F.R. Note No. 391
Research Programme Objective E1/9

DEPARTMENT OF SCIENTIFIC AND INDUSTRIAL RESEARCH AND FIRE OFFICES' COMMITTEE
JOINT FIRE RESEARCH ORGANIZATION

ROOF VENTING OF BURNING ENCLOSURES

PART II. THE CONSTRUCTION OF A MODEL AND SOME FLOW PATTERNS OBTAINED WITH IT

by

D. L. Simms, P. L. Hinkley and Alison Bisset

Summary

The construction of a small scale model suitable for studying the flow of heat from a convective source in a building is described.

Experiments show that the conduction loss has been made small enough for it not to affect the scaling laws derived from consideration of natural convection.

F1000/12/346
April 1959

Fire Research Station,
Boreham Wood,
Herts.

Notation and units used in this series

Symbol	Meaning	Dimensions		Conversion factor used F.P.S units to C.G.S units
		F.P.S. units	C.G.S units	
C_p	specific heat at constant pressure	B.t.u lb^{-1} $^oF^{-1}$	cal gm^{-1} $^oC^{-1}$	1
g	acceleration due to gravity	ft/s^2	cm/s^2	30
h	height of opening	ft	cm.	30
H	height of model	ft	cm.	30
K	thermal conductivity	B.t.u. $s^{-1} ft^{-1}$ $^oF^{-1}$	cal $s^{-1} cm^{-1}$ $^oC^{-1}$	15
L	characteristic height dimension	ft	cm.	30
l	path length of air in model	ft	cm.	30
Q	heat input	B.t.u/s	cal/s	252
Q_o	heat output			
T	absolute temperature	^oR	^oK	0.555
T_o	" ambient "	^oR	^oK	0.555
t	time taken for a small volume of gas to cover corresponding paths in prototype and model	s	s	-
V	characteristic velocity	ft/s	cm/s	30
v	velocity of gas at any given point in prototype and model			
w	width of inlet	ft	cm	30
y	distance measured vertically	ft	cm	30
y_o	distance measured from bottom of floor	ft	cm	30
θ	temperature above ambient at any given point in prototype and model	^oF	^oC	0.555
ρ	density	lb/ft^3	gm/cm^3	0.016

F.R. Note No. 391
Research Programme
Objective E1/9

ROOF VENTING OF BURNING ENCLOSURES

PART II. THE CONSTRUCTION OF A MODEL AND SOME FLOW PATTERNS OBTAINED WITH IT

by

D. L. Simms, P. L. Hinkley and Alison Bisset

1. [illegible]

1. Introduction

To study all the factors involved in the venting of fires on full scale would be prohibitive both in time and money, making it essential for model techniques to be used. The scaling laws for turbulent convection flow are well established so that models can be used to study the flow of convected heat. The model used in the present experiments represents on a $^1/12$ scale one bay of a factory having a flat roof (fig. 1), divided from an adjoining bay by a fire curtain.

2. Conditions for similarity

The flow pattern is determined by the shape of the enclosure and the strength and position of the heat source. The model is large enough for the flow to be turbulent so that the effect of viscous forces on the main stream may be neglected. If heat losses by radiation and conduction are small and skin friction is not significant, the relations between the temperature and velocity at any point for different heat inputs and different scales may be obtained.

By conservation of heat, neglecting conduction and radiation loss

$$Q \propto \rho v \theta L^2 \tag{1}$$

or

$$Q \propto \frac{v\theta}{T} L^2 \tag{2}$$

From simple buoyancy considerations

$$v \propto \left(\frac{L\theta}{T_0}\right)^{\frac{1}{2}} \tag{3}$$

and hence from (2)

$$\theta \propto \left(\frac{Q^2 T^2}{L^5}\right)^{\frac{1}{3}} \tag{4}$$

$$v \propto \left(\frac{Q T}{L}\right)^{\frac{1}{3}} \tag{5}$$

If equations (4) and (5) are satisfied for a given value of L, then the results can be extrapolated to larger scales.

3. Construction of model

3.1. Requirements

The model has to be well insulated to reduce heat loss by conduction, to be well sealed to prevent the escape of hot gases and to have a small heat capacity so that the wall temperature does not affect the air temperature when a change in operating conditions such as venting takes place. It must withstand an interior temperature of at least 570°F (300°C) without damage.

3.2. Design of model and instrumentation

Diagrams of the model are shown in figs. 2 and 3. The construction of the roof allowed any gas explosion in the model to be safely vented (1). The front could be partially closed by a curtain (fig. 4). Heat losses by leaks were reduced to a minimum by sealing the box with aluminium foil stuck by silicate paint.

The heat source consisted of 21 fantail burners giving blue diffusion flames with little radiation, arranged in a line 2 in. above the floor (fig. 5). The position of the heater could be varied. The gas supply was measured by a gas meter and could be controlled by a needle valve.

The temperatures at various points were measured by thermocouples made of 44 S.W.G. chromel-alumel spot welded and silver soldered on to 26 S.W.G. chromel-alumel supports.

The air velocity was measured by a radiation-compensated hot wire anemometer (2). The instrument is accurate to within 5 per cent at ambient temperatures of up to 120°F (50°C) greater than the ambient temperature at which it was calibrated (3).

4. Experimental procedure and results

4.1. Experimental method

The arrangement of thermocouples is shown in fig. 6. An anemometer and a thermocouple could be held in any position in the opening. The gas flow was set and the model allowed to reach equilibrium. Velocity and temperature profiles along the centre of the vertical plane of the opening and the temperature distribution within the enclosure were determined. In one experiment the temperature and velocity distributions across the width of the opening were measured.

In several experiments the vent was removed and the time taken for the new equilibrium to be established was noted.

4.2. Temperatures and Air Velocities

Nine sets of experiments were carried out, all with the heater at the end of the model away from the inlet, comprising three heat inputs at each of three sizes of inlet with no vent.

The temperature and velocity distributions across the opening were found to be independent of the position except within 2 inches (5 cm) of either wall, so that the model may be regarded as being representative of one dimensional flow over a very wide section.

The results for the temperatures and air velocities at different heat inputs may be compared by using equations (4) and (5). Plotting

$^{\theta}/(QT)^{2/3}$ and $^{v}/(QT)^{1/3}$ as reduced temperatures and velocities respectively

enables the data for various heat inputs to be correlated. Thus in fig. 7 the temperature profiles for the different heat outputs are reduced to one line and the shape dependent only upon the size of the opening and its relation to the height of the box: Similarly, fig. 8 shows the velocity profile for the three sizes of opening and Table 1 gives the temperatures at various points in the enclosure. The results show that equation (4) and (5) correctly predict the effect of changes in heat output.

4.3. Convective heat output

The measured temperature and velocity profiles can be used to calculate the convective heat output (Table II) and compared with the actual calorific value of the fuel.

The specific heat of the flue gases was estimated from the composition of the town gas and the air/fuel ratio assuming complete combustion. A correction was made for the latent heat of water vapour resulting from the combustion of the gas (Table II).

Table II

Opening height		Heat calculated from calorific value of fuel		Heat measured in convective flow	
in.	(cm)	B.T.U/s	(cal/s)	B.T.U/s	(cal/s)
5	(12.7)	0.87	(220)	0.91	(230)
		1.83	(460)	1.43	(360)
		2.5	(630)	2.38	(600)
9	(23)	0.48	(120)	0.36	(90)
		0.95	(240)	0.71	(180)
		2.06	(520)	1.37	(345)
16	(41)	0.44	(110)	0.36	(90)
		0.95	(240)	1.11	(280)
		2.1	(530)	2.62	(660)

The agreement between the calculated output and the measured heat input is satisfactory; the source of the largest errors is the velocity measurements which are not accurate at high velocities. Heat losses by conduction and radiation are therefore unimportant so that the insulation of the model is adequate and scaling on the basis of equation (4) and (5) is valid.

4.4. Time constants

When a vent was opened about 90 per cent of the total change of temperatures and velocities occurred almost immediately, the remaining 10 per cent of the change took many minutes.

Because the convective transfer takes place with a time constant of seconds given by $\left(l^2/gH\right)^{\frac{1}{2}}$, whilst conduction through the walls has a very long time constant, the relative magnitudes of the effects implies that the heat loss to the walls is about 10 per cent. This is satisfactory confirmation of the relative size of the conduction heat loss suggested by section 4.3, and, indeed, it is the simplest means of assessing whether conduction to the walls is significant in any model of convective heat flow.

5. Discussion and conclusions

The temperature and velocity at a given point has been shown to follow simple scaling laws derived from a consideration of natural turbulent convection. and calculation has confirmed that the radiation and conduction loss is small. The model can therefore be used to predict the mean temperature and velocity pattern in a larger enclosure.

These experiments to test the construction of the model have been for the worst case. With the smaller openings there was more scatter in the data than for the larger openings, because presumably the lower velocity within the enclosure led to some damping of the turbulence. Opening a vent would increase the bulk flow and this damping would then be of lower extent.

6. References

1. Top reliefs in box ovens. Explosion reliefs for industrial drying ovens. Part I. Industrial Gas Development Committee Report No. 612/56. Oct. 56.
2. BIGMORE, R. H. A radiation compensated anemometer. Joint Fire Research Organization, F. R Note No. 297/1957.
3. COWDREY, C. F. Temperature and pressure corrections to be applied to the shielded hot wire anemometer at speeds for which natural convective cooling is negligible. J. App. Phys. 1958, 9 (3) 113-6.
4. HIRD, D and FISCHL, C. F. Fire Hazards of Internal Linings. Department of Scientific and Industrial Research, National Building Studies Special Report No. 22. London, 1954. Her Majesty's Stationery Office.

TABLE I. REDUCED TEMPERATURES INSIDE MODEL

(for positions see Fig. 6)

THERMOCOUPLE			1	2	3	4	5,6,7	8	9	10	11	12
Height of opening in.		Heat Input B.T.U/s	Reduced Temperatures $\theta/(QF)^{2/3}$ $^{\circ}R^{\frac{1}{3}}$ $s^{2/3}$ $B.t.u.^{-2/3}$									
16	36	2.1	0.88	1.3	0	0	1.7	3.1	0	0	0	0
	37	0.95	1.0	1.4	0.08	0	1.8	3.7	0	0	0	0
	38	0.44	0.98	1.4	0.08	0	1.9	2.8	0	0	0	0
		mean of readings	0.96	1.4	0.08	0	1.8	3.2	0	0	0	0
9	35	2.06	2.3	2.3	2.0	1.7	2.7	1.6	0	0	0.08	0
	32b	0.95	2.1	2.5	2.2	1.7	2.6	1.0	0	0	0	0
	33	0.48	2.7	2.8	2.4	1.8	3.2	1.3	0	0	0.16	0
		mean of readings	2.4	2.5	2.2	1.7	2.9	1.3	0	0	0.12	0
5	17	2.5	2.5	-	2.2	2.2	2.9	-	0.16	0.59	0.70	0.42
	18	1.83	2.7	-	2.6	2.5	3.1	2.2	0.33	0.58	0.48	0.50
	31	0.87	3.7	3.8	3.3	3.2	4.2	2.9	0	0	0	0
		mean of readings	3.0	3.8	2.7	2.6	3.4	2.6	0.25	0.59	0.59	0.46

Key 0 No reading (in air inflow)

\- No thermocouple

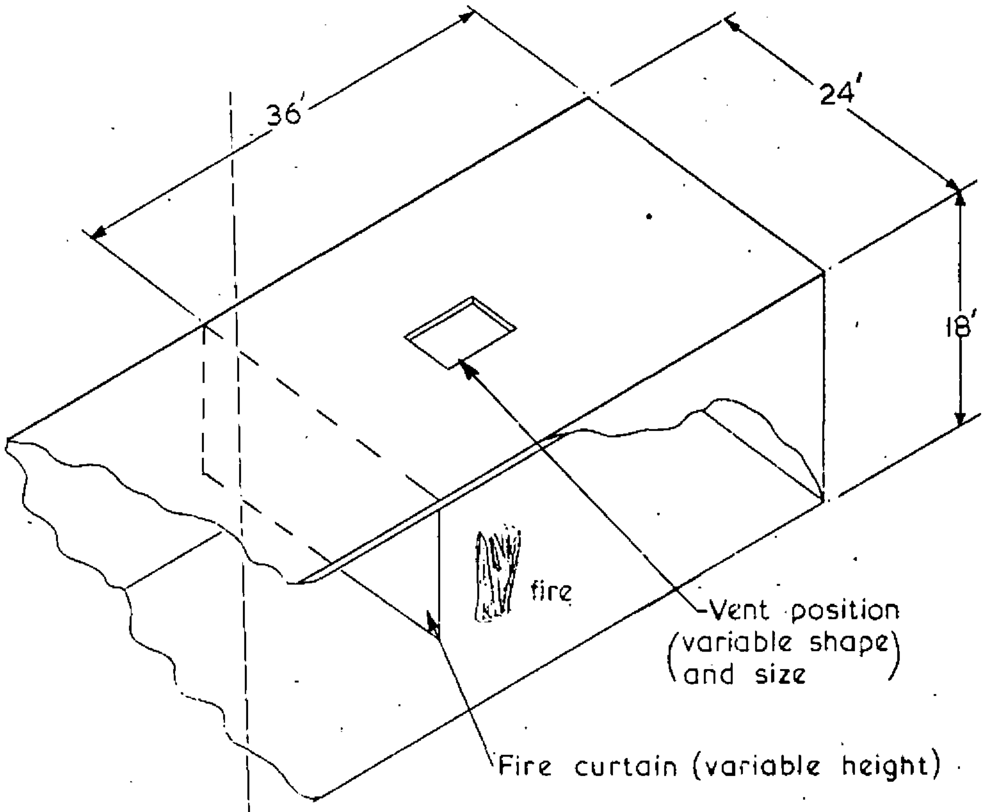

FIG. 1. PROTOTYPE

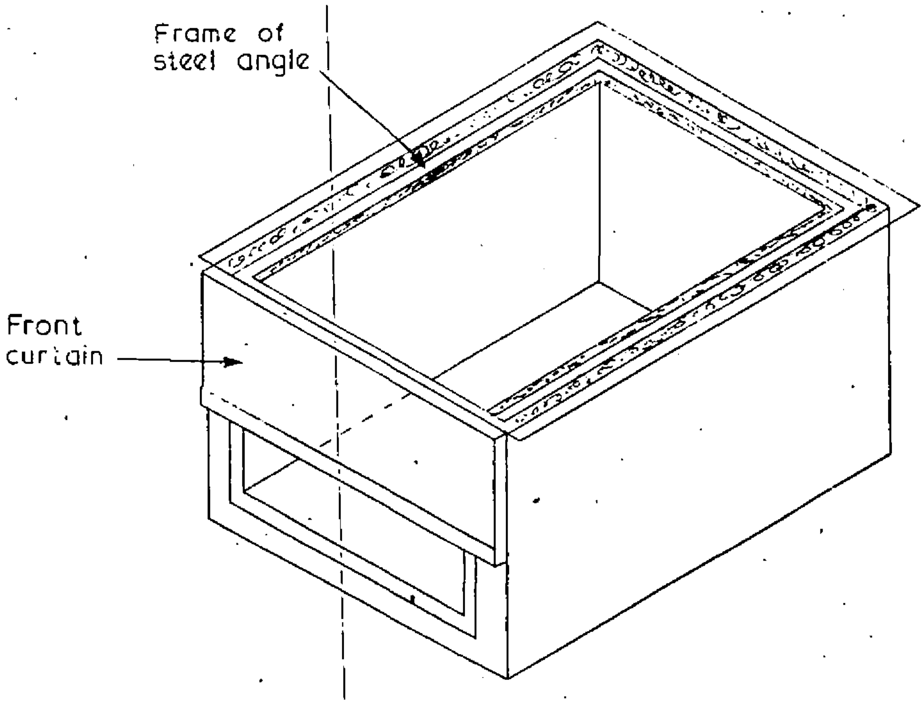

FIG. 2. ISOMETRIC VIEW OF MODEL
WITHOUT ROOF

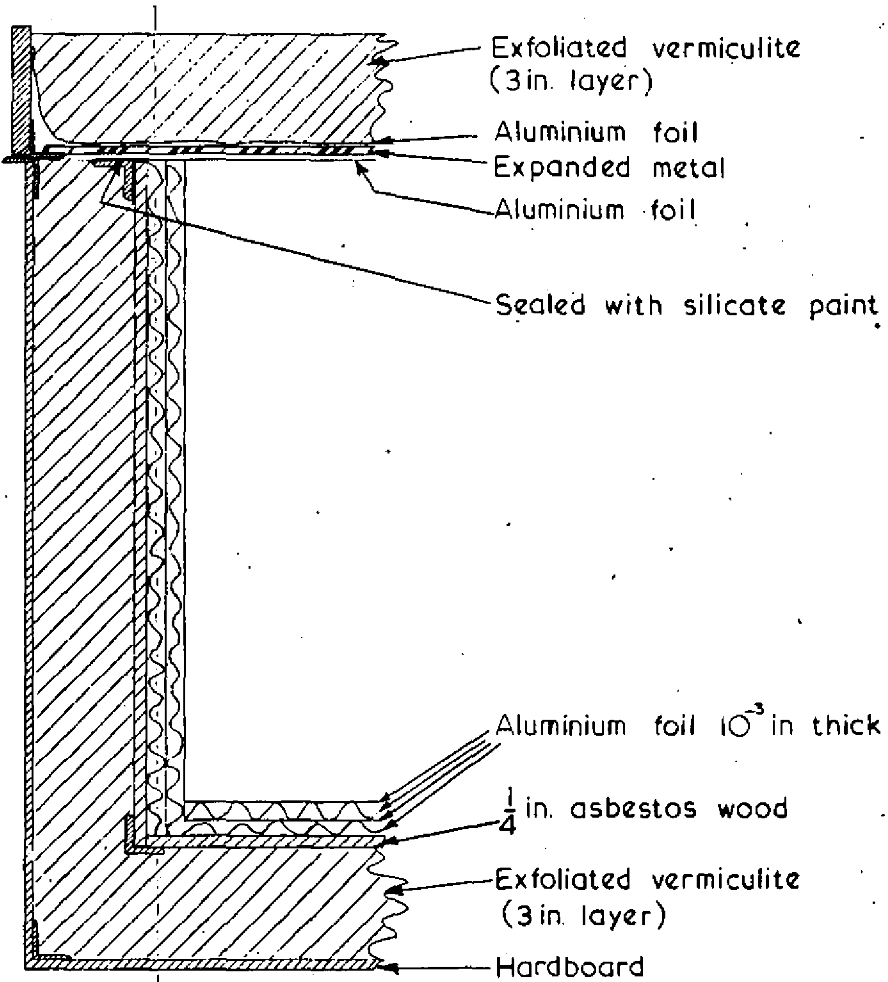

FIG. 3. SECTION THROUGH MODEL

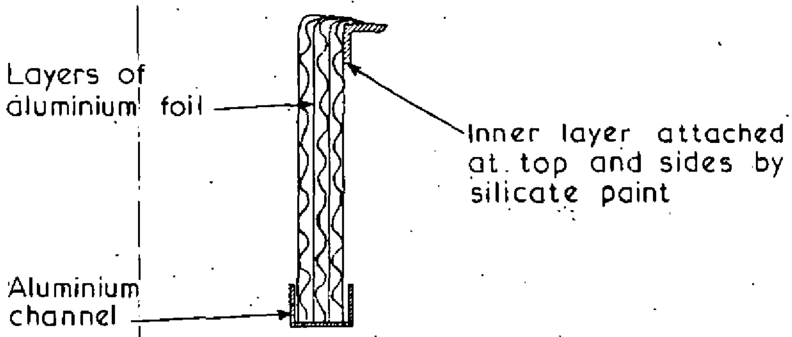

FIG. 4. SECTION THROUGH FRONT CURTAIN

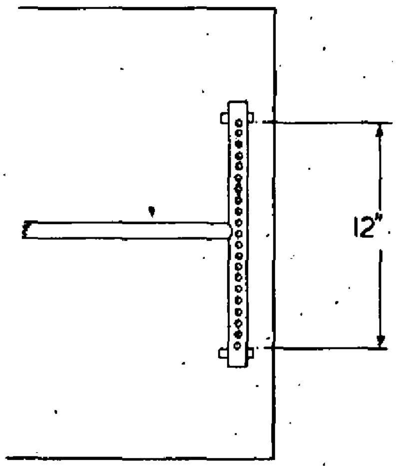

FIG. 5. HEATER

1/3209 F.R. 391

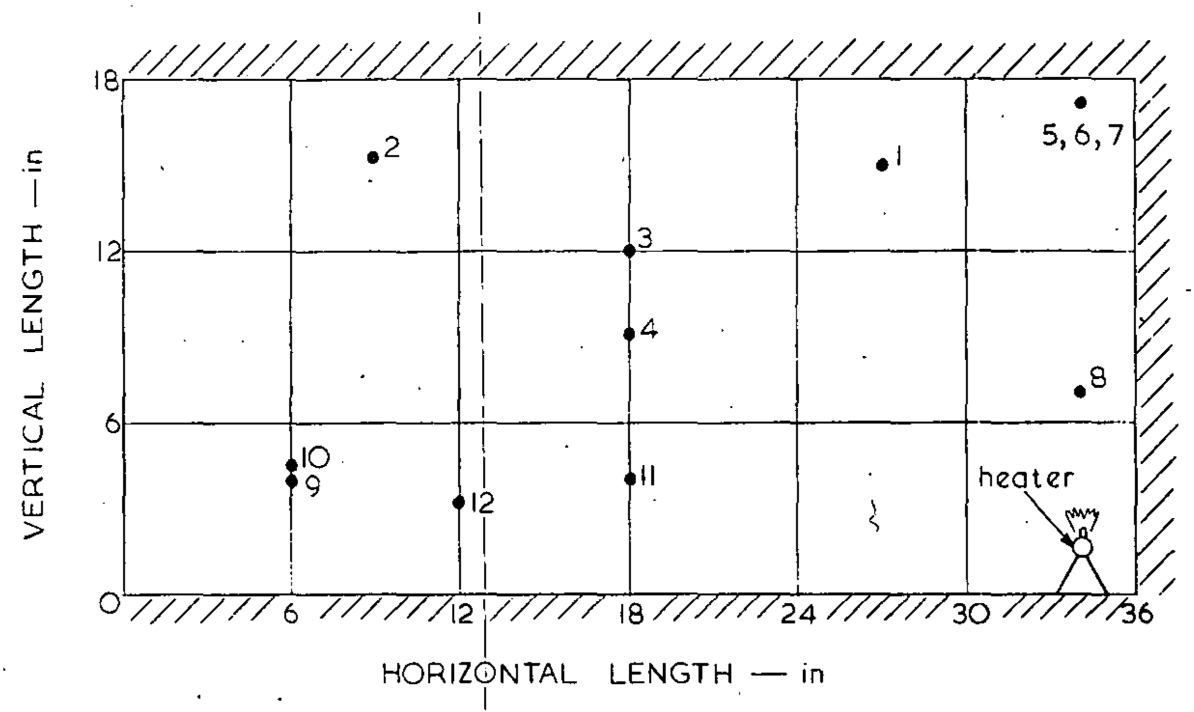

FIG. 6A. POSITION OF THERMOCOUPLES (ELEVATION)

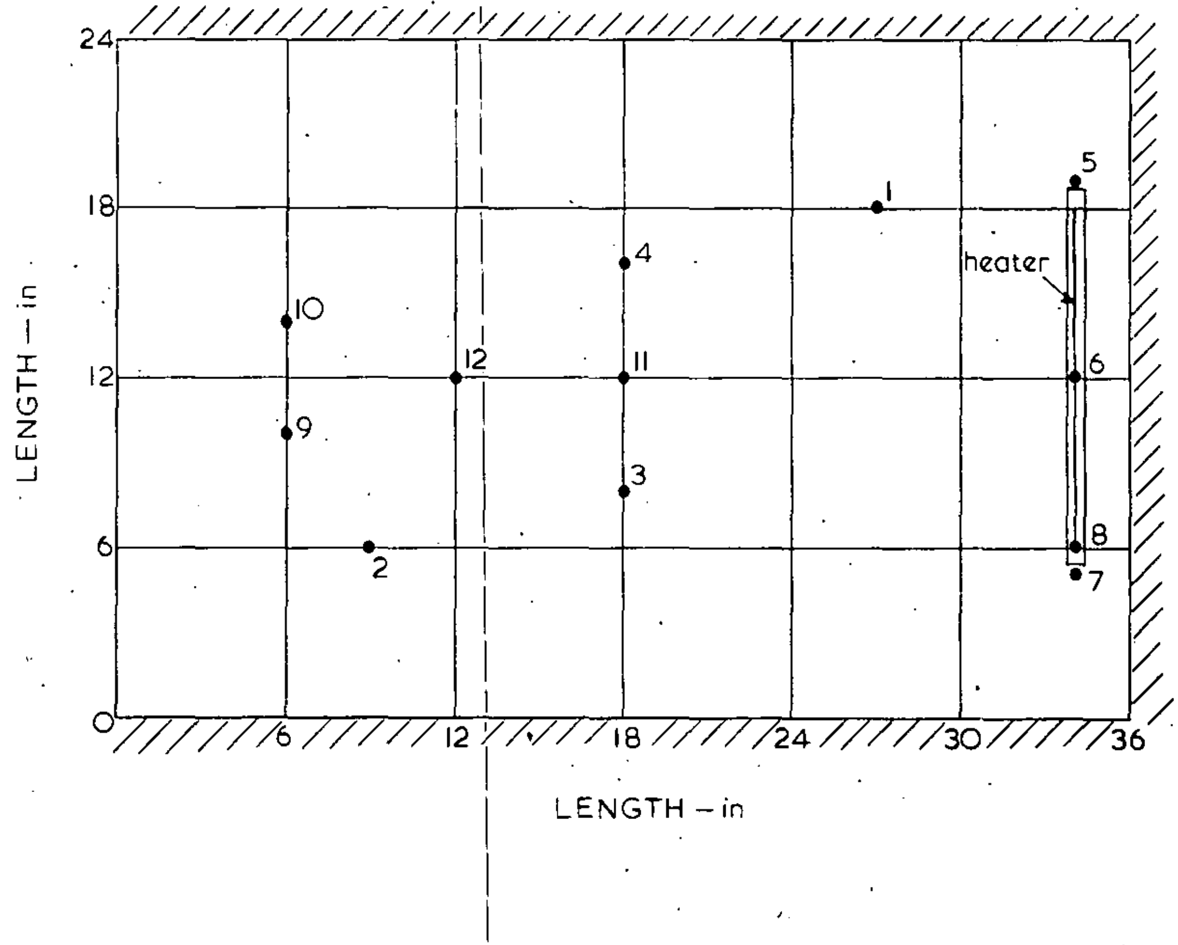

FIG. 6B. POSITION OF THERMOCOUPLES (PLAN)

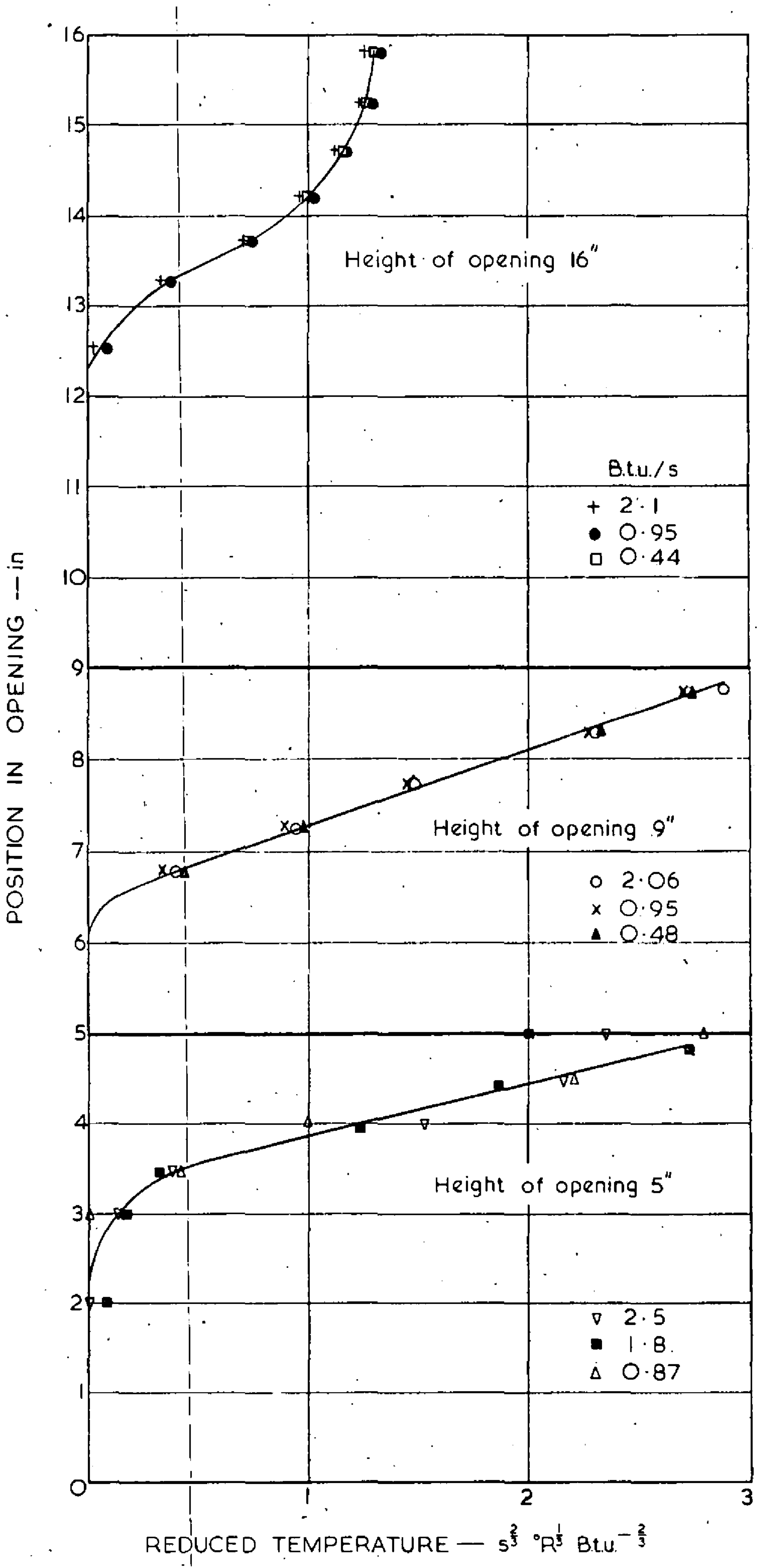

FIG.7 REDUCED TEMPERATURE PROFILES

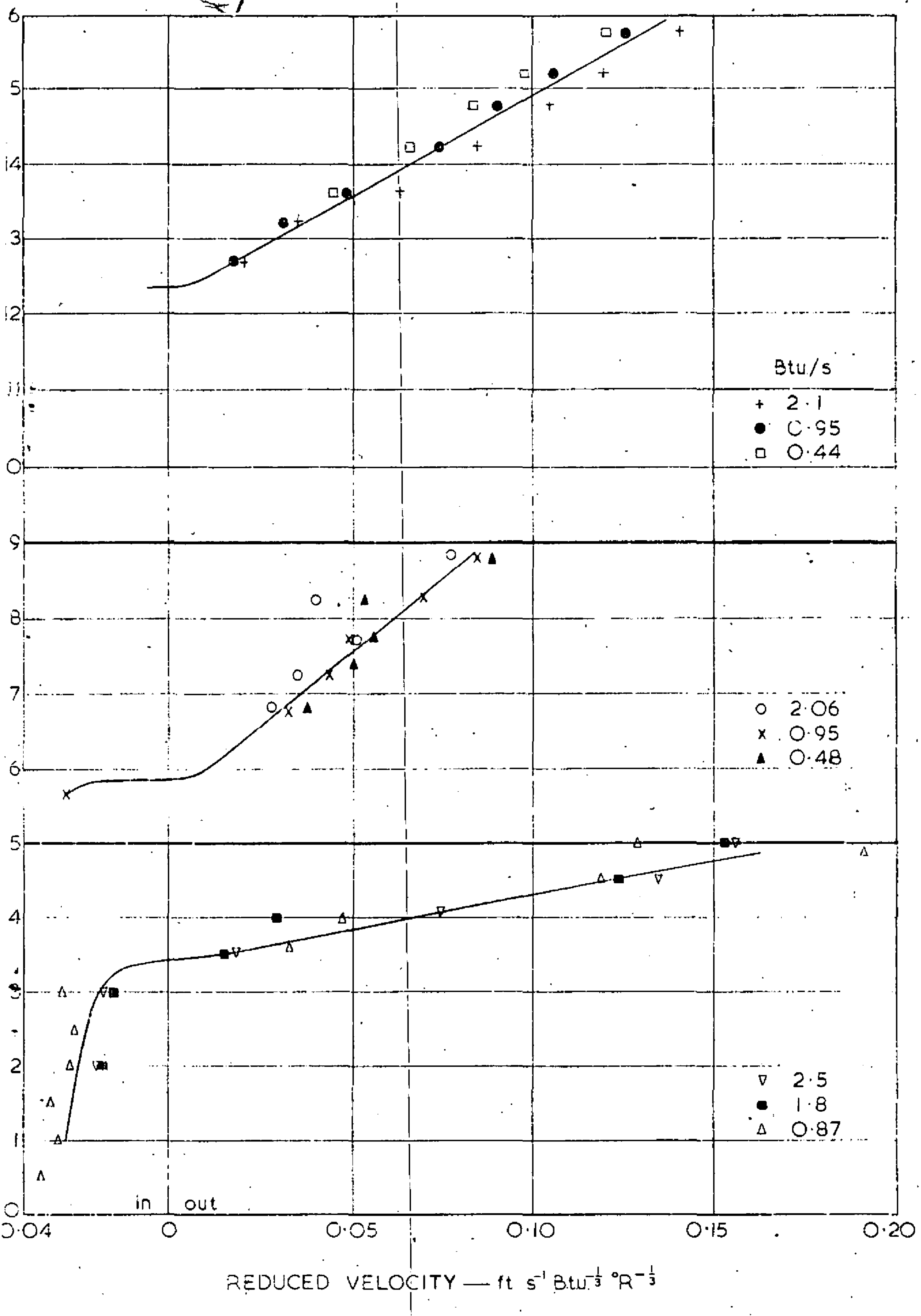

FIG. 8. REDUCED VELOCITY PROFILES