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FULLY-DEVELOPED FIRES IN A SINGLE COMPARTMENT

PART II : EXPERIMENTS WITH TOWN GAS FUEL AND ONE SMALL WINDOW OPENING

by

A. J. M. HESELDEN

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SUMMARY

Experiments are described with a small compartment supplied through a large burner with town gas fuel. Above a critical fuel flow the heat transfer decreased with increasing flow. One reason for this was that the rate of heat release within the compartment depended directly on the rate of flow of air into the compartment; the rate of air inflow fell as fuel flow was increased, the total flow of gases out of the compartment increasing slightly, so that the flame temperature and heat transfer fell with increasing fuel flow. The other reason was due to the change in the distribution of the flame within the compartment. Measurements of the convective and radiative components of heat transfer, of flame and wall radiation, and of the heat balance of the compartment are reported and discussed.

The measurements are in broad agreement with calculations based on a very simple theoretical model, which in principle can extend the results of these and similar experiments to other scales, fuels and wall materials.

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1. Introduction

Experiments have shown [1, 2, 3] that after an initial period of growth a fire burning in an enclosure reaches a quasi-stationary state, usually when volatiles are being liberated from most fuel surfaces, when the rate of burning is relatively constant, though its value may vary with the various parameters of the compartment, e.g. size, amount of fuel, window area etc. This period is of major importance for determining the fire resistance requirements of the containing structure. The constancy of the burning rate implies control by some process or combination of processes, which may be different for different situations. A knowledge of these processes would enable more effective use to be made of experimental data on fires in compartments and provide a proper basis for the development of modelling techniques.

The decomposition of the solid fuel (usually wood) is maintained by heat returned to it partly by radiation and convection transfer from the flame zone and perhaps recirculating combustion products, but also by radiation from the walls and ceiling of the compartment as these heat up. The present report describes measurements of the feedback of heat from the flame and its components and the heat balance of an artificial fire supplied with town gas fuel in a compartment measuring internally 72 cm high, 79 cm wide and 91 cm deep[4] with a window opening 79 cm wide, 19 cm high, the top of the opening 19.5 cm from the ceiling. The first report[4] of this series described the apparatus used and measurements made.

2. Results

2.1. Flame appearance and position

The flame was produced on the burner[4], the base of the flame covering the whole of the burner area. A board in the plane of the window opening extended for 1.2 m above the top of the opening. At low gas flows the flame had the shape shown in Fig. 1a. As the fuel flow increased the shape changed to that sketched in Figs. 1b and 1c and a potential core of unburned gas could be seen at the base of the flame. The larger the fuel flow the greater the height of the flame above the window opening.

2.2. Heat flow within the compartment

2.2.1. Heat flow to cold receiving surfaces

Heat flows were measured at a number of points with a water-cooled heat-flux meter[5]. Fig. 2 shows that as the town gas flow increased from 2.5 g/s, the lowest flow that was used, the total downward heat transfer to a point 13 cm above the burner centre rose to a maximum at a town gas flow of about 3-4 g/s and then fell steeply. It was observed that as the town gas flow was increased, the tip of the flame just emerged from the window opening at the same flow (about 3-4 g/s). This will henceforth be called the critical flow. At town gas flows lower than the critical flow the flame was small and was probably largely unaffected by the enclosing compartment so that flame size and heat flow increased with increasing gas flow. However, above the critical flow the heat flow fell as the gas flow increased. The flames from fully-developed fires emerge from the window opening and therefore in connection with fire resistance the region of interest is that above this critical fuel flow.

Figure 3 shows the variation of downward heat flow along a horizontal line 13 cm above the burner, from the centre of the rear wall to the centre of the front, and Fig. 4 shows the variation of sideways heat flow to the heat-flux meter placed with its receiving surface vertical, in a horizontal line perpendicular to the plane of the window opening, 10 cm above the burner, 13 cm from the side wall. Not every feature of Figs. 3 and 4 can be explained but the following conclusions can be drawn:

(1) At the lowest town gas flows the heat flow was greatest towards the rear of the compartment, and at the highest gas flows it was greatest towards the front. This was probably due to the changes in flame shape with town gas flow since at low flows the main radiating body of flame was in the rear part of the model (Fig. 1), and at high flows the flame became thinner at the rear and there was more flame in the front. The heat transfer to points within the flame will be enhanced by convection transfer and an estimate of the difference in the heat transfer rate in and out of the flame can be obtained from the measurements for the position in the flame zone above the burner, referred to below in Table 1. These data show that the downward convection transfer to a horizontal surface was about 0.2 cal $cm^{-2}s^{-1}$. Comparison with relations[6] established for various surface orientations shows that the convection transfer to a vertical surface could be expected to be about twice that of Table 1, i.e. about 0.4-0.5 cal $cm^{-2}s^{-1}$.

(2) For any given position heat flow fell as fuel gas flow increased.

(3) For any given gas flow the mean heat flow along both horizontal paths was nearly the same.

The heat flux to a point in the compartment is the sum of three components:-

(a) radiation from the flame, both direct and reflected from the wall

(b) radiation from the wall, some of which will be absorbed by the flame

(c) convection transfer from the flame or recirculating combustion products.

These components have been estimated[4] for three gas flows, for the point 13 cm above the burner centre, and are given in Table 1. Although the sum of the flame and wall radiation is known with reasonable accuracy the separate values are approximate since they depend to some extent on the value assumed for flame emissivity (Reference 1, Table 3), which is not known precisely. However, it can be said that the flame radiation was of the order of $\frac{1}{3}$ to $\frac{1}{2}$ of the wall radiation for all three gas flows. The convection transfer remained fairly constant as fuel flow increased and its relative importance therefore increased as fuel flow increased, and the total heat transfer fell.

Table 1

Components of downward heat transfer to cold heat-flux meter, 13 cm above burner centre

Town gas flow g/s	Heat transfer cal cm^{-2} s^{-1}				
	Flame radiation (R_F)	Wall radiation after flame absorption (R'_W)	Flame convection (C)	Sum (i) (R_F + R'_W + C)	Independent total (ii)
4.5	0.7	1.6	0.25	2.55	2.6
12	0.25	0.7	0.2	1.15	1.0 - 1.1
21	0.1	0.35	0.2	0.65	0.6

Notes for Table 1

(i) (C) and (R_F + R'_W) were measured directly with the convection/radiation heat-flux meter(4). $R'_W = R_W(1-a)$ where R_W is the wall radiation and a an effective flame absorptivity. R_W was measured by abruptly turning off the town gas supply, and R'_W and R_F were obtained assuming $a = 0.1$.

(ii) From water-flow calorimeter(5).

Readings of total heat flow for 3 other positions shown in Fig. 5 are given in Fig. 6 with the curve found for the position over the centre of the burner. The heat flows to the total heat-flux meter inserted in a hole in the rear wall (Position A) were very similar to the downward heat flows to a point 13 cm above the burner centre (Position B), but the heat flows to points near the window opening (Positions C and D) although similar to those for positions A and B at town gas flows greater than about 9 g/s were much lower at the low town gas flows. This is probably due to the configuration of the flame and the effect of this configuration on the interior surface temperature. There was probably hardly any convection transfer to positions C and D.

2.2.2. Heat flow through hot walls

The heat flow was measured through the walls at two points and through the ceiling at one point, and since the temperature on the inner surface was also measured at these points (Fig. 9), it was possible to estimate the equivalent heat flow to a cold body assuming the body to be small enough not to affect the flame temperature. There are differences between the values calculated (Table 2), probably due to the configuration of the flame, but they are in broad agreement with the measured downward heat flow to a cold heat-flow meter at a point 13 cm above the burner centre, as would be expected in view of the importance of wall radiation (Section 2.2.1).

TABLE 2

Heat flows to interior surfaces

Position	Town gas flow g/s	Measured temperature of inner surface of wall or ceiling °C	Calculated radiation from inner surface of wall or ceiling[a] cal cm^{-2}s^{-1}	Measured heat flow through wall or ceiling[b] cal cm^{-2}s^{-1}	Approximate equivalent heat flow to a cold black body in the wall or ceiling[c] cal cm^{-2}s^{-1}	Measured downward heat flow to a cold surface 13 cm above burner centre[d] cal cm^{-2}s^{-1}
Centre of side	4.5	840	1.95	0.24	2.2	2.5
	12	675	1.07	0.22	1.3	0.8
	21	590	0.71	0.16	0.9	0.3
Centre of back	4.5	880	2.26	0.25	2.5	2.5
	12	615	0.78	0.16	0.9	0.8
	21	440	0.33	0.10	0.4	0.3
Centre of ceiling	4.5	865	2.15	0.33	2.5	2.5
	12	640	0.90	0.24	1.1	0.8
	21	505	0.48	0.16	0.6	0.3

(a) Assuming an emissivity of 0.95[7]

(b) Calculated from measurements of exterior surface temperature

(c) Obtained by adding values in the previous two columns and neglecting convection transfer.

(d) Subtracting the convection transfer of 0.2 cal cm^{-2}s^{-1}

TABLE 3

Flow and combustion data

Town gas flow[a] g/s	Air flow into window opening[b] g/s	Flow of gases out of window opening[c] g/s	Thermal capacity of effluent gas[d]		Heat release within compartment[e] k cal/s	Mean temperature of gases flowing out of window opening[f] °C
			cal g^{-1} $degC^{-1}$	cal s^{-1} $degC^{-1}$		
4.5	32	36.5	0.30	11	24	735
12	28	40	0.37	15	21	720
21	23	44	0.44	19	18	565

(a) Measured by orifice plate and manometer.

(b) Calculated by method described in ref. (4).

(c) Adding values in the previous two columns.

(d) Using values quoted by Spiers[4], and the calculated flows of combustion products and unburned fuels.

(e) Assuming all air entering window opening was used in combustion, taking values for the air fuel ratio and calorific value from reference (4), and assuming combustion was complete.

(f) Estimate from unweighted mean of 4 thermocouples in window plane[4].

2.3. Air flow into compartment and heat release

Because the flame just began to emerge from the window opening at about the critical flow, the rate of entry of air into the compartment was more than sufficient for combustion of the whole of the fuel gas at flows below the critical fuel gas flow. Additional measurements summarised in Table 3 were taken for three flows above the critical flow, and the air flowing into the compartment was calculated(4).

Table 3 shows that as the fuel flow increased from 4.5 to 21 g/s the air flow into the compartment decreased by about 25 per cent; this was because the neutral axis fell and the area over which air entered the model decreased.

Measurements of the concentration of oxygen in the effluent gases have not yet been made for the window opening of the compartment described in this report, but measurements have been made for a window opening 30 cm wide and 35 cm high with the box over the burner removed(4), so that the gas entered the compartment through 24 nozzles each 2 cm in diameter. For town gas flows of 6 and 12 g/s the measured oxygen concentration in the upper half of the window opening was less than 0.5 per cent and for a flow of 22 g/s values were obtained varying from 0.3-0.7 per cent near the top of the window opening to 2-3.5 per cent at a third of the height from the top. These values imply that substantially all the oxygen entering the window opening was being consumed in the combustion.

Measured carbon dioxide concentrations (carbon monoxide was not measured in these experiments) were about 70 per cent of the values expected from complete combustion, but the incompleteness of combustion will have quite a small effect on the heat release within the compartment, since there is practically no excess air. It has been calculated that if the town gas burned to produce water vapour, nitrogen and carbon monoxide alone, the heat release within the compartment would be only 10 per cent lower than that given by stoichiometric combustion, because although less heat is evolved by incomplete combustion per gram of town gas, less oxygen is required, and a larger proportion of the town gas can burn inside the compartment. Table 4 shows that the sum of the various heat losses measured in the present experiments is approximately equal to the heat that would be released by complete combustion of town gas with the inflowing air, so that even though the geometry and burner construction were rather different the oxygen utilisation must still have been high.

For these fuel flows the heat released within the compartment was therefore directly controlled by the air inflow and fell with increasing fuel flow.

Table 4

Measured components of heat loss

Town gas flow g/s	Measured heat losses kcal/s				Calculated heat release kcal/s
	Through walls, ceiling and floor	Sensible heat (Enthalpy) of hot effluent gases	Radiation through window opening	Total	
4.5	9	8	2	19	24
12	8	10	2	20	21
21	6	11	1	18	18

2.4. Heat losses

Heat released within the compartment was lost in three main ways:-

(1) By conduction through the walls, ceiling and floor.

(2) By radiation through the window opening.

(3) By the hot effluent gases as sensible heat.

Values of these losses derived from measurements(4) for three gas flows are given in Table 4. The specific heat of the effluent gases increased with increasing fuel flow (Table 3) since the specific heat of town gas is much higher than that of its combustion products and a larger proportion of unburned town gas left the compartment at higher flows. Above the critical flow the temperature of the effluent gas, estimated from readings of thermocouples(4), fell as flow increased. If an allowance were made for the probable error in the thermocouple reading due to radiation loss(4) this fall would be even more marked.

2.5. Calculations of temperature and heat flow

Temperatures and heat flows have been calculated for a highly simplified theoretical model of the compartment to find to what extent it is possible to predict heat flow and temperature.

The assumptions made were as follows:-

(1) The heat release within the compartment was due to complete combustion of fuel with the whole of the inlet air, and was therefore known.

(2) The flame filled the whole of the compartment, its temperature was uniform and was equal to the temperature of the effluent gases, and its emissivity with respect to all points on the inner surface was constant.

(3) The temperature of the inner wall surface was uniform.

Calculations were made for various values of flame emissivity and for 3 fuel gas flows by equating the steady state heat transfer from the flame to the interior surface of the compartment to the rate of conduction of heat through the walls, ceiling, etc. and to the rate of loss of heat from the outside surfaces. The latter can be expressed as a function of outer surface temperature and, if the thermal resistance of the wall is known, as a function of the interior surface temperature.

The heat balance for the flame within the compartment can be written

$$\underset{1}{Q_1} = \underset{2}{(T_F - T_o)Mc_F} + \underset{3}{Q_2} + \underset{4}{A\sigma E\,(T_F^{\,4} - T_W^{\,4})} + \underset{5}{Ah\,(T_F - T_W)^{4/3}} \quad \ldots\ldots\ (1)$$

where Term 1 is the rate of heat release within the compartment

Term 2 is the rate of sensible heat loss in the effluent gases, of absolute temperature T_F, mass flow M, and specific heat c_F. T_o is the inlet air and fuel absolute temperature.

Term 3 is the rate of radiation loss from the window, which is fairly small relative to the heat release when the window is small, and therefore unlikely to be of major importance in the heat balance.

Term 4 is the net rate of radiation transfer from the flame to the wall, at temperature T_W. A is the area of the inner surface and E is the effective emissivity of flame and wall.

Term 5 is the rate of convection transfer from the flame to the wall, 'h' being a mean convection coefficient for all surfaces within the compartment. (4 + 5) can be expressed as a known function of T_W and the equations solved, by successive approximation, to give T_F and T_W for assumed values of E.

To simplify the calculation a constant value for Q_2 was assumed, even though this must vary with E, T_F and T_W.

A value of h of 3×10^{-5} cal $cm^{-2}s^{-1}$ $degC^{-4/3}$ was obtained by transformation of the direct measurements of convection transfer (Section 2.2.1.) and conversion to the equivalent transfer to a vertical surface or a cold surface facing downwards.

Since convection transfer appeared to take place only to the upper part of the compartment (Fig. 1) this value was thought to be too high to assume for a uniform heat transfer over the whole interior of the compartment and a value of 1.5×10^{-5} cal $cm^{-2}s^{-1}$ $degC^{-4/3}$ was taken. This is probably sufficiently accurate when convection transfer is small compared with flame radiation, i.e. for the lower fuel gas flows or for higher flame emissivities. The thermal resistance of the wall was estimated from measurements of the internal and external surface temperature and heat flow(4). A wall emissivity and absorptivity of 0.95 was assumed(7).

Calculated and experimental values are compared in Tables 5, 6 and 7. The effective flame emissivity, though not known precisely, is unlikely to exceed about 0.2(4), so that comparison between the experimental values and the values calculated should be made for flame emissivities of 0.1 and 0.2.

There is reasonable agreement between calculated and experimental values, in view of experimental errors and the errors introduced by the over-simplification of the theoretical model.

Calculated values are also given for one town gas flow in a compartment of the same internal dimensions but with concrete walls 10 cm thick of thermal conductivity 2×10^{-3} cal $cm^{-1}s^{-1}$ $degC^{-1}$. The effect of the higher insulation is seen to be to increase flame and inner wall temperatures and to decrease heat transfer to the walls.

Table 5

Calculated and measured temperatures

Town gas flow g/s	Effective flame emissivity E	Temperature (°K) Flame C	Flame M	Inner surface of wall C	Inner surface of wall M	Outer surface of wall C	Outer surface of wall M
4.5+	0.1 0.2 0.5 1.0	1320 1270 1210 1190	1010	1110 1140 1150 1160	1140	650 630 640 640	630
4.5*	1.0	1340		1330		535	
12+	0.1 0.2 0.5 1.0	1120 1080 1040 1030	990	880 930 970 980	920	550 570 580 580	570
21+	0.1 0.2 0.5 1.0	950 920 880 880	840	720 760 810 830	780	500 510 530 540	510

C = calculated

M = mean of measurements at several points

+ = 1.3-cm thick asbestos wood (Thermal conductivity 7×10^{-4} cal s^{-1} cm^{-1} $degC^{-1}$)

* = 10-cm thick concrete (Thermal conductivity 2×10^{-3} cal s^{-1} cm^{-1} $degC^{-1}$)

Area of walls, ceiling and floor 37,000 cm^2

Area of window opening 1,500 cm^2

Table 6

Calculated and measured heat flows

Town gas flow g/s	Effective flame emissivity E	Heat flow to cold heat-flow meter cal $cm^{-2} s^{-1}$: Flame radiation C	Flame radiation M_1	Wall radiation (after flame absorption) C	Wall radiation (after flame absorption) M_1	Flame convection C	Flame convection M_1	Total C	Total M_1	Heat flow through walls ceiling and floor cal $cm^{-2} s^{-1}$: C	M_2
4.5+	0.1	0.4	0.7	1.8	1.6	0.15	0.25	2.4	2.55	0.25	0.25
	0.2	0.7	0.9	1.7	1.4	0.15	0.25	2.55	2.55	0.27	
	0.5	1.5	–	1.1	–	0.1	–	2.7	–	0.27	
	1.0	2.7	–	0	–	0.1	–	2.8	–	0.28	
4.5*	1.0	4.5	–	0	–	0.15	–	4.65	–	0.16	
12+	0.1	0.2	0.25	0.7	0.7	0.1	0.2	1.0	1.15	0.18	0.20
	0.2	0.4	0.35	0.8	0.6	0.1	0.2	1.3	1.15	0.19	
	0.5	0.8	–	0.5	–	0.1	–	1.4	–	0.21	
	1.0	1.5	–	0	–	0.1	–	1.6	–	0.22	
21+	0.1	0.1	0.1	0.3	0.35	0.1	0.2	0.5	0.65	0.12	0.14
	0.2	0.2	0.15	0.35	0.3	0.1	0.2	0.65	0.65	0.14	
	0.5	0.4	–	0.25	–	0.1	–	0.75	–	0.15	
	1.0	0.8	–	0	–	0.1	–	0.9	–	0.16	

C = Calculated

M_1 = Measured downward heat flow to a point 13 cm above the burner centre

M_2 = Mean of measurements at several positions

+ = 1.3-cm thick asbestos wood

* = 10-cm thick concrete

Table 7

Calculated and measured heat losses

Town gas flow g/s	Effective flame emissivity E	Heat loss kcal/s								
		Flame radiation to wall	Flame convection to wall	Total to wall		Sensible loss in effluent gases		Radiation from window opening		
		C	C	C	M	C	M	C	C'	M
4.5+	0.1	7.8	1.4	9.2	9	11.3	8	3.3	3	2
	0.2	9.3	0.7	10.0		10.8		3.7	3.3	
	0.5	9.9	0.3	10.2		10.1		3.9	3.8	
	1.0	10.2	0.1	10.3		9.9		4.1	3.8	
4.5*	1.0	5.9	0	5.9		11.6		6.7	6.5	
12+	0.1	4.9	1.6	6.5	8	12.5	10	1.4	2	2
	0.2	6.2	0.9	7.1		11.8		1.7	2	
	0.5	7.2	0.4	7.6		11.3		2.0	2	
	1.0	7.8	0.2	8.0		11.1		2.3	2	
21+	0.1	2.8	1.6	4.4	6	12.6	11	0.6	1	1
	0.2	4.0	1.0	5.0		12.1		0.8	1	
	0.5	5.1	0.4	5.5		11.2		1.0	1	
	1.0	5.8	0.1	5.9		11.2		1.2	1	

C = Calculated from final flame and wall temperatures

C' = Value assumed in first stage of calculation

M = Measured

\+ = 1.3-cm thick asbestos wood

* = 10-cm thick concrete

Discussion

A fire in a compartment decomposes solid fuel into a gaseous fuel which burns in a diffusion flame, mainly turbulent if the flame is large enough, entraining air. It is likely that the rate of reaction of the gaseous fuel with oxygen is much more rapid than the rate of turbulent mixing of fuel and air[10]. With most fires flames emerge from the window and therefore insufficient air can mix with the gaseous fuel to burn it completely within the compartment and the remaining fuel flows out of the upper part of the window opening and burns outside the window by entraining additional air. When the window opening is large enough not to restrict the entry of air significantly the rate of combustion within the compartment, and therefore the heat release, may be limited by the rate of entrainment of air by the flame, but when the window is small the rate at which air can enter it may limit the combustion rate[11].

The duration of a fire will depend on the total quantity of solid fuel and the rate at which it burns. The temperature of the flame will be determined by the heat release, and the relative importance of the heat losses from the flame within the compartment. These will depend among other things on flame and wall temperature and flame shape and emissivity.

The fire may therefore depend to some extent on the wall material since the heat lost from the flame to the wall depends both on flame parameters such as emissivity and temperature, and wall parameters, e.g. its thermal properties and temperature. The way in which the released heat divides itself between the various losses depends on flame temperature and emissivity and its geometry which in turn depend on the heat loss from the flame. Emissivity may be linked with flame temperature, a higher temperature producing more cracking of hydrocarbons and more soot. Simms et al[3] found that increasing the insulation of a compartment increased the temperature of wood fires.

In the present experiments the mechanisms controlling the rate of feed-back of heat to a solid or liquid fuel were being sought, so that the burning rate could be found by equating feedback of heat and the heat required to produce gaseous fuel from the solid or liquid fuel[8].

The latter will be determined for wood fuel by exposing specimens to known intensities of radiation, measured by a cold thermopile, and this will help to overcome the difficulty which arises in converting measured gross heat transfer to a cold heat-flow meter to the equivalent net heat transfer to the burning wood, whose surface temperature is not usually known.

For the town gas flames in the present experiments the wall radiation was between two and three times the flame radiation, but as shown in Table 6 the more emissive the flame within the compartment, the more important flame radiation, and the less important wall radiation are likely to be. In a real fire the wall temperature will not usually attain an equilibrium value and the wall radiation may then be less important.

Above the critical flow, as fuel flow increased the temperature of the effluent gases fell (Fig. 7) and the radiation transfer from the flame fell relatively more than convection transfer (Table 1). Tables 5, 6 and 7 suggest that in part this is due to the stronger temperature dependence of radiation transfer.

Above the critical fuel flow, the heat release and heat transfer rates within the compartment decreased with increasing fuel flow. The heat release depended on the rate of flow of oxygen into the model, and therefore decreased with increasing fuel flow. The heat transfer decreased partly because of the lower flame temperature (and possibly also emissivity) produced by the sharing of a smaller heat release with a larger flow of effluent gases of higher specific heat (Table 3), and partly because of the change in the flame shape caused by the higher fuel gas flow. This altered the distribution of heat flow within the compartment. In Fig. 8 it can be seen that the heat flow to the centre of the rear wall and ceiling fell more rapidly than that to the centre of the side walls due probably to the effect of the flame shape on the distribution of heat flow within the model and the variation of flame shape with fuel flow. Similarly the curves of Figs. 3, 4, 6 show that the variation with fuel flow of heat flow was less at the front of the model than it was at the rear. This might well be more marked in the transient period when the walls were much cooler than the flame, a situation corresponding more to an actual fire.

The percentage fall in heat flow with fuel flow to a heat flow meter 13 cm above the burner centre was much larger than the percentage fall in heat flow to the wall, and this is a consequence of the low temperature of the heat flow meter receiver. When allowance is made for the inner surface temperature, the two heat flows agree broadly (Table 2).

The calculations of Section 2.5 can be taken to indicate approximately the expected variation of heat transfer and temperature with flame emissivity and it can be seen from Table 6 that both total heat flow to a cold heat flow meter and walls heat flow are relatively insensitive to flame emissivity since the increase of flame radiation with increasing emissivity is opposed by a decrease in wall radiation due to increased absorption in the flame. Further, flame radiation is not proportional to emissivity since the flame temperature falls as emissivity increases.

The relative proportions of the three main heat losses are not likely to be constant as gas flow changes. Even if the position, shape and emissivity of the flame were unaltered, its temperature must fall as fuel flow increases, since the heat release decreases and the thermal capacity of effluent gases increases. On the other hand, if the flame temperature were unaltered as fuel flow increased the flame emissivity must be less so that the radiation loss can be smaller and the sensible heat loss maintained.

The effect of fuel gas flow on heat transfer is also seen from the calculations. As fuel flow increases from 4.5 to 21 g/s the net heat flow to the wall is halved, and the total heat flow to a cold surface is reduced to about $1/5$. Convection transfer to the walls although unimportant at small gas flows, becomes relatively much more important at higher flows, although it is never important with higher flame emissivities, corresponding approximately to larger scales.

The calculated values of total heat transfer to a cold heat flow meter given in Table 6, plotted against fuel flow in Fig 10, ought to be given by any other gaseous fuel having the same calorific value per gram of stoichiometric oxygen as town gas*. Tentatively, this can include wood volatiles which have a net calorific value probably between 2,500 and 3,500 cal/g, and a stoichiometric air/fuel ratio probably between 4 and 4.5 g/g, making a calorific value of between 600 and 900 cal/g of stoichiometric air. The corresponding value for the town gas[4] is 780 cal/g air. For the size of compartment in these experiments the curves for emissivities of 0.2 to 0.5, which in any case are very similar, probably correspond most closely to the values for wood volatiles.

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*Apart from differences in density and specific heat of the fuels and combustion products, the effect of which will only be marked when air flow is comparable with the fuel flow, the same mass rate of flow of different fuels will give the same height of the neutral axis and the same rate of air inflow.

It should also be possible to derive the heat transfer for particular situations for other scales. For a large compartment geometrically similar to that employed in these experiments but n times linearly larger, with the same thermal resistance in the walls, apart from window radiation and convection to the walls, which are small compared with the other heat losses, the heat flows of Table 6 for E = 1 would be given by fuel flows increased by n^2 provided the window opening (assumed to be the same shape) were increased in linear dimension by $n^{4/5}$. This is because air flow into a window opening depends not only on an area but also on a velocity, and would increase by $n^{5/2}$ if there were no distortion.

Conclusions

1. For the same small window opening the fuel flow affected the position and shape of the flame and the flow of air into the compartment, the air flow decreasing as the fuel flow was increased. The distribution of the heat transfer within the model was affected by the flame shape.

2. The flame was relatively transparent and the heat transfer to a cold receiver was predominantly by radiation from the walls.

3. The total rate of heat release within the compartment depended directly on the rate of flow of air into the window opening and fell as the fuel flow increased so that the relative proportions of the heat losses from the compartment changed with fuel flow. The net heat transfer to the inner surface of the wall decreased and the sensible loss in the effluent gas increased with increasing fuel flow.

4. These variations are in reasonable agreement with calculated values for a very simple theoretical model.

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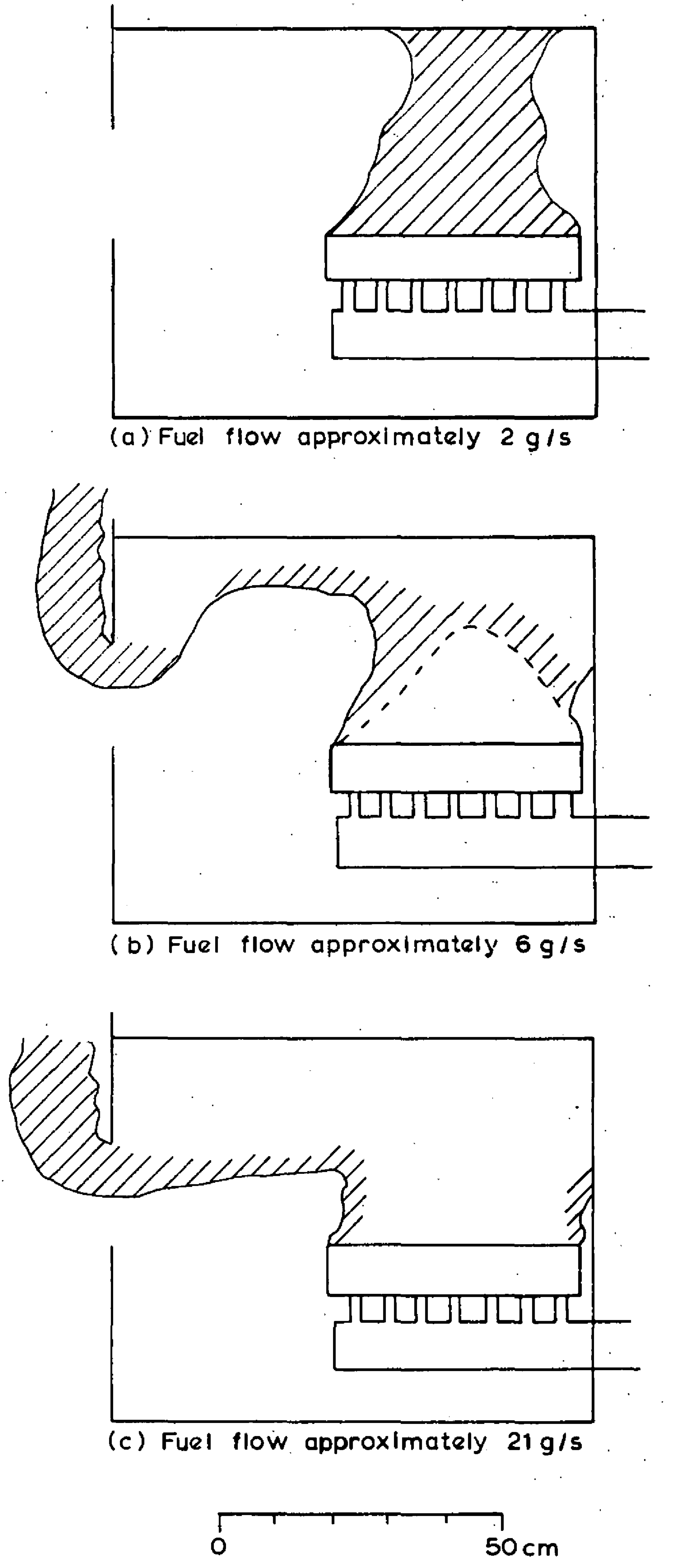

FIG.1. FLAME SHAPE AND FUEL FLOW

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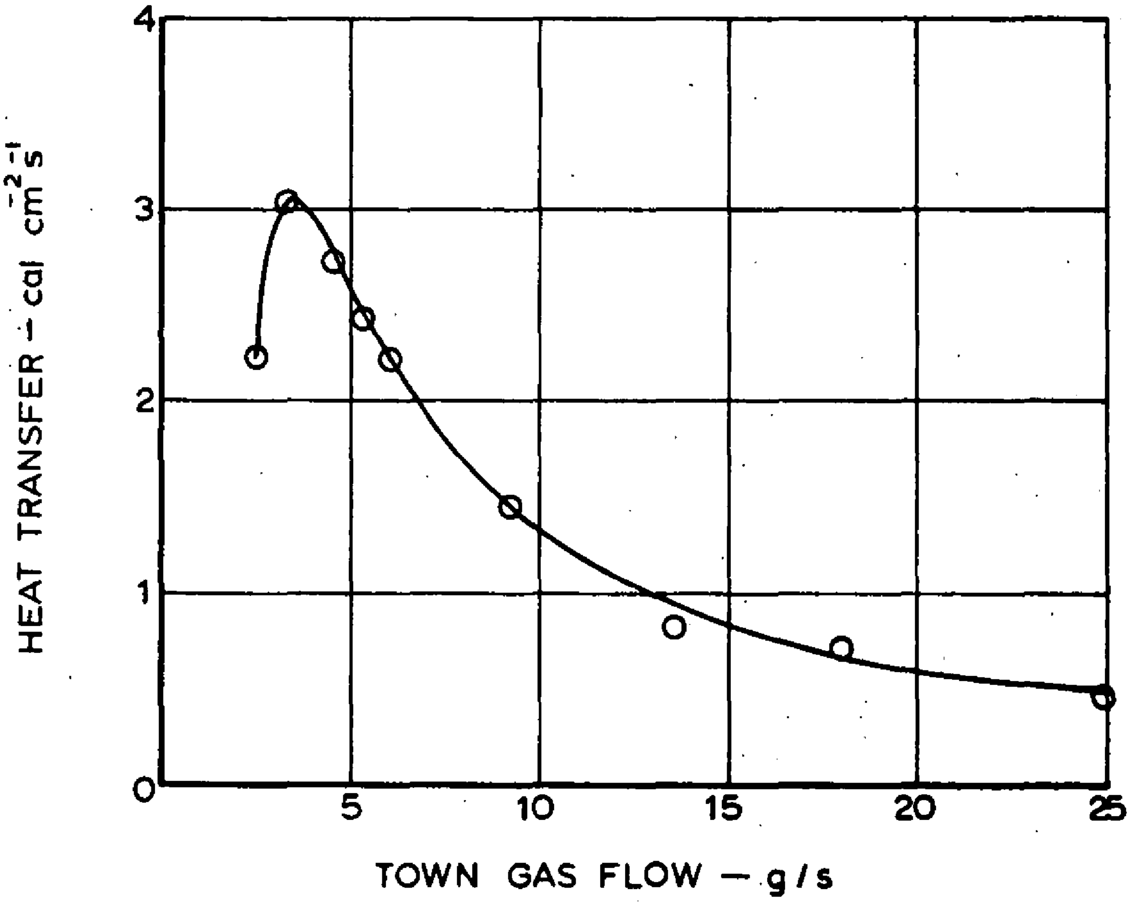

FIG.2. DOWNWARD HEAT TRANSFER TO THE WATER-COOLED HEAT FLOW METER 13 cm ABOVE BURNER CENTRE

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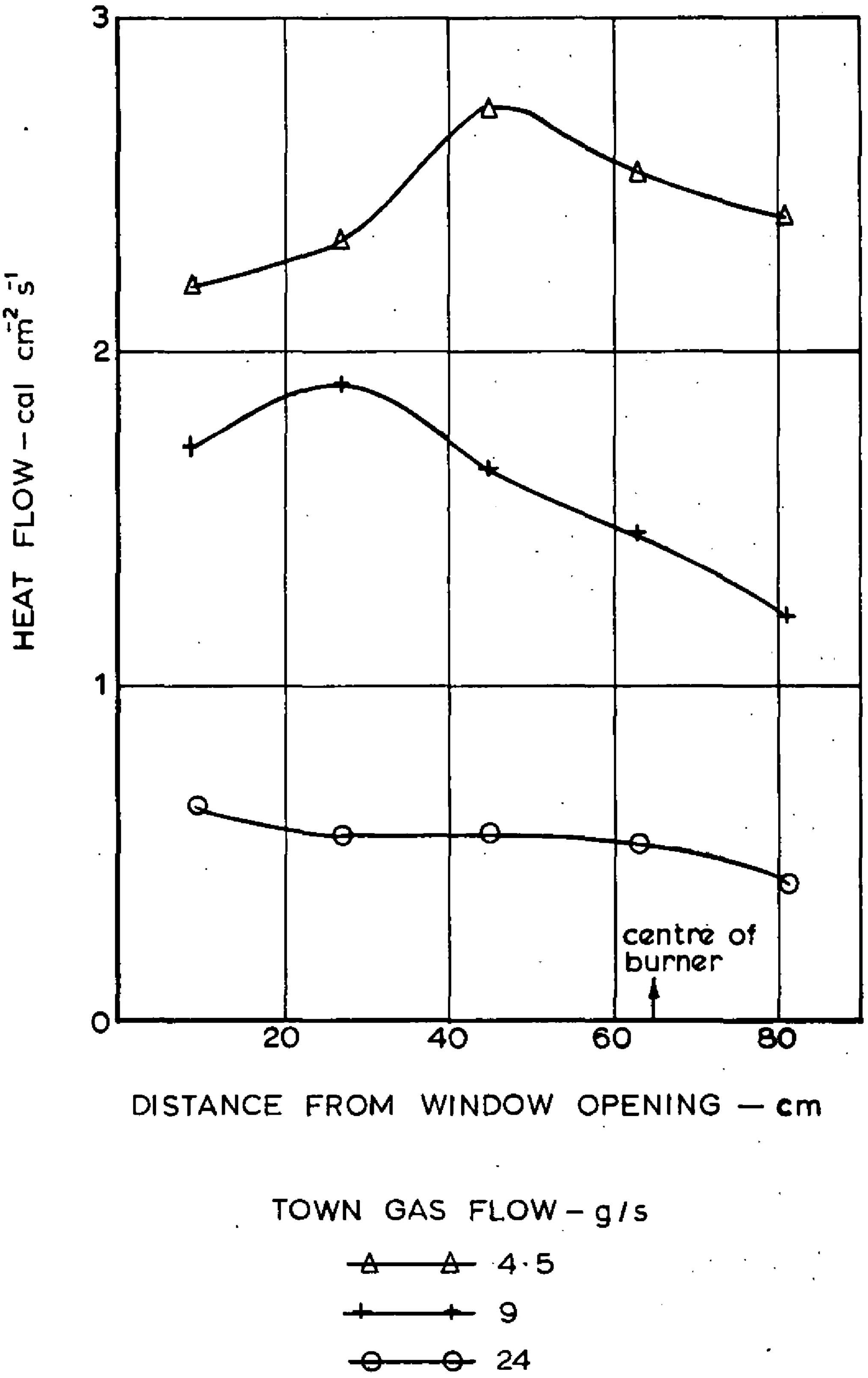

FIG. 3. DOWNWARD HEAT-FLOW ALONG A HORIZONTAL LINE FROM REAR WALL CENTRE TO FRONT WALL CENTRE, 13 cm ABOVE BURNER

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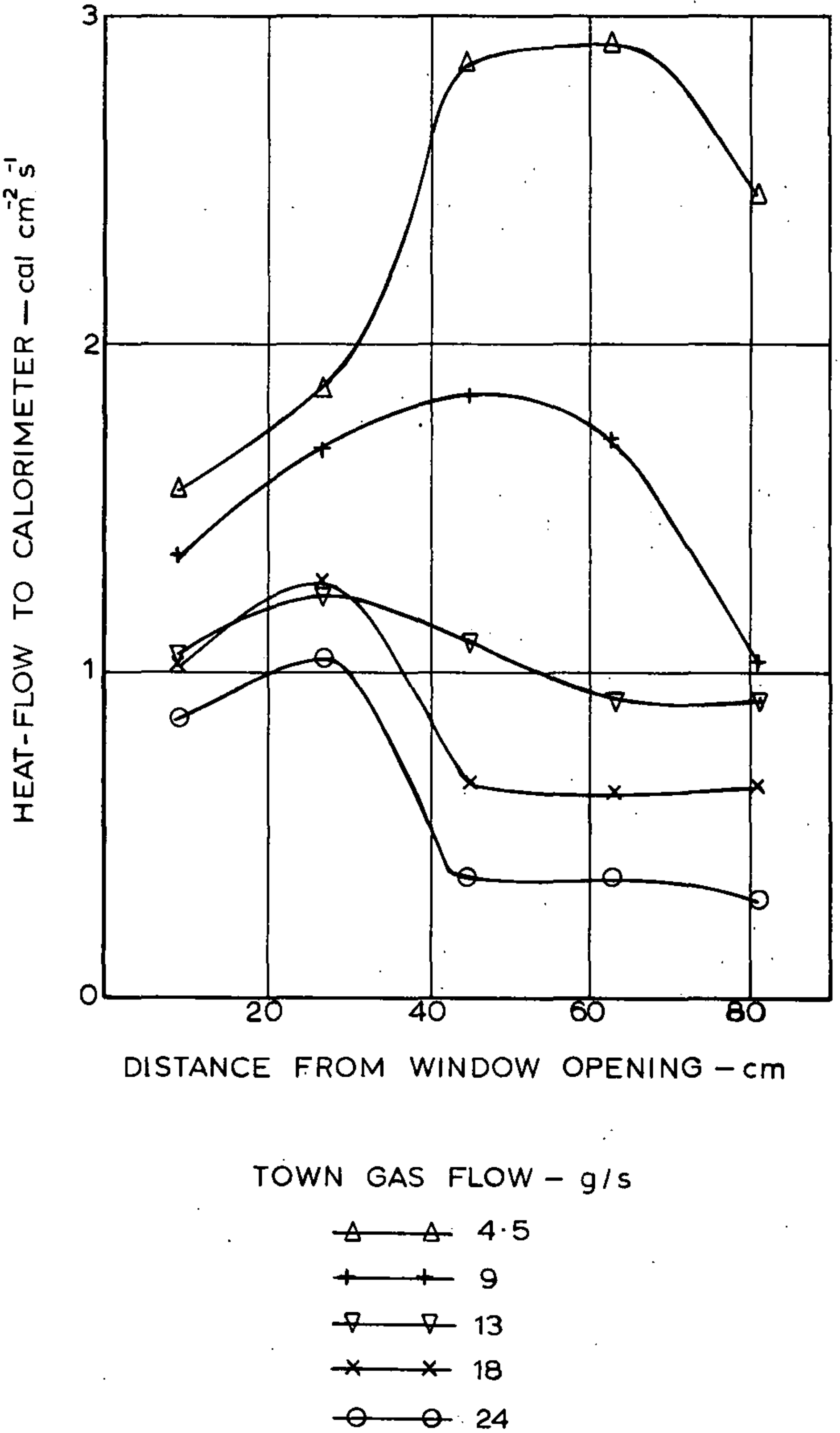

FIG.4. HEAT-FLOW TO A VERTICAL RECEIVING SURFACE IN A LINE PERPENDICULAR TO THE WINDOW OPENING 10 cm ABOVE THE BURNER AND 13 cm FROM THE SIDE WALL

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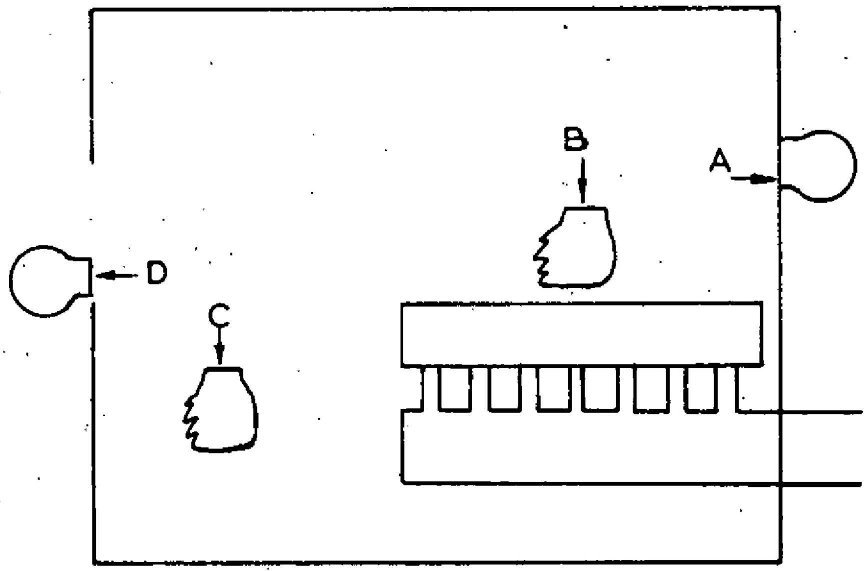

FIG.5. HEAT - FLOW METER POSITIONS FOR FIG.6.
THE ARROWS INDICATE THE GENERAL DIRECTION OF THE HEAT FLOW MEASURED.

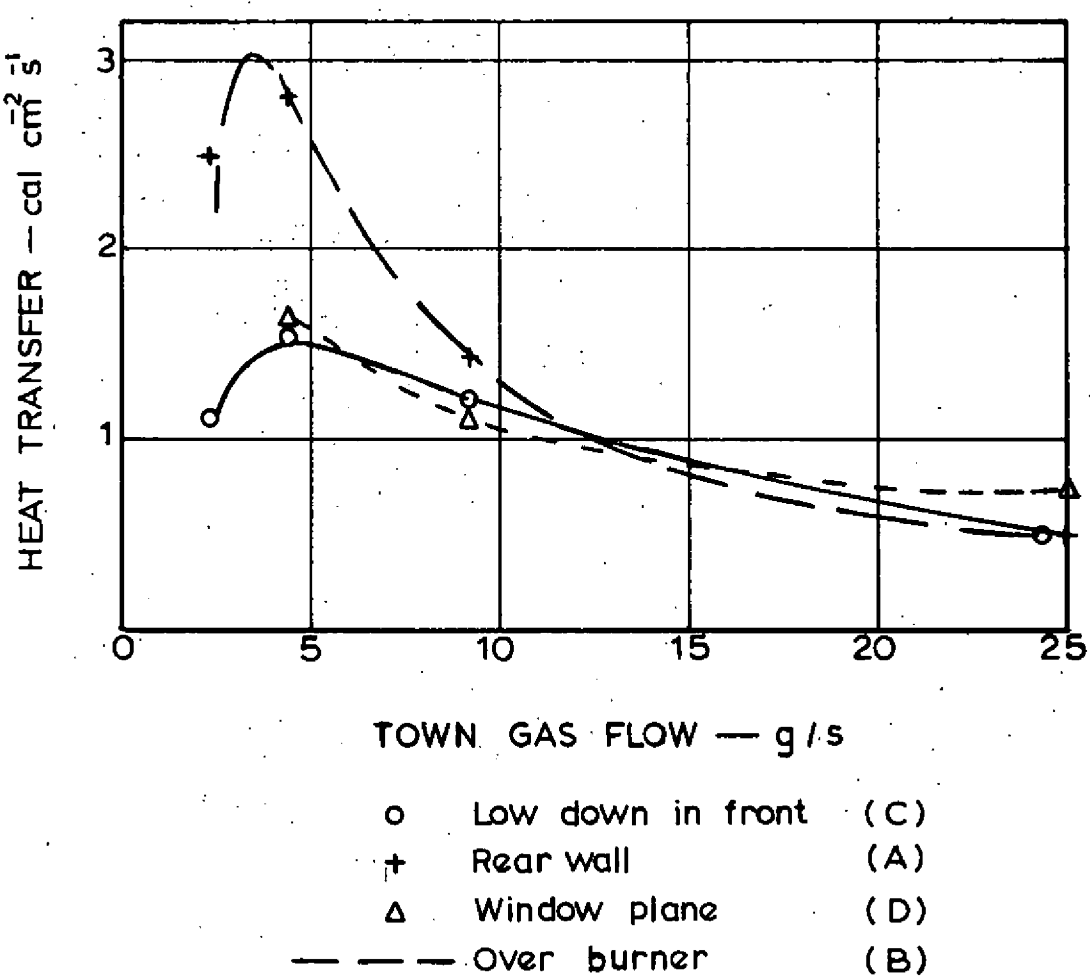

FIG 6 HEAT TRANSFER TO WATER-COOLED HEAT FLOW METER.
THE SMOOTH LINE OF FIG.2. IS REPRODUCED ABOVE, WITHOUT THE EXPERIMENTAL POINTS WHICH ESTABLISHED IT

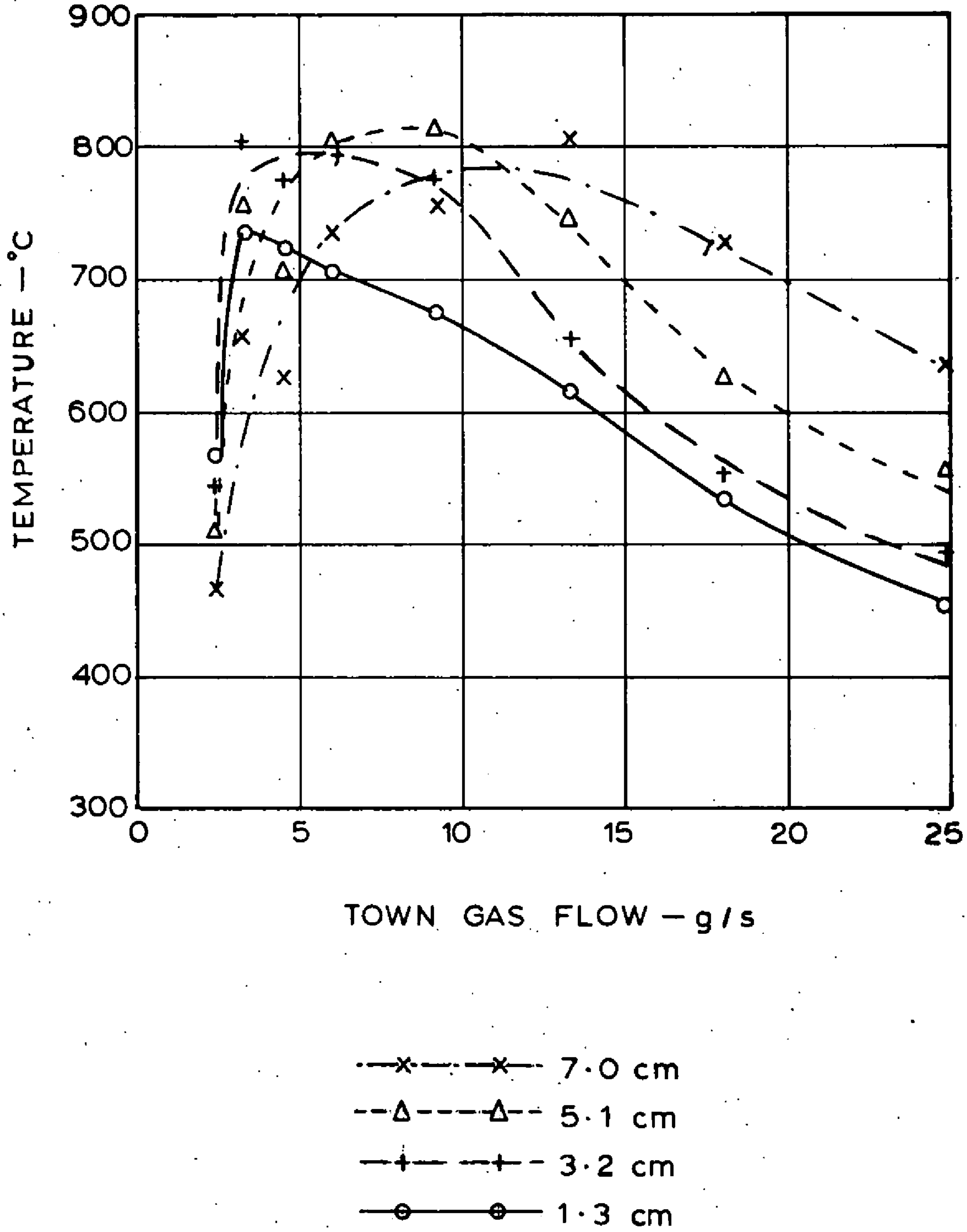

FIG.7. THERMOCOUPLE TEMPERATURES IN THE EFFLUENT GASES.
THE DISTANCE AGAINST EACH CURVE IS THE DISTANCE FROM THE TOP OF THE WINDOW OPENING

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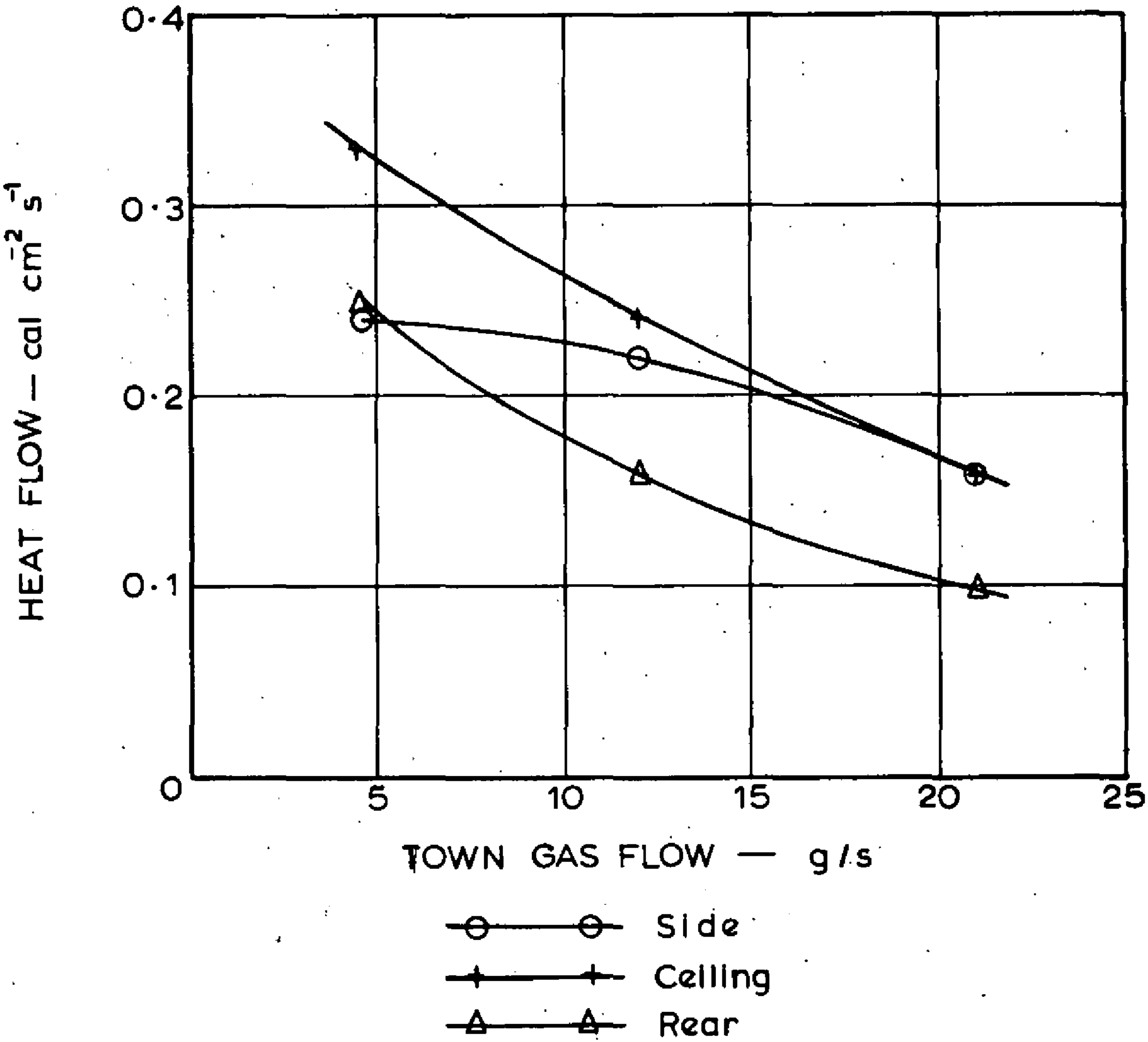

FIG.8. HEAT FLOW THROUGH WALL AND CEILING PLUGS

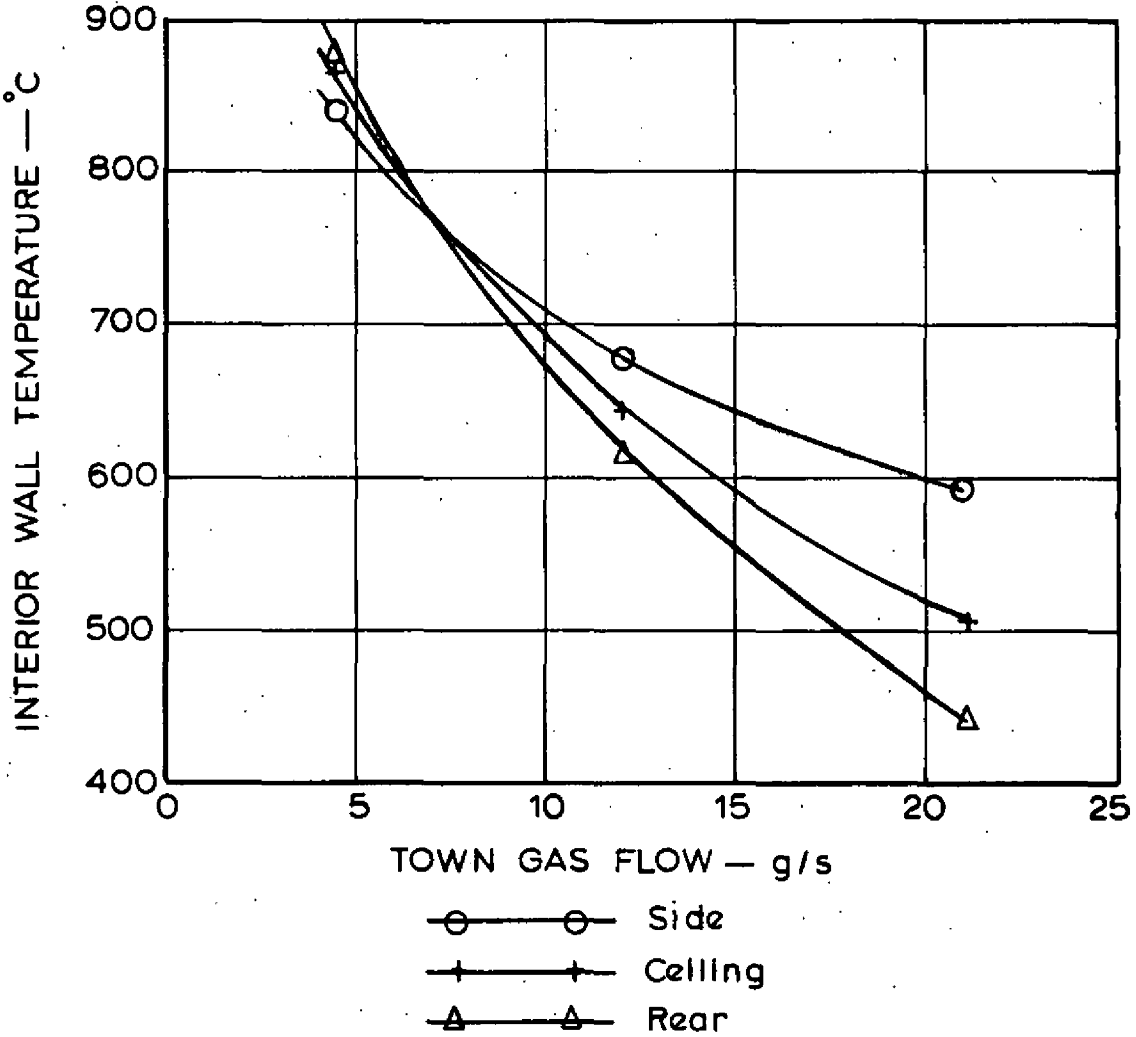

FIG. 9. INTERIOR WALL TEMPERATURES

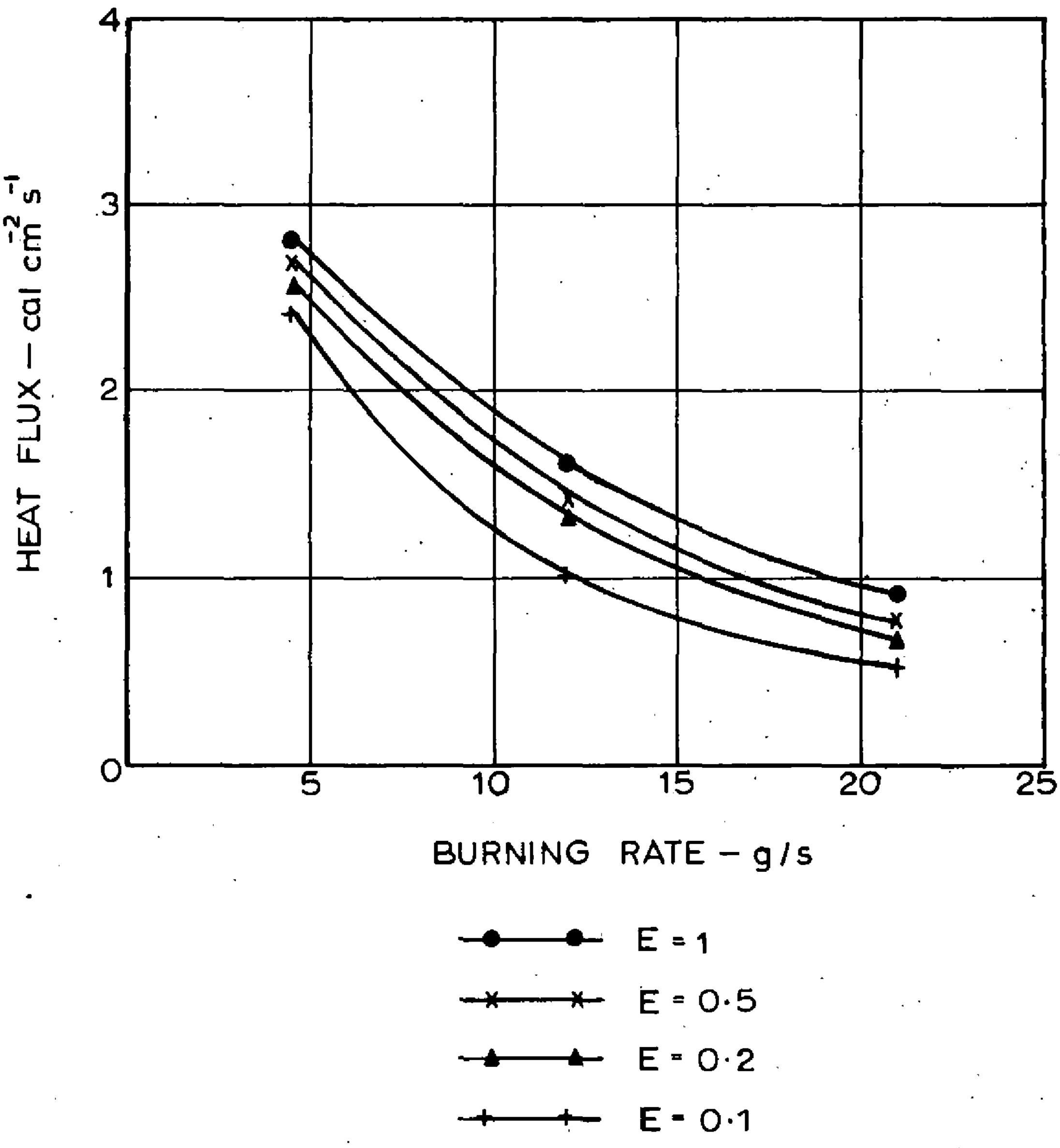

FIG.10. CALCULATED TOTAL HEAT TRANSFER TO A COLD HEAT-FLOW METER WITHIN THE COMPARTMENT (FROM TABLE 6)

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