importance. I_R is ordinarily less than 15 per cent of the sum on the right-hand side of Eq. (2:1) and amounts generally to 5-10 per cent.

Term I_L

The term I_L, which designates heat loss by convection through the openings of the compartment, generally has a great influence on the calculated temperatures when Eq. (2:1) is used.

I_L can be written

$$I_L = c_p \, (\vartheta_g - \vartheta_o) \cdot Q \qquad (2:2)$$

where

c_p = specific heat of gases,

ϑ_g = temperature of gases in the compartment,

ϑ_o = ambient temperature,

Q = rate of gas flow from compartment.

Kawagoe [4] has deduced the following expression for Q from theoretical considerations.

$$Q = \varphi \cdot A \cdot \sqrt{H} \qquad (2:3)$$

where

A = window area,

H = window height,

φ = constant.

The constant of proportionality φ is approximately independent of temperature above $300^{\circ}C$. φ varies slightly with the rate of burning R and is about $0.55\ kg \cdot s^{-1} \cdot m^{-5/2}$ when the rate of burning R is zero and $0.6\ kg \cdot s^{-1} \cdot m^{-5/2}$ when R reaches its maximum value. Eq. (2:3) with these values of φ was used by the authors in [3].

The deduction of Eq. (2:3) is based on the assumptions that the pressure inside the compartment varies linearly along the vertical axis and that the vertical accelerations of gases are negligible. Furthermore, the temperatures in the compartment are assumed to be uniform. These assumptions imply that the horizontal gas velocities in the opening can be determined from Bernoulli's theorem, i.e. the pressure difference at any level determines the magnitude of the gas velocity. By integrating this over the window opening the rate of gas flow can be obtained according to Eq. (2:3).

Thomas et al. [5] have pointed out that Eq. (2:3) is correct for small openings, but when the opening area for a given compartment is increased to a certain extent the above assumptions no longer satisfactorily describe the problem. As the opening becomes larger the vertical accelerations of gases have to be considered, which in turn means that the pressure differences and horizontal velocities decrease. As a result the total gas flow will be less than that given by Eq. (2:3). It is difficult, however, to estimate the limit beyond which Eq. (2:3) no longer is valid, since this probably depends on a lot of factors such as shape of windows and compartment.

Thus, we cannot exclude the possibility that the factor φ in the case of large openings is considerably less than the theoretical value given by Eq. (2:3). This means that the term I_L in the equation of heat balance as determined by the authors in [3] might be overestimated for compartments with large openings. We shall return to this question in connection with the analysis of fire tests in the next section.

Term I_C

The remaining term in the equation of heat balance to be discussed is the rate of heat release during combustion, I_C. This term is very difficult to estimate, since, in a complex manner, it depends on a lot of parameters. Such parameters are type, amount, thickness, porosity and distribution of the fuel or fire load, the geometry of the compartment and the shape and size of the window area.

The problem was tackled by the authors in [3] in the following way.

What could be assumed to be known for certain was that the total amount of heat energy which is released during the whole course of a fire must be equal to the fire load, expressed in terms of heat energy, i.e.

$$\int_0^\infty I_C \cdot dt = M \cdot W \qquad (2:4)$$

where M = total fire load in kg,

W = heat value of fuel in $MJ \cdot kg^{-1}$,

t = time coordinate.

In other words the area below the time graph of the rate of heat release I_C should be constant for a given fire load. The possibility of incomplete combustion and combustion occurring in flames outside the compartment was neglected, this being on the safe side.The question was how the given amount of energy should be distributed over the actual period of time.

To answer this question full scale fire tests reported in literature were analysed. For tests with sufficient data for a calculation with the equation of heat balance to be possible, a time graph of I_C was chosen on trial. On the basis of this a time-temperature curve was calculated and compared to the measured temperatures. If needed, the time graph of I_C was changed and used for a new calculation. This was repeated until the calculated and measured time-temperature curves were in agreement. Such comparative calculations were performed for some 30 full scale tests and on the basis of the time graphs of I_C so obtained a generalisation was made.

As a result polygonal curves of the type shown in fig. 1b were obtained; one for each combination of opening factor $A \cdot \sqrt{H}/A_t$ and fire load $q = M \cdot W/A_t \cdot A_t$ = total bounding surface area of enclosed space. The level of the horizontal (maximum) branch of the curves was assumed to be proportional to $A \cdot \sqrt{H}$, i.e. the maximum rate of burning was determined by the rate of air supply.

$$R_{max} = 5.5 \cdot A\sqrt{H} \ kg \cdot min^{-1} \qquad (2:5)$$

The comparative calculations showed that the effective heat release corresponding to this weight loss should be about 10-12 MJ/kg of wood (2500 - 2800 kcal/kg). A value suggested by Kawagoe [6] and also included in the Swedish Building Regulations 1967 was chosen, viz. 10.8 MJ/kg (2575 kcal/kg). It should be pointed out that this value is only valid during the period of maximum burning intensity i.e. during the flame phase. During the cooling down period, for instance, the heat release corresponding to 1 kg of weight loss generally is larger than 2575 kcal.

The description of I_C outlined above, is of course simplified compared to the actual behaviour. However, since the area below the I_C-time curve is correct, possible errors should concern the distribution of energy only. When used to calculate a temperature-time curve, which in turn is used to determine the temperature of a structural element, the final result should be very little affected by such an error. The approach has the great advantage that the complete process of fire development including the cooling down period is described by a realistic temperature-time curve.

Nevertheless, from the investigation of full scale tests mentioned above it appeared that far from all test fires followed eq. (2:5). Furthermore, after this investigation was performed results from additional full scale tests have become available to the authors. In many of these tests the rate of burning was not controlled by ventilation but by the design of the fuel bed and consequently differred from the values used in [3]. To investigate the influence of this discrepancy, comparative calculations were performed for some ofthe new tests. The results will be described in the next section and in connection with that the rate of burning and rate of gas flow will be discussed in some detail.

3. Comparative calculations for an analysis of some fire tests

Comparative calculations have been performed for the following fire test series, cf. table 1.

a) 10 out of a series of full scale tests performed at the Fire Research Station, Boreham Wood, Herts, England. The tests have been described and analysed in a series of publications [7, 8, 9].

b) 5 out of a series of full scale tests performed by Ehm and Arnault in Metz, France [10].

c) 6 out of a series of model scale tests performed by Leif Nilsson at Lund Institute of Technology, Sweden [11]. In these tests a cubical compartment (side 1 m) was used.

TABLE 1

TEST NO.	$\frac{A\sqrt{H}}{A_t}$ $m^{1/2}$	FIRE LOAD		R_{80-30} measured value $kg\cdot s^{-1}$	$R = 0.092A\sqrt{H}$ $kg\cdot s^{-1}$	$\frac{R_{80-30}}{0.092A\sqrt{H}}$ (4):(5)	$\frac{Q_a}{\phi\cdot A\sqrt{H}}$ used in calculations	$I_{C,av}$ used in calculations $MJ\cdot s^{-1}$	W_{av} $MJ\cdot kg^{-1}$ (8):(4)	Performed at
		$kg\cdot m^{-2}$ floor area	$MJ\cdot m^{-2}$ of bounding surface area							
(1)	(2)	(3)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	
V (1/8)	0.03	60	250	0.53	0.35	1.51	1.0	5.4	10.2	FIRE RESEARCH STATION (Floor area 29m²)
O (1/4)	0.06	7.5	31	0.20	0.69	0.29	1.0 0.8	- 2.4	- 12.0	
G (1/4)	0.06	15	62	0.33	0.69	0.49	1.0 0.8	- 4.4	- 13.5	
M (1/4)	0.06	30	125	0.63	0.69	0.93	1.0	8.0	12.5	
U (1/4)	0.06	60	250	0.77	0.69	1.13	1.0	9.3	12.0	
P (1/2)	0.12	7.5	31	0.20	1.38	0.14	1.0 0.8	2.7 2.3	13.5 11.6	
D (1/4)	0.12	15	62	0.37	1.38	0.27	1.0 0.8	- 4.5	- 12.1	
N (1/2)	0.12	30	125	0.63	1.38	0.46	1.0 0.8	8.0 6.5	12.7 10.3	
R (1/2)	0.12	30	125	0.73	1.38	0.53	0.8	8.0	11.0	
L (1/2)	0.12	60	250	1.07	1.38	0.77	1.0 0.8	- 12.6	- 11.9	
III 30/10%	0.015	30	90	0.170	0.095	1.79	1.0	1.54	9.0	METZ (Floor area 12.4 m²)
I 15/25%	0.055	15	45	0.141	0.35	0.40	1.0 0.9 0.8	- 1.68 1.57	- 11.9 11.2	
II 30/25%	0.055	30	90	0.28	0.35	0.80	1.0	3.52	12.6	
V 60/25%	0.055	60	180	0.34	0.35	0.97	1.0	4.18	12.3	
IV 30/40%	0.091	30	90	0.29	0.57	0.51	1.0 0.8	- 3.73	- 12.8	
47	0.02	13	38	0.015	0.011	1.36	1.0	0.154	10.3	LUND INSTITUTE OF TECHNOLOGY (floor area = 1 m²)
43	0.04	12.8	37.5	0.026	0.022	1.18	1.0	0.284	10.9	
32	0.07	12.4	36.2	0.031	0.038	0.82	1.0 0.8 0.6	- 0.378 0.328	- 12.2 10.6	
37	0.07	12.5	36.5	0.020	0.038	0.53	1.0 0.8 0.6	- - 0.251	- - 12.5	
9	0.114	12.8	37.5	0.033	0.063	0.52	0.8 0.6	- 0.405	- 12.3	
5	0.114	12.1	35.5	0.020	0.063	0.32	0.8 0.6	- 0.194	- 9.7	

Theoretical calculations with the equation of heat balance (2:1) were made accordning to the trial and error method described in the previous section. As examples, the calculated time-temperature curves for three of the above mentioned tests - one from each series - are shown as full-line curves in figures 1a, 2a and 3a compared to the measured temperatures (dotted curves). The corresponding time graphs of the rate of heat release I_C are shown as curve 1 in figures 1b, 2b, 3b. In these

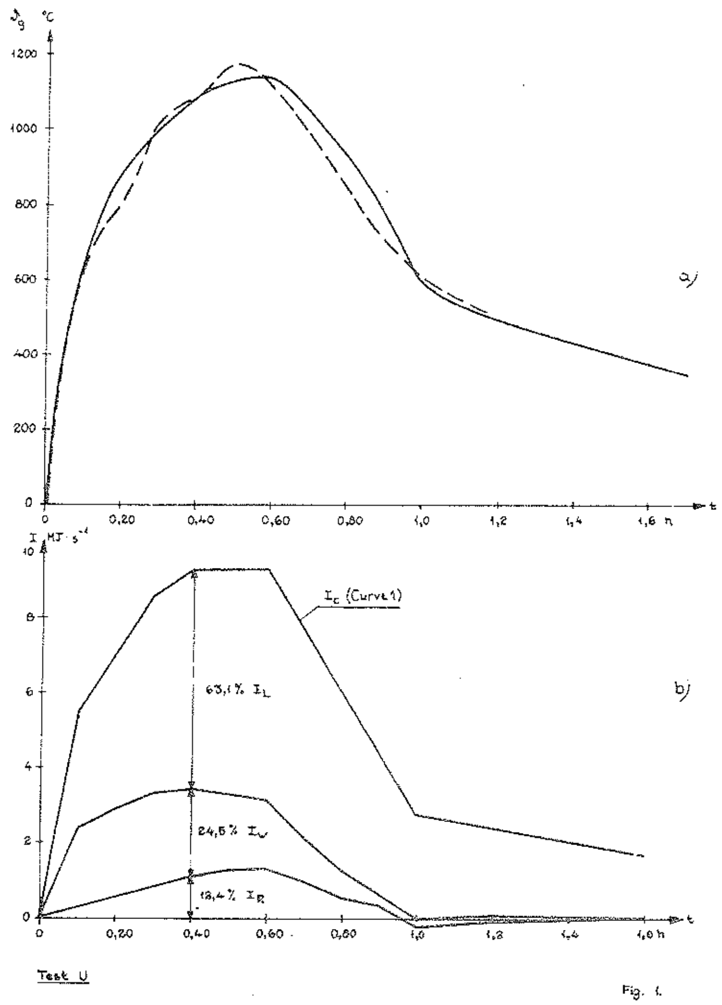


Fig. 1

Gastemperature-time curve for test U of the JFRO-series. The opening factor $A\sqrt{H}/A_t = 0.06\ m^{1/2}$ and the fire load $q = 250\ MJ \cdot m^{-2}$ of bounding surface area. Measured (dash-line curve) and computed (full-line curve) temperatures. Fig. b gives the time variation of terms I_C, I_L, I_W and I_R.

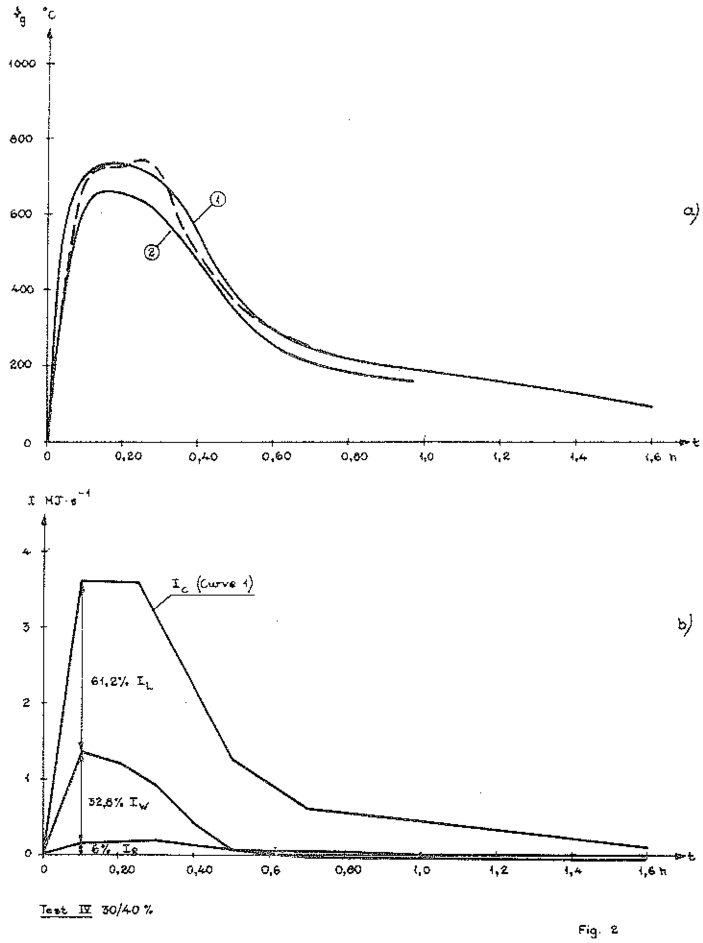


Fig. 2

Gastemperature-time curves for test IV, 30/40% in the Metz-series. The opening factor $A\sqrt{H}/A_t = 0.091\ m^{1/2}$ and the fire load $q = 90\ MJ \cdot m^{-2}$ of bounding surface area. Dash-line curve = measured temperature. Full-line curve = computed temperatures. Curve 1 is computed with $Q = 0.8 \cdot \rho \cdot A\sqrt{H}\ kg \cdot s^{-1}$, curve 2 with $Q = \rho \cdot A\sqrt{H}\ kg \cdot s^{-1}$. Fig. b gives the time variation of terms I_C, I_L, I_W and I_R.

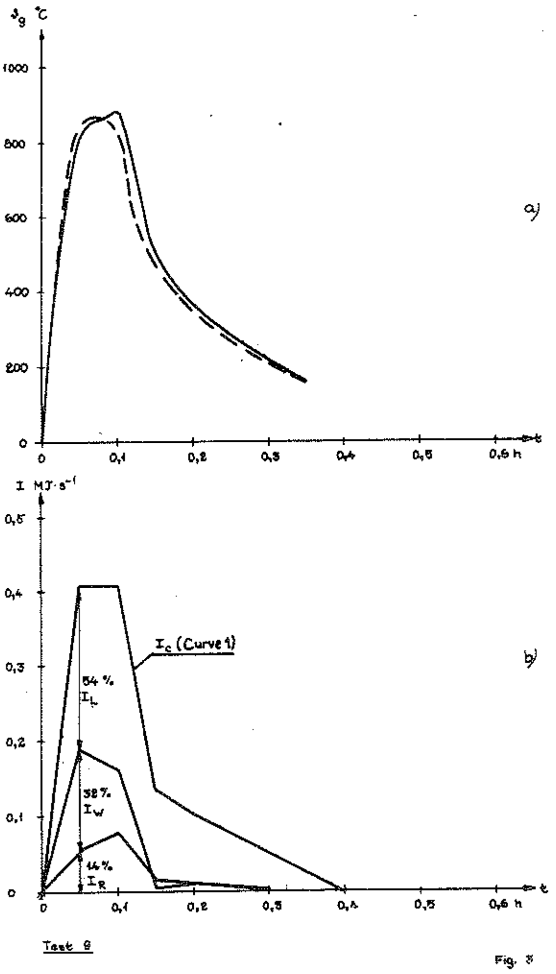


Fig. 3

Gastemperature-time curve for test 9 of the Nilsson-series. The opening factor $A\sqrt{H}/A_t = 0.114\ m^{1/2}$ and the fire load $q = 37.5\ MJ \cdot m^{-2}$ of bounding surface area. Dash-line curve = measured temperature. Full-line curve = computed temperature. Fig. b gives the time variation of terms I_C, I_L, I_W and I_R.

figures the proportions between the terms I_L, I_W and I_R are illustrated. It is seen that the predominant term is I_L, i.e. the heat lost by gas flow through openings.

The results of all the comparative calculations have been summarized in table 1. The meaning of the separate columns of this table will be explained in what follows.

Column 1: Test identification symbol; the symbols agree with symbols used in sources.
Column 2: Opening factor $A\sqrt{H}/A_t$ in $m^{1/2}$, where A = total window area, H = window height, A_t = bounding surface area of compartment.
Column 3: Fire load expressed in kg per m^2 floor area as well as in MJ (= 10^6J) per m^2 of bounding surface area (1MJ = $0.24 \cdot 10^3$ kcal).
Column 4: Rate of weight loss R_{80-30} measured in tests. Index 80-30 means an average over that period of time during which the amount of fuel decreases from 80% to 30% of initial amount.
Column 5: Rate of weight loss determined from Kawagoe's formula $R = 5.5\ A\sqrt{H}\ kg \cdot min^{-1}$.

Column 6: The ratio between the measured value R_{80-30} and the theoretical value 5.5 $A\sqrt{H}$. When this ratio is about unity the rate of burning is said to be ventilation controlled.

Column 7: Ratio between rate of gas flow Q_a assumed in calculations and the value given by Eq. (2:3) $Q = \varphi \cdot A\sqrt{H}$.

Column 8: $I_{C,av}$ is the value of I_C used in calculations, averaged over the same period of time as the measured values of R_{80-30} in column 4. Each value in this column corresponds to a state with the calculated time-temperature curve in agreement with the measured one. In some cases it was impossible to obtain agreement, because the available amount of energy was not enough to rise the I_C-time curve to the required level. Such cases have been marked with a stroke in columns 8 and 9.

Column 9: W_{av} is the ratio between $I_{C,av}$ and R_{80-30} and thus constitutes the effective heat corresponding to unit weight loss during the 80-30 period. The effective heat value so obtained does not include those parts of the released heat which are fed back to unburnt fuel or released in flames outside the compartment.

With the results of table 1 as a basis we will now proceed to discuss the rate of gas flow and the rate of burning.

Rate of gas flow out of openings

When describing the term I_L in section 2 a brief outline was made of the considerations in [5] concerning the rate of gas flow into and out of a compartment in fire. From the discussion made by Thomas et al. it follows that Eq. (2:3)

$$Q = \varphi \; A\sqrt{H}$$

will overestimate the rate of gas flow out of the compartment when the factor $A\sqrt{H}$ becomes large. It was estimated in [5] that the total air inflow would be reduced to about 1/4 of the inflow given by the condition that there is no vertical acceleration. A similar reduction can be expected for the outgoing flows in the two cases. The compartment considered was a cubical one with one side open. This is an extreme case, and the estimate had to be based on some rather uncertain assumptions. The tendency, however, of the air inflow per unit window area to grow smaller as the window area grows larger was in this investigation confirmed in two ways, firstly by the results of the comparative theoretical analysis summarized in table 1, secondly by an approximation performed on the results by Nilsson [11].

To take the results in table 1 first it can be seen that in some cases calculations were made for alternative values of the assumed gas outflow Q_a. Those values listed in column 7 for the same test represent upper and lower values of the gas outflow and they have been achieved in the following way: For each full scale test the process of fire development was numerically simulated under these conditions as a first approximation:

$$Q_a = \varphi A\sqrt{H}$$

$$M W = \int I_C \, dt$$

Under these stipulations, computations were made with the released effect I_C distributed in different ways during the course of the fire development until agreement between the experimental and the theoretical temperature-time curve for the combustion gases was obtained. For several tests, however, it was not possible to obtain agreement between calculated and measured temperature-time curves as long as $Q_a/\varphi A\sqrt{H}$ was unity. The amount of energy given from the beginning was not large enough, or, in other words, the rate of gas flow Q_a had to be reduced. In fig. 2a the influence of a change in the rate of gas flow is illustrated. The dotted line shows the temperature-time curve measured in test no. IV (30/40%) of the Metz series, and the influence of a change of the rate of the gas flow is illustrated by the two continuous lines. Curve no. 2 shows the temperatures obtained when the rate of gas flow is assumed to be $Q_a = \varphi A\sqrt{H}$ and curve no. 1 shows the temperature-time curve when $Q_a = 0.8 \cdot \varphi A\sqrt{H}$, all other factors not being changed.

The method of distributing the rate of energy release I_C in the time so that experimental and theoretical temperature-time curves coincide is not an exact one. An exact coincidence for the complete process of fire development is hard to obtain. Therefore, for some tests where $Q_a = \varphi A\sqrt{H}$ and a certain effect distribution had given satisfactory results a new computation could be done with $Q_a = c \cdot \varphi A\sqrt{H}$; $c < 1$. In order to get the new temperature-time curve to agree with the experimental curve a redistribution of the term I_C was then necessary. A moderate change of the maximum level of energy release during the flame phase is balanced by small changes during the cooling down period so that the condition $M \cdot W = \int I_C \, dt$ is still fulfilled. Such a redistribution was only possible for certain values of c. It is the lowest of these values that are printed in column 7 on the second line for some of the tests in table 1. Column 7 thus for some of the tests gives upper and lower bounds for the total outflow of gases through the openings. For $Q_a = \varphi A\sqrt{H}$ the time integral of the temperature-time curve as a rule is somewhat less than the same integral for the experimental curve, for $Q_a = c\,\varphi A\sqrt{H}$, $c < 1$, somewhat greater.

Different distributions of the term I_C give different effective heat values of the fuel, cf. column 9. If in table 1 W_{av} is the same for two different tests, take for example tests O and U in the Fire Research Station series, the rate of gas flow given in column 7 can be compared directly for these tests. In the case of tests O and U the rate of gas flow seems to be less in test O than in test U, provided that the real heat value is about the same in both cases.

The variation of the rate of gas flow can also be estimated from measurements of temperature and rate of burning in the following manner: Approximately, the terms of the equation of the heat balance can be written

$$I_C = k_1 \cdot R$$

$$I_L = k_2 \cdot Q \cdot \vartheta_g$$

$$I_W = k_3 \cdot \vartheta_g$$

$$I_R = k_4 \cdot \vartheta_g$$

where k_1 to k_4 are assumed to be constants. Substituting these expressions in Eq. (2:1), we find the relation

$$Q = \frac{k_1 \cdot R}{k_2 \cdot \vartheta_g} - \frac{(k_3 + k_4)}{k_2} = C_1 \frac{R}{\vartheta_g} - C_2$$

saying that Q is approximately proportional to the ratio R/ϑ_g. The approximations are considerable, especially for the terms I_W and I_R, but since I_L and I_C are the predominant terms the above relation makes some sense. Thus if we plot the ratio $R_{80-30}/\vartheta_{80-30}$, obtained from the model test series made by Nilsson [11], against $A\sqrt{H}/A_t$ we get the values shown by crosses in Fig. 4. Each point in the diagram is the mean of 10 tests with varying porosity factors for the fuel. If $Q = \varphi A\sqrt{H}$ the curve will be a straight line. It is seen that for the higher values of $A\sqrt{H}/A_t$, $A\sqrt{H}/A_t = 0.07$ and $0.114\ m^{1/2}$, $Q/\varphi A\sqrt{H} < 1$. This is consistent with the results obtained in table 1.

From these results can furthermore the following summing up regarding the rate of gas flow out of the compartment be made:

1. The reduction of the rate of gas flow Q compared to the formula $Q = \varphi A\sqrt{H}$ seems to occur at high values of $A\sqrt{H}/A_t$, $A\sqrt{H}/A_t > 0.06\ m^{1/2}$.

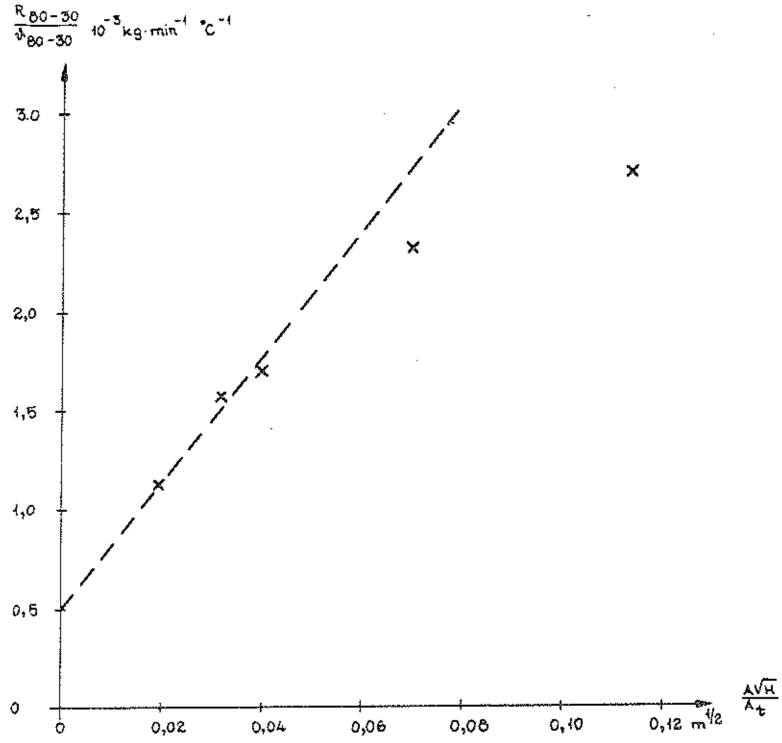


Fig. 4

Relation between R_{80-30} and the opening factor $A\sqrt{H}/A_t$ for tests in the Nilsson-series. Each point in the diagram is the mean of 10 tests with varying porosity factors. Model scale fires in a cubical compartment 1 m x 1 m x 1 m, with pile of 2.5 cm wooden sticks.

2. Even for large window openings the reduction seems to be not more than a factor 0.7 - 0.8.

3. The reduction seems to set in at about the same values of $A\sqrt{H}/A_t$ for all the three test series in spite of the fact that different scales and shapes are represented.

4. The reduction does not occur in cases where the burning is ventilation controlled, i.e. when the amount of fuel is large.

5. The ratios between R_{80-30} and $R = 5.5\ A\sqrt{H}$ is always less than the $Q_a/\varphi\ A\sqrt{H}$-ratios in the fuel bed controlled regime. Cf. columns 6 and 7 in table 1. This means that the reduction in Q has no influence on the rate of burning in this regime.

Rate of burning

Kawagoe [4] assumed that the rate of burning was limited by the available air supply, i.e. that the maximum rate of burning R was proportional to $A\sqrt{H}$ (cf. Eq. 2:3), which yields the well-known formula

$R = 5.5\ A\sqrt{H}\ \text{kg}\cdot\text{min}^{-1}$ (2:5).

Many fire tests in model scale as well as full scale have shown results in agreement with this formula. However studies, mainly at the Fire Research Station, have shown that this formula for R cannot be generally applied without reservations [5]. Table 1 shows, that in some cases the rate of burning is divergent from the value given by Eq. (2:5). These cases mostly apply to two different kinds of fire compartment, the first kind having comparatively large windows and small amounts of fuel. The second kind is characterized by the opening area being smaller than the average value.

We study the compartment with the large openings first. It has been established that when for a given compartment and fuel load the ventilation area becomes larger the rate of weight loss reaches an upper limit and does not increase if the opening is made still larger. While for smaller openings the rate of burning as a rule is not very much affected by factors like the amount of fuel, the thickness and the porosity of the fuel, i.e. by the fuel bed design, these factors become highly important when the opening area has passed this transition point. The two different regimes have been termed "ventilation controlled" and "fuel bed controlled". In the latter each particular design of the fuel bed has an upper limit of the rate of burning, irrespective of the ventilation area. This upper limit is not necessarily identical with the rate of weight loss of the same fuel bed burning in the open because of the termal input to the fuel from the surrounding constructions.

If wood cribs are used as fuel the main parameters determining the rate of burning when the openings are large are the amount of fuel, the thickness of the sticks in the crib, the spacing between the sticks and the distribution of fuel in the compartment. Fig. 5 shows the result of some 20 full scale tests at the Fire Research Station. The tests were so designed that most of them were controlled by the fuel bed. For all fire loads, except the highest, an increase in the ventilation area did not give a corresponding increase in the rate of burning. The decisive factor was the amount of fuel, the rate of burning almost linearly increasing with the fire load. The thickness of the sticks (4.5 x 4.5 cm) and the spacing of the cribs were the same throughout the tests. The distribution of the fuel in the compartment was varied but was not found to have any significant influence.

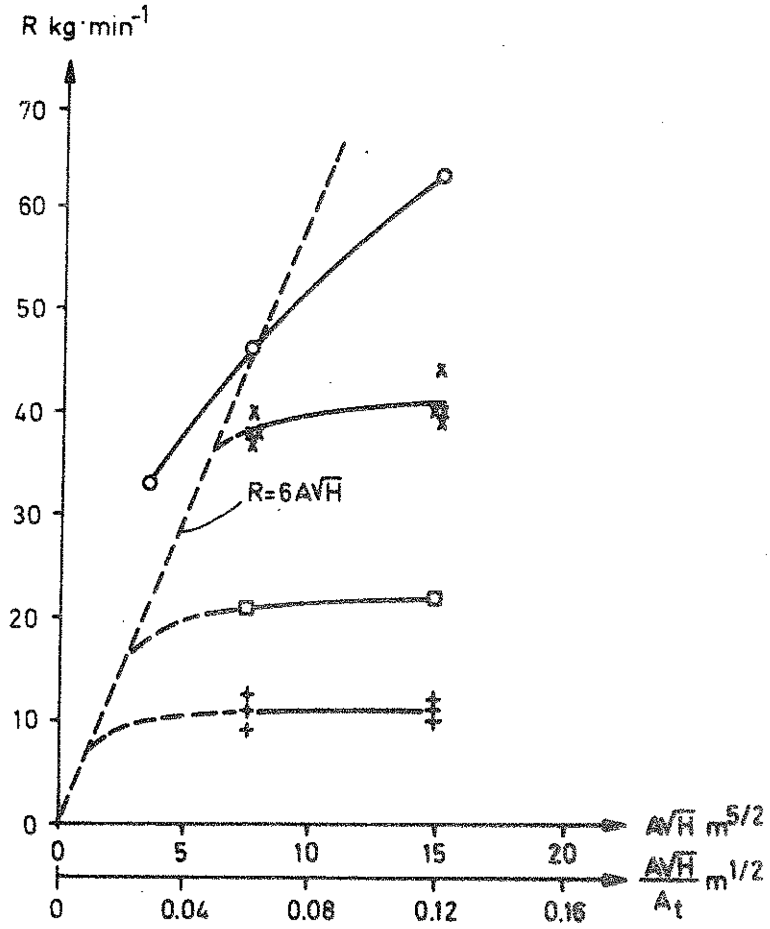


Fig. 5

Relation between R_{80-30} and air flow factor $A\sqrt{H}$ or opening factor $A\sqrt{H}/A_t$ at various fire loads. Full-scale tests at JFRO [6]. Fire load densities are referred to floor area of compartment.

Symbol	Fire load density	
	kg/m²	lb/ft²
o	60	12·4
x	30	6·2
□	15	3·1
+	7.5	1·55

Similar tests have been performed in Metz [10] by Ehm and Arnault giving on the whole the same dependence on the rate of burning of the amount of fire load. Another thickness of the sticks (7.5 x 4.5 cm) and another crib design were used.

Nilsson [11] has recently published the results of a fire test series in model scale. The main parameters studied were the air flow factor and the porosity of the fuel. Fig. 6 shows the rate of burning plotted against the porosity factor ϕ as defined by Gross [12] for various values of the opening factor $A\sqrt{H}/A_t$. The fire load is held constant 2 $kg \cdot m^{-2}$ of bounding surface area and the sticks have a thickness of 2.5 cm. As seen from the figure the porosity hardly affects the rate of burning at the lower values of $A\sqrt{H}$. For the largest values of $A\sqrt{H}$ however the rate of burning strongly varies with ϕ.

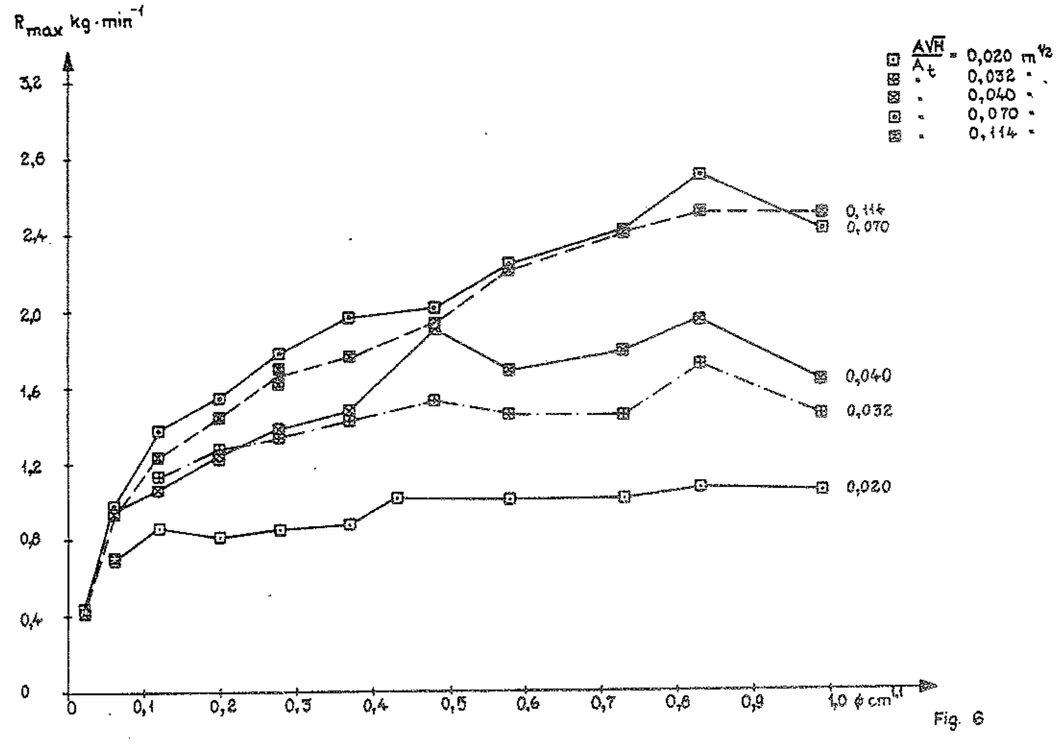


Fig. 6 Relation between maximum rate of burning R_{max} and the porosity factor ϕ as defined by Gross at varying values of the opening factor $A\sqrt{H}/A_t$. Model scale tests, cubical compartment 1 m x 1 m x 1 m, fire load consisting of 2.5 cm wooden sticks piled in a crib. Fire loads ≈ 37 MJ·m^{-2} of bounding surface area for each test.

In a simplified manner the behaviour in the fuel bed controlled regime can be explained by the fact that the exposed surface area of the fuel determines the production of combustible volatiles (pyrolysis) and hence the rate of burning. The surface area increases with increasing fire load or decreasing stick thickness for the same fuel load. A change in the porosity of the fuel causes a change in the flow pattern of volatiles and air inside the crib and in this way influences the rate of burning. But it seems that the more decisive factor is the amount of fuel.

The value of $R = 5.5\ A\sqrt{H}$ follows from the condition that the combustion or weight loss of 1 kg of wood always requires the same amount of inflowing air, irrespective of the amount, porosity and particle shape of the fuel. There is, however, experimental evidence that does not support the assumption that the rate of weight loss is limited by the available air supply. Thus Law [5] has shown that if a piece of wood is exposed to radiation in an inert atmosphere such as nitrogen, it will decompose at almost the same rate as in the open air. Consequently, the radiation-exposed fuel in a fire compartment ought to approximately decompose in proportion to the increasing surface area. The fact that this is not so and that the burning rate in a majority of cases stays at the level $R = 5.5\ A\sqrt{H}\ kg \cdot min^{-1}$ is then explained in this way [5]: The larger fuel surface area implies a larger production of volatiles and hence a larger loss of weight. The air supply is independent of the rate of burning and supposed to be rather small. Therefore the oxygen required for the combustion is not present inside the compartment. The result will be incomplete combustion inside the compartment (formation of hydrocarbons) or burning outside the compartment. This means that the heat produced will not correspond to the production of volatiles. The excess of volatiles will reduce the air inflow and extract heat from the combustion. A negative feedback to the fuel is thus produced and the production of volatiles decreases. A point of equilibrium for the burning rate R is reached. $R = K\ A\sqrt{H}$. We can conclude that the constant of proportionality k must have some correlation to the ratio (amount of fuel)/(amount of inflowing air). But experiments in different countries and in different scales have shown that for ventilation controlled fires k will be in the range 5 - 6 for the values of $M/A\sqrt{H}$ of most general occurence [15].

For fire compartments with rather small ventilation area and a comparatively high fire load, i.e. for larger values of $M/A\sqrt{H}$, it is clear from table 1 that k can attain the double value. This higher rate of burning in $kg \cdot min^{-1}$ will partly when it comes to the temperature be eliminated by the fact that the negative feedback reduces the effective calorific value of the fuel. See for example test nr III 30/10% in the Metz series in table 1.

We will now proceed to discuss, what the results from the tests in table 1 and some other tests analysed in [3] can tell us about the rate of burning in the fuel bed controlled domain.

The rate of burning or the energy release will be stated in $kg \cdot min^{-1}$. It would have been more appropriate to state the energy release in $joule \cdot s^{-1}$ but this would have made comparisons with measured values more difficult as it is clear that the ratio energy release/kg wood is not a constant.

We are here concerned with the questions: Given characteristic data for the fuel and the fire compartment, can it be determined in advance if the fire process is ventilation controlled or not? And if fuel bed controlled, what will the rate of burning be? Fig. 7 gives the rate of heat release, expressed as $R/A\sqrt{H}$, as functions of the opening factor $A\sqrt{H}/A_t$. Only full scale tests are considered. The R used is R_{80-30} obtained by measurement. We could also have used $I_{C,\ 80-30}$ in the table 1, column 7, divided by the effective heat value. This is done for the experiments performed by Ödeen [13], since in these tests there was no weighing of the fuel. For further details, see [3]. The tests performed at Studsvik and some of the test in Japan are analyzed in [3]. The tests at Metz with concentrated fuel loads and most of the tests from Japan are included without any comparative theoretical analysis done. The measure of R_{80-30} from a curve that told how the fire load in kg varies with the time sometimes gave a rather uncertain value, especially for the tests from Japan.

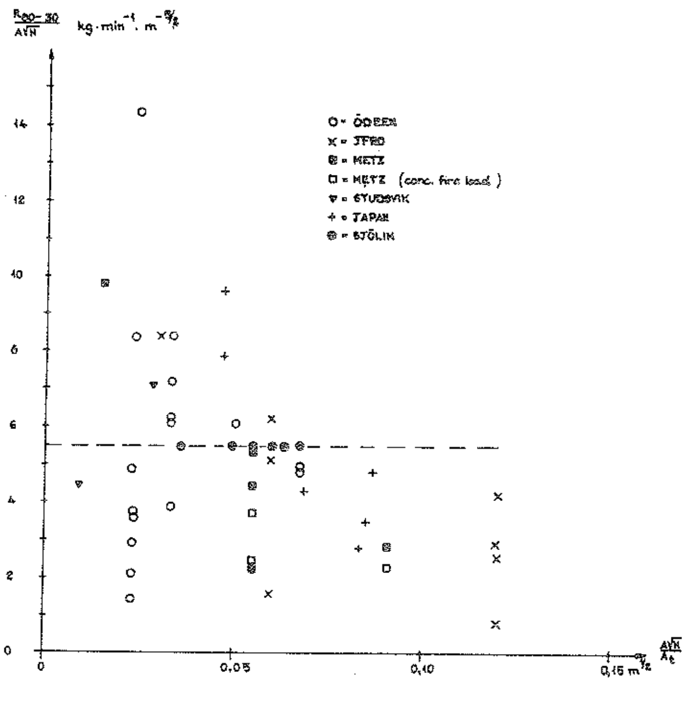


Fig. 7

Fig. 7

Relation between $R_{80-30}/A\sqrt{H}$ and the opening factor $A\sqrt{H}/A_t$ for full-scale fire tests performed at various research stations.

There is a wide scatter in Fig. 7, especially for lower values of $A\sqrt{H}/A_t$. This scatter is mostly caused by tests marked by O. In these tests the ventilation was forced by use of fans. This means that the negative feed-back mechanism associated with ventilation control was partly suppressed. Excluding Ödeen's experiments, there is a tendency in fig. 7 that $R/A\sqrt{H}$ is generally decreasing with increasing values of $A\sqrt{H}/A_t$. For $A\sqrt{H}/A_t$ = 0.03 - 0.07 $m^{1/2}$ the values of $R/A\sqrt{H}$ is near the theoretical value 5 - 6.

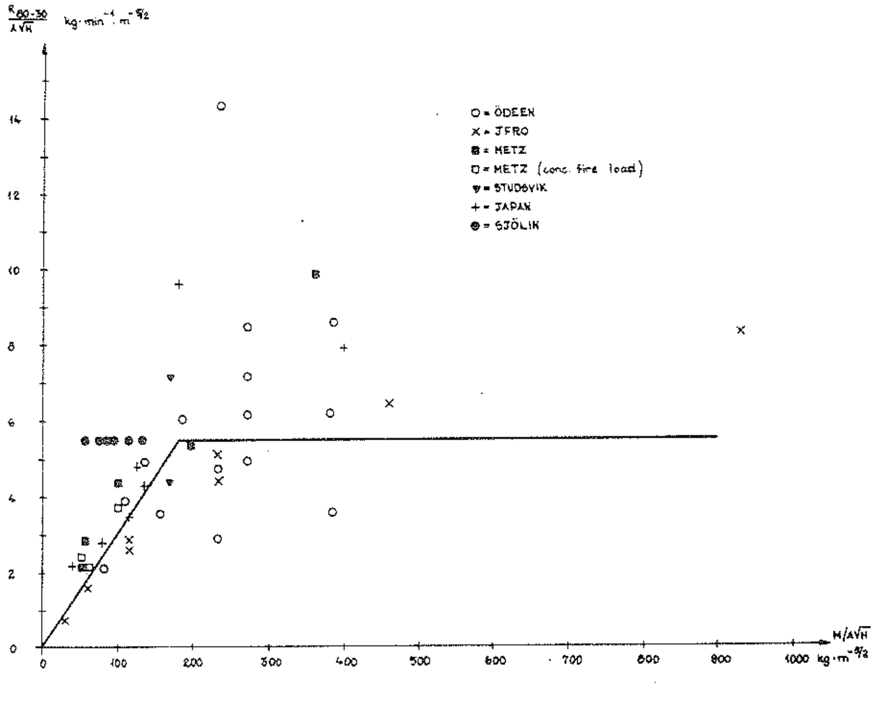
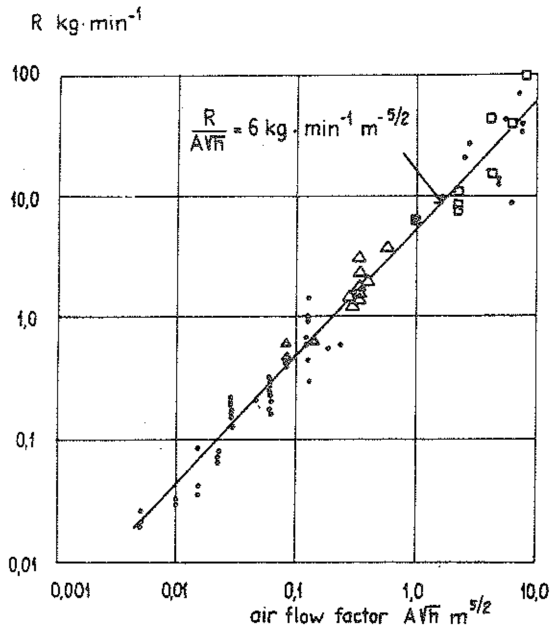


Fig. 8

Fig. 8 Relation between $R_{80-30}/A\sqrt{H}$ and $M/A\sqrt{H}$ for full-scale fire test performed at various research stations.



Experiment	Floor area ft²	m²	Symbol
J.F.R.O.	1	0,097	•
	4	0,37	•
	9	0,83	△
J.F.R.O.	34	3,2	▣
J.F.R.O.	100	9,0	•
Kawagoe	11	1,0	△
	≈100	≈9,0	□

Fig. 9

Fig. 9

Relation between mean rate of burning R and air flow factor $A\sqrt{H}$.

If we follow Heselden [8] and try to correlate $R/A\sqrt{H}$ with the total fuel load/ventilation factor we get Fig. 8. If the rate of burning in the fuel bed controlled area was directly proportional to M up to the point where the ventilation control sets in, we would get the inclined line suggested in the plot. The scatter in Fig. 8 is considerable, especially in the ventilation controlled regime.

The tests accounted for in Fig. 8 make no adequate basis for a determination of R. In [15] and [16] compilations have been made of the results regarding R obtained in a great many fire tests in several laboratories. Fig. 9 is taken from [16], where it is moreover stated that Eq. (2:5) is "for the present the only useful relation for the estimation of fire temperature vs. time curve in buildings". In [15] it is concluded on the basis of tests carried out in Denmark, Japan, USA and USSR that the value of the konstant k in the relation $R = k\ A\sqrt{H}$ is about 5.5 to 6 $kg \cdot min^{-1} \cdot m^{-5/2}$. Prerequisite conditions are that there is sufficient fuel for the fire to be ventilation controlled and that the ventilation area A is approximately one-quarter of the area of one side of the compartment, or less. The burning rate in the ventilation controlled regime is on this basis taken to be constant and equal to 5.5 $A\sqrt{H}$ in fig. 8.

If we try to reduce the scatter in fig. 8, some variable describing the design of the fuel bed must be introduced. The most simple one is r or the hydraulic radius of the fuel. In [13] r was used for this purpose, and to be able to include these experiments in our analysis, we do the same. r expresses the ratio of the total volume of the fuel to its total bounding surface area. If the width and the thickness of a rectangular piece of wood are denoted by w and t we get

$$r = \frac{w \cdot t}{2(w+t)}$$

w = t (square cross section) gives
$r = t/4$ and if $w >> t$ we get $r = t/2$. In Fig. 10 $R/A\sqrt{H}$ is plotted against $M/(r\ A\sqrt{H})$. Only tests for which r is stated could be plotted.

As seen, the points in the plot are distributed around a line from the origin of coordinates up to $R/A\sqrt{H} \approx 5.5$. The slope of this line in a way denotes the average rate of burning of all the tests in the fuel bed controlled regime and is thus a parameter of interest. Let v_i denote the rate of burning or the progress of the charred layer expressed in $m \cdot min^{-1}$. Then v_i for each test is given by the slope of the straight line connecting the tests in the $R/A\sqrt{H}$ - $M/(r \cdot A\sqrt{H})$ plot. For if the weight loss R is measured in kg·min^{-1} and v_i in m·min^{-1} then

$$R = \frac{M \cdot v_i}{r} \text{ kg·min}^{-1}$$

and hence

$$\frac{R}{A\sqrt{H}} = v_i \frac{M}{rA\sqrt{H}}$$

The average value of v_i, $v_{i,av}$ given by Fig. 10 will be

$$v_{i,av} = 0.3 \text{ mm·min}^{-1}$$

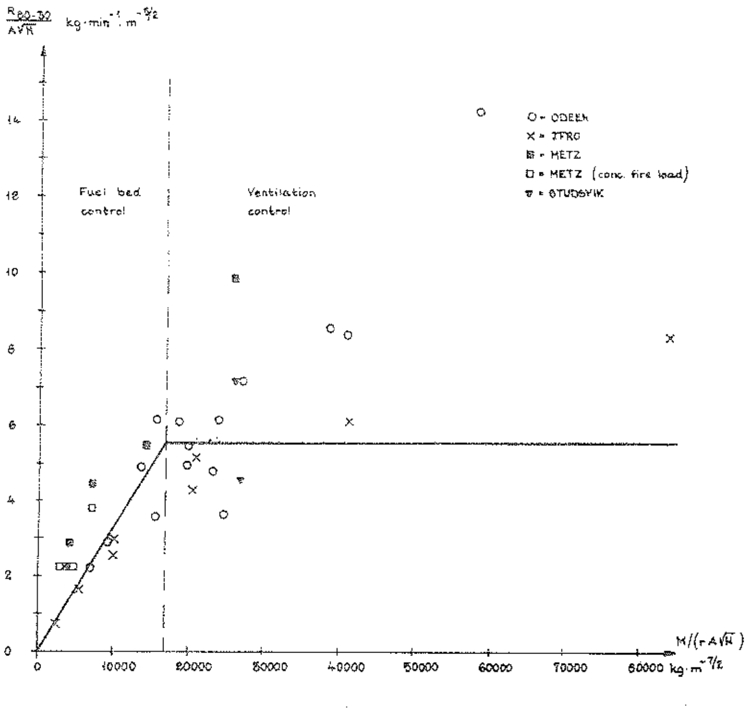


Fig. 10

Fig. 10

Relation between R_{80-30} $/A\sqrt{H}$ and $M/(rA\sqrt{H})$ for full-scale fire tests performed at various stations.

The values of the ratios $M/A\sqrt{H}$ and $M/(r \cdot A\sqrt{H})$ that marks the transition from fuel bed control to ventilation control will as averages be $M/A\sqrt{H} = 175$ kg m$^{-5/2}$ and $M/(r \cdot A\sqrt{H}) = 17\,000$ kg·m$^{-7/2}$. In theory a reasonable estimate of the rate of burning can now be made if M, $A\sqrt{H}$ and r are known and the fire process is fuel bed controlled, i.e. $M/(r\ A\sqrt{H}) < 17\,000$ m$^{-7/2}$. Then $R = M \cdot v_{i,av}/r$ kg·min^{-1} and the temperature-time curve can be computed from the energy balance equation with the reduction of the rate of gas flow derived earlier taken into account. But it must be emphasized that the value $v_{i,av}$ given in this way is to be seen as a rough effective average value. The rate of burning depends besides M, r and $A\sqrt{H}$ on other factors. Most important of them is probably the effective ventilation area inside the fuel bed (the porosity ϕ, see Fig. 6) and the moisture content of the fuel. The materials in walls, ceiling and floor surrounding the fire compartment also have an influence, different materials providing different amounts of thermal feedback to the fuel bed. Moreover, the ratio width/depth of the compartment must have an influence on the burning rate.

When a particular fire load display produces a fuel bed controlled fire, this is always a favourable effect. In most practical cases an accurate value of r is hard to come by. This means that $M/A\sqrt{H}$ must be used as a measure in what way the fire will be controlled. It must be stressed that the use of the favourable effect of $M/A\sqrt{H} < 175\ kg \cdot m^{-5/2}$ as a prerequisite condition has the fact, that the fuel consists of wood sticks piled in cribs. In one of the JFRO-tests [7] with $M/A\sqrt{H} = 58\ kg \cdot m^{-5/2}$ the fire load was altered from wood cribs to equivalent amounts of fibre insulating board lining the walls and the ceiling. By this the fire exposed surface was doubled. The maximum rate of burning was increased by a factor ≈ 3. The change of type of fuel thus had the effect that the process of fire became controlled by the ventilation.

More examples that the ratio $M/A\sqrt{H}$ cannot alone determine if the fire process will be ventilation controlled or not, is offered by Sjölin [14]. In [3] some of his tests are theoretically analysed. Sjölin's experiments are of a special value as the fire load consists of authentic furniture. 6 of these tests had an effective $M/A\sqrt{H}$-value = 58, 74, 135, 115, 91 and 83 $kg \cdot m^{-5/2}$. In 4 of these a door burnt through a few minutes after flash over and the $A\sqrt{H}$ value used includes this door. For all of them good agreement between theoretical and experimental time-temperature curves was obtained with the maximum value of $R = 5.5\ A\sqrt{H}$, i.e. the fire process was for all the tests controlled by ventilation. An assumption that these fires behaved as if the fuel had been sticks piled in cribs would have led to erroneous results on the unsafe side.

In the ventilation controlled regime Fig. 7 indicates that the constant k in the equation $R = k \cdot A\sqrt{H}\ kg \cdot min^{-1}$ as explained earlier exceeds the value 5-6 for the opening factor $A\sqrt{H}/A_t \leq 0.035\ m^{1/2}$. There is a tendency in Fig. 7 indicating that the divergence of k from 5.5 grows larger as $A\sqrt{H}/A_t$ grows smaller. In practical cases, such as compartments, schools and office buildings $A\sqrt{H}/A_t$ rather seldom is lower than 0.03 $m^{1/2}$. Furthermore, burning under insufficient air access leads to fires with a lower heat value of the fuel. Together with the fact that a more accurate value of R for the general case cannot be predicted, this makes $R = 5.5\ A\sqrt{H}\ kg \cdot min^{-1}$ the most reliable estimate in the area $A\sqrt{H}/A_t < 0.04\ m^{1/2}$ and $M/A\sqrt{H} \geq 175\ kg \cdot m^{-5/2}$.

4. Conclusions

1. The numerical simulation of the process of fire development has proved a valuable tool when investigating the influence of different parameters of this process. The solution of the energy or heat balance equation requires certain assumptions regarding the rate of energy release, I_C, during the fire. Independent of how I_C varies with the time, the condition

   $$W \cdot M = \int I_C dt$$

   must always be fullfilled.

2. For fires where the fire load consists of wood sticks piled in cribs the comparative theoretical analysis has given results that can be formulated in this way: As a rough estimate, the process of fire development ceases to be ventilation controlled when

   $M/A\sqrt{H} \leq 175 \text{ kg}\cdot\text{m}^{-5/2}$ or

   $M/(rA\sqrt{H}) \leq 17000 \text{ kg}\cdot\text{m}^{-7/2}$

3. In this area, the fuel bed controlled regime, the mean rate of burning R_{80-30} for all the full scale tests is approximately proportional to M and M/r. The rate of burning for the individual test, however, can considerably diverge from the mean value.

4. Swedish tests with the fire load consisting of furniture and with $M/A\sqrt{H} < 175 \text{ kg}\cdot\text{m}^{-5/2}$ have in a number of cases resulted in a ventilation controlled process of fire development.

5. The amount of fuel is seen to be of importance to the rate of burning in the ventilation controlled regime, too. This is especially true for very small openings. But the influence is weaker, the rate of burning not linearly increasing with M as in the fuel bed controlled regime.

6. The rate of gas flow out of openings is proportional to $A\sqrt{H}$ as deduced by Kawagoe. For compartments with large openings and a rather small fire load, there is a reduction of this proportionality, but only to a factor 0.7 - 0.8 for openings up to $A\sqrt{H}/A_t = 0.12 \text{ m}^{1/2}$.

7. For ventilation controlled fires the most representative value of the burning rate is $R = 5.5\ A\sqrt{H}\ kg \cdot min^{-1}$. This formula is confirmed in [15] and [16], where fire tests performed by research bodies in several countries were compared with regard to the rate of burning. Provided that there is sufficient fuel for the fire to be controlled by the air inflow and that the value of the opening factor $A\sqrt{H}/A_t$ is higher than about 0.02 $m^{1/2}$, the most reliable estimate for R is the value given above.

When for a given fire compartment the amount of initial fuel is reduced below a certain limit, the design of the fuel bed will be decisive for R. The value of this transition point, however, varies with the fire load display. Even if the type of the fire load is restricted to cribs of wooden sticks, the boundary between the two regimes is fluctuating. If fire loads with a more authentic exposure geometry, such as furniture, are included the difficulty in predicting the actual rate of burning will increase. The tests performed by Sjölin with furniture as fire load underscore this difficulty, giving ventilation controlled fire processes for markedly low $M/A\sqrt{H}$-values.

When a process of fire development has a lower rate of burning than that given by the air inflow, the result will be a fire of less severity with regard to the fire resistance of structural elements. Sometimes this favourable effect will be neutralized by a reduction of the theoretical rate of gas flow and it is far from clear under which initial conditions regarding the fire compartment and the fire load a fuel bed controlled process will take place. A general use of the favourable effect of such a process in a theoretical structural fire engineering design will require further studies of how the geometrical and thermal properties of the combustible material and of the fire compartment affect the process of fire. Therefore, the conclusion must be this: Given characteristic data for the fire compartment and the fire load, the only possible estimate of the temperature-time curve of the combustion gases - with regard to the requirement of giving results not on the unsafe side - is obtained using the assumptions that the process of fire development is ventilation controlled and that the rate of burning is $5.5\ A\sqrt{H}\ kg \cdot min^{-1}$.

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