THE CONTRIBUTION OF CONVEYED COAL TO MINE HEAT PROBLEMS

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ABSTRACT

As coal mines get deeper, more mechanised and more productive the heat load on the ventilation system increases. In certain cases to the point where serious environmental problems may arise. To continue mining in these demanding conditions the sources of heat must be identified and evaluated so ameliorative measures may be taken. Due to the trend towards mining at greater rates and further from the shaft, mined coal on the conveyors is being recognised as a heat source of growing importance. This thesis describes its investigation.

Reviews of heat sources, psychrometry, heat stress indices and heat transfer are included to provide a background framework. The evaluation of the heat released by conveyed coal itself consists of theoretical treatment and laboratory investigations of heat transfer through broken coal. A model conveyor and its instrumentation constructed in a duct are described along with underground measurements at mines. The information obtained from theoretical, laboratory and on site investigations is analysed and summarised to provide a basis for future prediction.

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NOTATION

A	Area	m^2
a	Psychrometric constant	$^oC^{-1}$
a	Diffusivity	m^2/s
C	Specific heat	kJ/kg K
D, d	Diameter	m
e	Partial pressure of water vapour	kPa
F_{ev}	Emissivity and view factor	-
g	Gravitational acceleration	m/s^2
h	Heat transfer coefficient	$W/m^{2o}C$
k	Thermal conductivity	W/m^oC
L	Latent heat of evaporation of water	kJ/kg
ℓ	Length	m
M, m	Mass	kg
$\dot{M}$, $\dot{m}$	Mass flow rate	kg/s
P, p	Pressure	kPa
Q	Heat flux	kW
q	Heat flux per unit area	kW/m^2
R	Gas constant	kJ/kg K
R_e	Reynolds number	-
r	Radius	m
S	Sigma heat	kJ/kg K
T	Temperature absolute	K
t	Temperature celsius	^oC
U	Velocity	m/s
V	Volume	m^3/kg
X	Moisture content of air	g/kg
x	Distance along x axis	m

ε	Emissivity	-
ρ	Density	kg/m^3
ϕ	Relative humidity	%
μ	Dynamic viscosity	kg/m s

Subscripts

a	Air
av	Average
b	Belt
c	Convective
db	Dry bulb
e	Evaporative
k	Conductive
ℓ	Lower surface
P	Constant pressure
r	Radiative
s	Saturated
sp	Specific
u	Upper surface
wb	Wet bulb

CHAPTER 1

RESEARCH OBJECTIVES

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1.1 INTRODUCTION

Coal will for the forseeable future provide a major source of energy for Britain and the world. It is relatively plentiful, environmentally safe, and available from many politically and economically stable areas. The contribution of coal as an energy source and its distribution world wide are shown in figures 1.1a and 1.1b [1].

In Britain the National Coal Board's 'Plan for Coal' [2] and later 'Plan 2000' reflect the confidence of the NCB in the future of the coal industry. Estimates for the year 2000 put UK coal production at 150-200 million tonnes per year. Comparison with 1980 production of 109 million tonnes indicates, at least, an additional 40 million tonnes of new and replacement capacity. To meet future requirements the trends are towards mining coal at greater depths, rates and distances from surface connections. Workings are becoming more concentrated with both more and larger machines installed. Many of these characteristics of modern high productivity mining are unfortunately those which produce and aggravate underground climatic problems. Difficulties are already being encountered at some collieries.

The main environmental problems will be caused by high temperatures which adversely affect the health, safety and performance of men. The ultimate aim of mine ventilation and air conditioning is to provide a climatic environment suitable for men in all under-

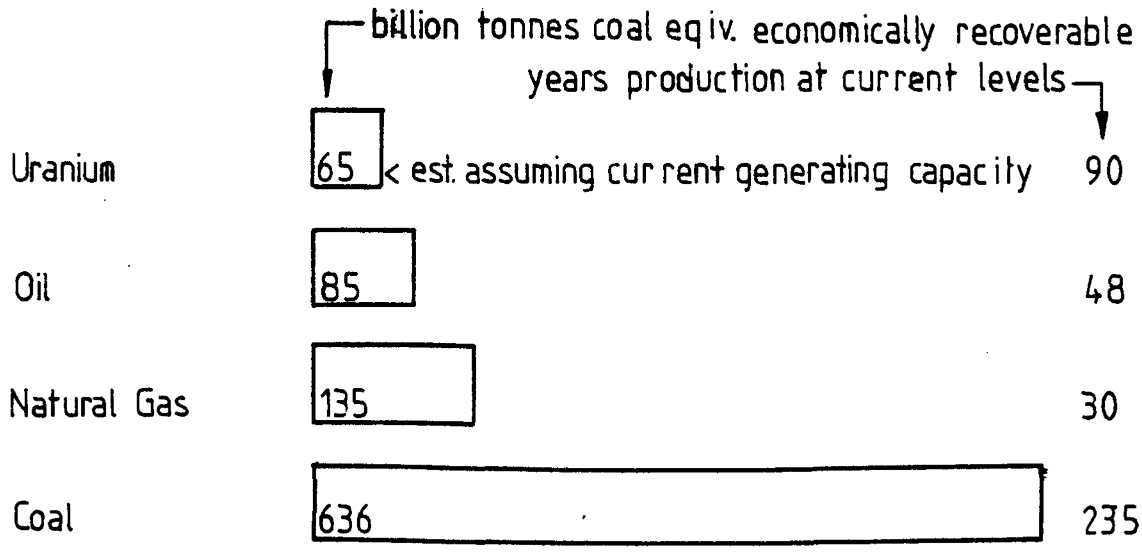

FIG 1.1a WORLD ENERGY RESOURCES

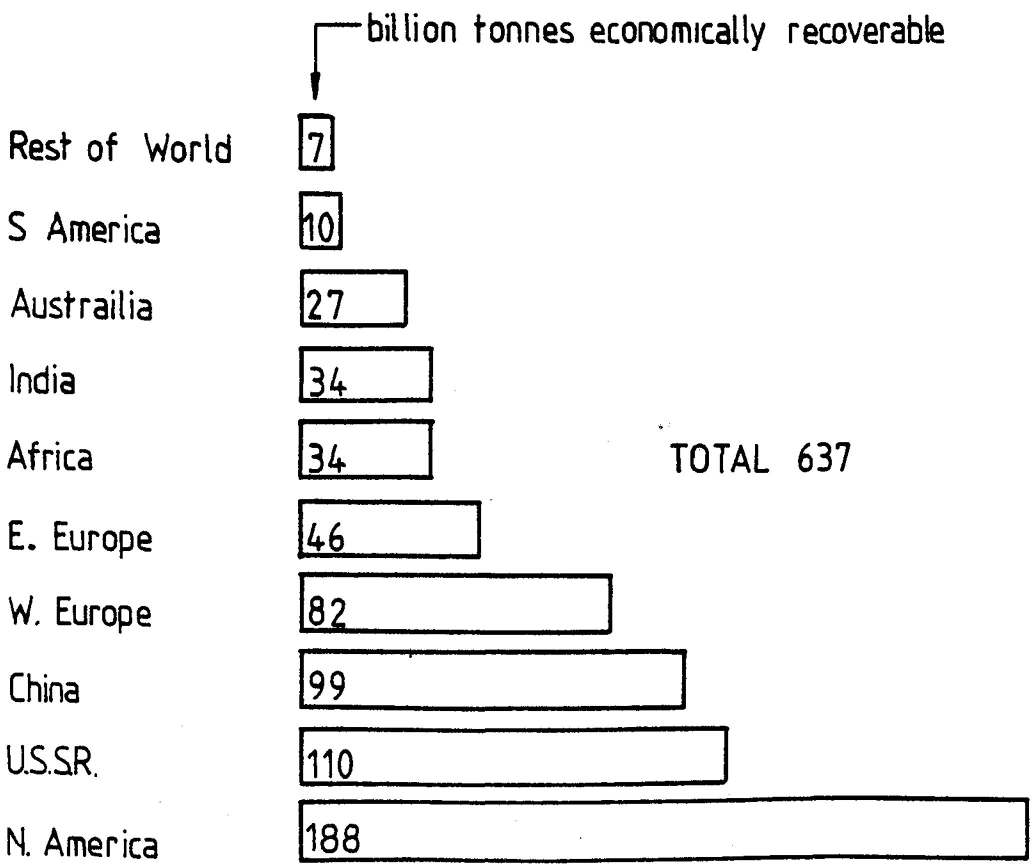

FIG 1.1b WORLD COAL DISTRIBUTION

ground work places with due consideration to safety and cost. It is therefore important to identify and evaluate contributing causes of a heat problem so that ameliorative measures may be taken. With regard to the solutions of heat problems these may range from simple measures to increase air velocity to, in extreme cases, the installation of a large scale air conditioning plant. The following section briefly describes the history of measures used to relieve the heat problems as mines have become progressively deeper, legislation tighter and workers expectations higher.

1.2 HISTORY OF UNDERGROUND HEAT PROBLEMS

The first trials of underground air cooling plant in British collieries appear to have been carried out as early as 1923 when researchers at the University of Birmingham [3] conducted tests at Holly Bank and Pendleton collieries in Lancashire. Underground refrigeration units of only some 15 kW capacity utilising the carbon dioxide cycle were tested and although giving valuable results no further work was undertaken until 1952 when investigations described by Bromilow [4] were made at Snowdown colliery in Kent. In this case a small direct evaporating air cooler, utilising refrigerant R12 was used in a forcing ventilation system for a heading.

Since World War II air cooling equipment has been installed in some Belgian and West German collieries. Recent demand has resulted in the modernisation of the equipment and its use has become much more widespread. In response to increasingly hot conditions in the Parkgate seam at Bevercotes Colliery, North Nottinghamshire Area, a cooling unit of West German design was installed in 1975 [5]. This system comprised of a 150 kW direct evaporator unit in conjunction with a chilled water circuit feeding air coolers in two headings. Following successful operation larger systems have been used, the most recent being a 750 kW refrigeration unit. This provides a chilled water supply to a district to give chilled service water via a heat exchanger and also an air chiller. Systems used in the future in hotter mines will probably be similar but larger with a centralised plant providing chilled water for both service water and air cooling. This follows the recent developments in South Africa discussed later.

Localised heat problems in headings and face ends have resulted in the development and production of self contained spot coolers of 30 kW capacity. Ten of these units are in operation to date in British collieries and following satisfactory use, negotiations between the NCB and the manufacturers have resulted in the production of a 50 kW model.

The country with the greatest experience of heat problems is South Africa. In 1924 when some mines were at a depth of 1,700 m and virgin rock temperature (VRT) of 30°C+, the first fatal case of heat stroke occurred. By 1931 this figure had reached 92. This resulted in immediate and increasingly successful research into heat physiology and the recognition of the benefits of acclimatisation. Also air cooling systems were evaluated and introduced. Refrigeration commenced with the installation of surface air cooling plant. As to be expected these systems were relatively costly and inefficient, mainly due to the poor positional efficiency of surface plant. (Positional efficiency increases as the cooling is

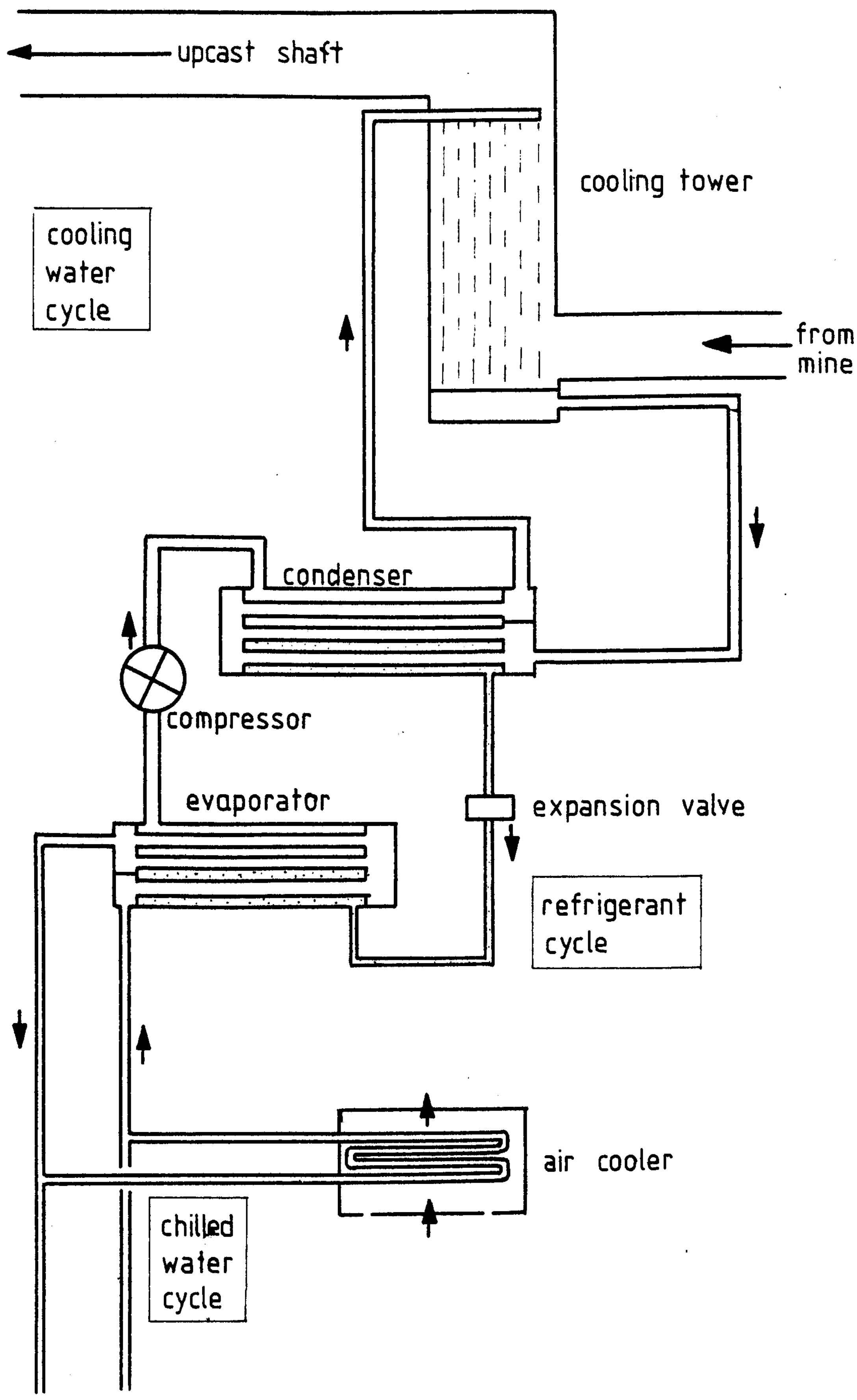

FIG. 1.2a CENTRALISED REFRIGERATION PLANT FOR LARGE MINE

applied nearer to the place where cooling is required.) Over the years better systems have been developed and refrigeration plants of up to 40 MW are now installed underground. A typical system comprises a centralised refrigeration plant rejecting heat via a cooling tower to the upcast shaft. The refrigeration unit supplies a chilled water circuit using insulated pipes around the mine with air coolers inserted where needed (fig. 1.2a).

A recent development in South Africa is particularly significant. The research workers at the Chamber of Mines of South Africa led by Whillier have found that chilling the service water for underground machinery is very effective. The cooling effect of using chilled water is principally due to the high thermal capacity of water. Positional efficiency is excellent due to cooling being applied at the point of work and the amount of water used being proportional to the work done. Also no costly water air heat exchangers are needed.

A large scale test was undertaken at Hartebeestfontein Gold Mine, South Africa, starting in 1975 [6] which used a chilled service water system and this has given excellent service to date. As a result the new Union Corporation 'Unisel' mine will have a chilled service water system as described by Howes and Green [7].

A similar system using a surface plant to chill water sent underground would be suitable for use in British collieries having extreme conditions. It is likely that this type of system will eventually be used.

1.3 EMPHASIS

This study is part of an integrated programme of research being carried out in the Department of Mining Engineering at the University of Nottingham to increase knowledge of mine environmental engineering. By concentrating on specific aspects it has been possible to build and improve an overall understanding of the problems and characteristics of the mine environment and hence make a useful contribution to industry by providing reliable predictions, solving existing problems and suggesting ameliorative measures, also assisting long term planning and the planning of new mines. The research is directed towards producing a suite of computer problems which may be used separately or in combinations to provide the services previously mentioned. In particular this project is a component in an attempt to evaluate the heat balance in the airflow around a coal mine.

With regard to relatively deep coal mines the main sources of heat until recently have unquestionably been accepted as strata heat and that from machinery. Strata heat was a problem even before the intensive mechanisation which has aggravated the situation. This problem is particularly acute on the face where up to 1 MW of installed power is becoming commonplace. Virtually all the electrical power supplied is degraded to heat in the immediate area. Since about 1975 though it has become apparent that the coal mined has itself contributed to the heat load, potentially on a large scale. Rather than the massive amounts of machinery installed on a face dissipating heat entirely directly to the air, a proportion of this goes to the coal during cutting which may already be at a high temperature.

On a highly mechanised face with a high rate of advance the coal output is high. It is highly fragmented coal, well wetted, and travels a long distance out of the mine usually against the incoming air. This situation is conducive to a very efficient transfer of heat from the coal as it cools from around strata temperature to surface ambient temperature on its journey outbye. Subsequently air travelling to the working areas of the mine, probably along long trunk conveyor roads will arrive at the face already well heated. In the future when it is hoped faces will frequently produce 1000 tonnes per shift the consequences of the conveyed coal heat source will be of even greater significance. The purpose of this study was to:-

(i) Find the potential magnitude of the heat from conveyed coal.

(ii) Identify and evaluate the heat in the coal.

(iii) Identify and evaluate variables and mechanisms affecting heat transfer from the coal.

(iv) Attempt to predict the amount of heat released from conveyed coal in a mine.

It is important to note that measurement and prediction of the scale and characteristics of sources of heat in a mine is complicated by the dynamic nature of the situation and the extremely complex interactions which take place, notably with the enormous strata heat sink.

1.4 LAYOUT

This thesis is arranged so that necessary background is included first. This is to allow a full understanding of the principles used in the research work and familiarise the reader with the research philosophy and reasoning. First is a review of heat sources found in mines. These are listed and discussed particularly with respect to their magnitude and interaction. Not all give problems in British collieries, but they illustrate the diversity of the problems encountered.

The most important effect of a poor environment is an adverse reaction of workers in it. Consequently human response to excess heat and heat stress indices are examined.

To facilitate understanding of the practical work a thorough knowledge of psychrometry, the study of heat in air and air temperature measurement is essential as is that of the various types of heat transfer which are also included.

The actual research described consists of a theoretical treatment followed by laboratory work to evaluate input parameters. Mainly the thermal characteristics of broken coal. Following this is a description of a fully instrumented model of a conveyor constructed in a duct where the air conditions could be controlled. The results of this are then examined and compared with data collected in underground investigations both before and after the laboratory work. The increased understanding of the mechanisms and magnitude of the heat transfer from conveyed coal is described together with some estimates for different situations.

CHAPTER 2

SOURCES OF HEAT IN MINES

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2.1 INTRODUCTION

Identification of all the sources from which heat enters mine ventilation air is essential before attempting any study of the complex subject of heat exchanges with mine air. It must be emphasised that the quantities of heat introduced into the air current from these sources may vary considerably from one mine, or situation to another. The degree of wetness of mine airways and working areas has a profound influence on heat flow and so a brief resumé of moisture sources appears at the end of the chapter.

2.2 AUTOCOMPRESSION

It is known that the temperature of air can change with change in pressure. The expansion of air performing work produces cooling, and compression heating. When air descends a mine shaft it is compressed at a rate of approximately 1200 N/m^2 per 100 m vertical travel under its own weight. The potential energy possessed by the air at the shaft top is converted to heat energy as the air descends. This is termed autocompression.

The increase in temperature to autocompression is calculated as follows:

$$\frac{\text{gravitational acceleration x vertical distance}}{\text{specific heat of air}}$$

For instance air descending a shaft 100 m deep will increase dry bulb temperature:-

$$\frac{9.81 \times 100}{1004} \simeq 1^{o}C \qquad (1)$$

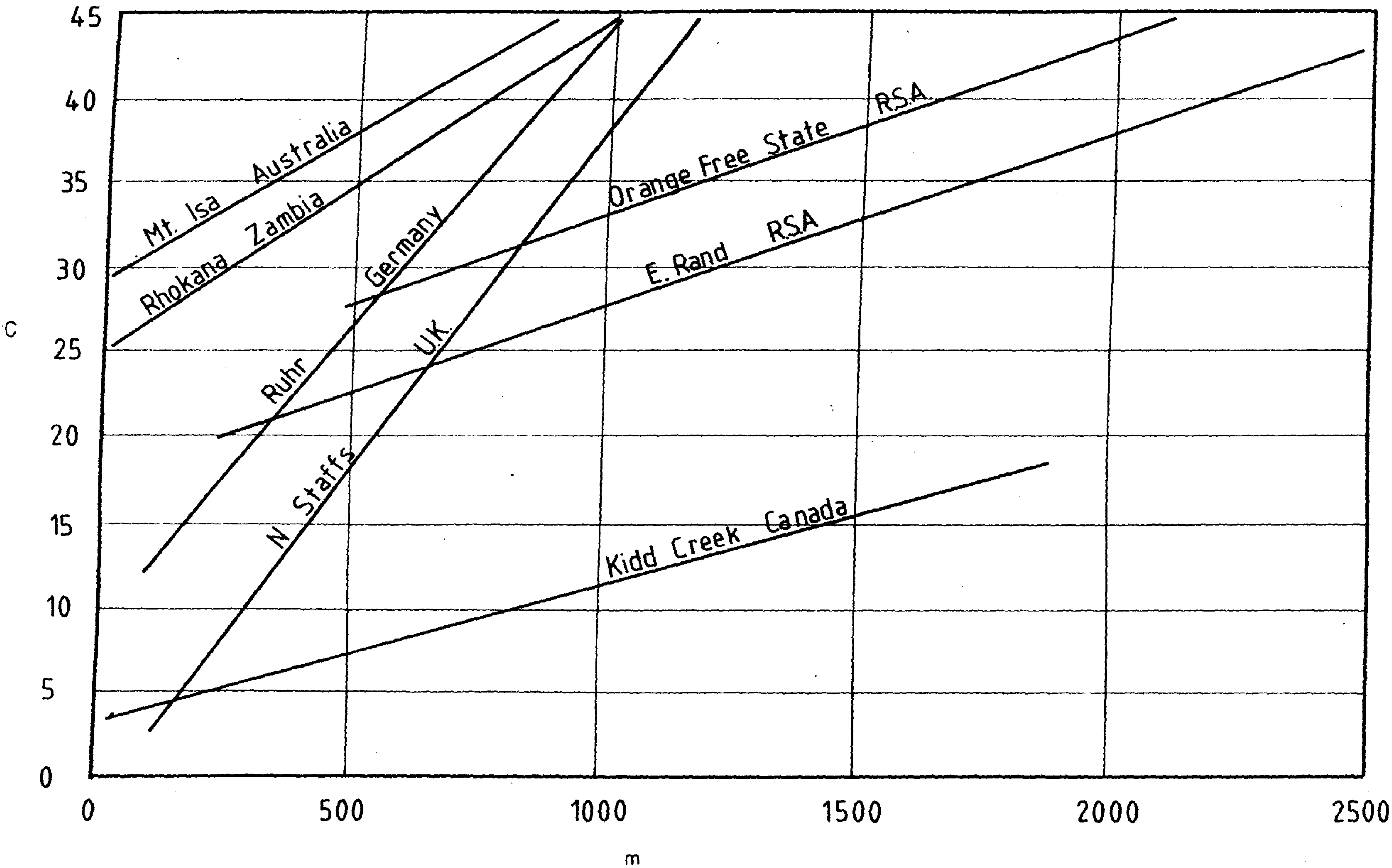
The increase in dry bulb temperature in a dry shaft is approximately $1^{o}C$ per 100 m. Evaporation from the walls of a wet shaft can reduce this to $0.4^{o}C$ per 100 m which corresponds to the wet bulb temperature increase (see Chapter 3, Psychrometry).

2.3 STRATA HEAT

The temperature of the rocks comprising the earths crust increases with increasing depth.

The temperature gradient varies according to the type of rock formations and so varies with location. It is inversely proportional to the thermal conductivity of the rock. Geothermal flow of heat from the hot core of the earth is about 0.05 W/m^2 and is practically constant over most of the earth's surface. The temperature of undisturbed strata at any point is known as the virgin rock temperature (VRT), or sometimes virgin strata temperature (VST). Figure 2.3a shows the geothermal gradients of some of the world's mining areas. It can be seen that coalfields have a steep geothermal gradient. This is caused by the carboniferous strata having a low thermal conductivity. The South African goldfields have, due to the high conductivity of the quartzite, a shallower geothermal gradient. In the latter case mines are able to penetrate much deeper before high VRTs are encountered. When rock is exposed by mining operations, providing the rock is at a higher temperature than the air, heat is liberated from the rock to be picked up by the ventilating air. The rate and amount of heat transferred to the air depends on several factors, some obvious,

FIG. 2.3a GEOTHERMAL GRADIENTS OF SOME OF THE WORLDS MINING AREAS



and some which will become more apparent after reading the chapters on psychrometry and heat transfer. These factors are:-

Within the rock

(i) Physical properties of the rock (thermal conductivity, specific heat and density).

(ii) Local geology.

(iii) Virgin rock temperature.

Across the rock air interface

(i) Temperature of the rock surface.

(ii) Length of time since exposure.

(iii) Wetness of rock surface.

(iv) Physical parameters of surface (roughness area, size and shape of roadway).

(v) Air flow rate and velocity across surface.

(vi) Psychrometric condition of air.

Heat from the broken rock in the goaf of a coal face is also considered as strata heat within the scope of this thesis.

2.4 HEAT FROM ROCK BROKEN IN MINING

The heat within broken rock has two components, the original strata heat and heat due to the breaking process whether caused by blasting or cutting. It is considered as separate to its component sources due to its magnitude and the fact that it does not reduce the ability of the strata to transfer heat.

Successful attempts have been made to theoretically assess the heat derived from broken rock in gold mines by Whillier and Van der Walt [8]. With regard to coal mines the situation is complicated by machine cutting and extensive conveyor systems. This subject is covered in more detail later in the thesis.

2.5 MACHINE HEAT

The usual sources of power underground are electricity, diesel and compressed air. Apart from the power used, the amount of heat generated varies according to the type of work done.

When work is done against gravity, pumping or hoisting, the work appears as increased potential energy of the material raised and does not appear as heat. In horizontal workings though all the energy supplied to machinery ultimately appears as heat.

2.5.1 Diesel machines

The heat produced by a diesel machine is equivalent to the calorific value of the fuel multiplied by the amount used. The most common users of diesel power are locomotives which generally do little work against gravity so most of the power used will appear as heat. In general though the amount of heat from a diesel locomotive is relatively small but in areas where ventilation is poor it could be significant.

2.5.2 Compressed air

Compressed air working by expansion does so at the expense of its own pressure energy which is reduced by an amount equal to the work done. If all of the work done by a compressed air motor is against friction the amount of heat produced is equal to the loss of heat due to the air expansion and the two cancel.

2.5.3 Electrical Power

The heat produced by electrical machinery is equal to the power supplied to the machine. This can easily be measured. The heat from the machine is dissipated as shown in figure 2.5a. The total rated power of machinery on a coal face may often exceed 1 MW and in the restricted environment of a face this could produce intolerable conditions. Fortunately utilisation is seldom more than 50% and few motors work at full power for more than a few minutes at a time. The heat from a machine may not be dissipated in strict relation to the power drawn as large machinery, say a shearer, has a lot of thermal inertia. It can store heat as it gets hot and release it later as it cools. The effects of storage are discussed later.

The last three mentioned heat sources, strata, broken rock, and machinery interact to a large extent on a coal face. The situation is best described graphically as figures 2.5b and c.

2.6 OTHER HEAT SOURCES

2.6.1 Oxidation

The heat produced by oxidation of wood, coal or other minerals is difficult to measure or predict as it is usually small. There are rare cases of mining in sulphide ores where large problems are caused.

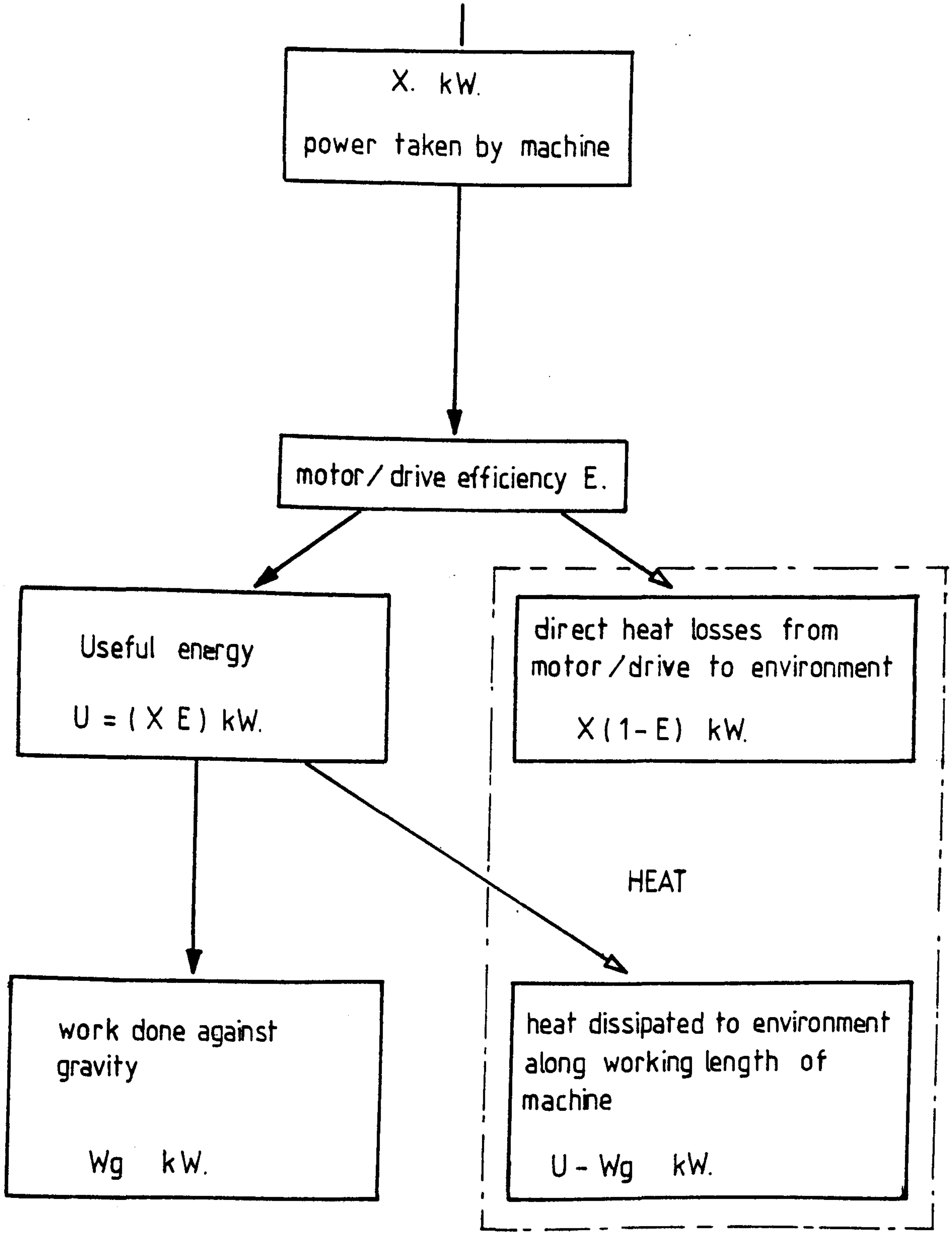


FIG 2.5a MACHINE HEAT / POWER DISTRIBUTION

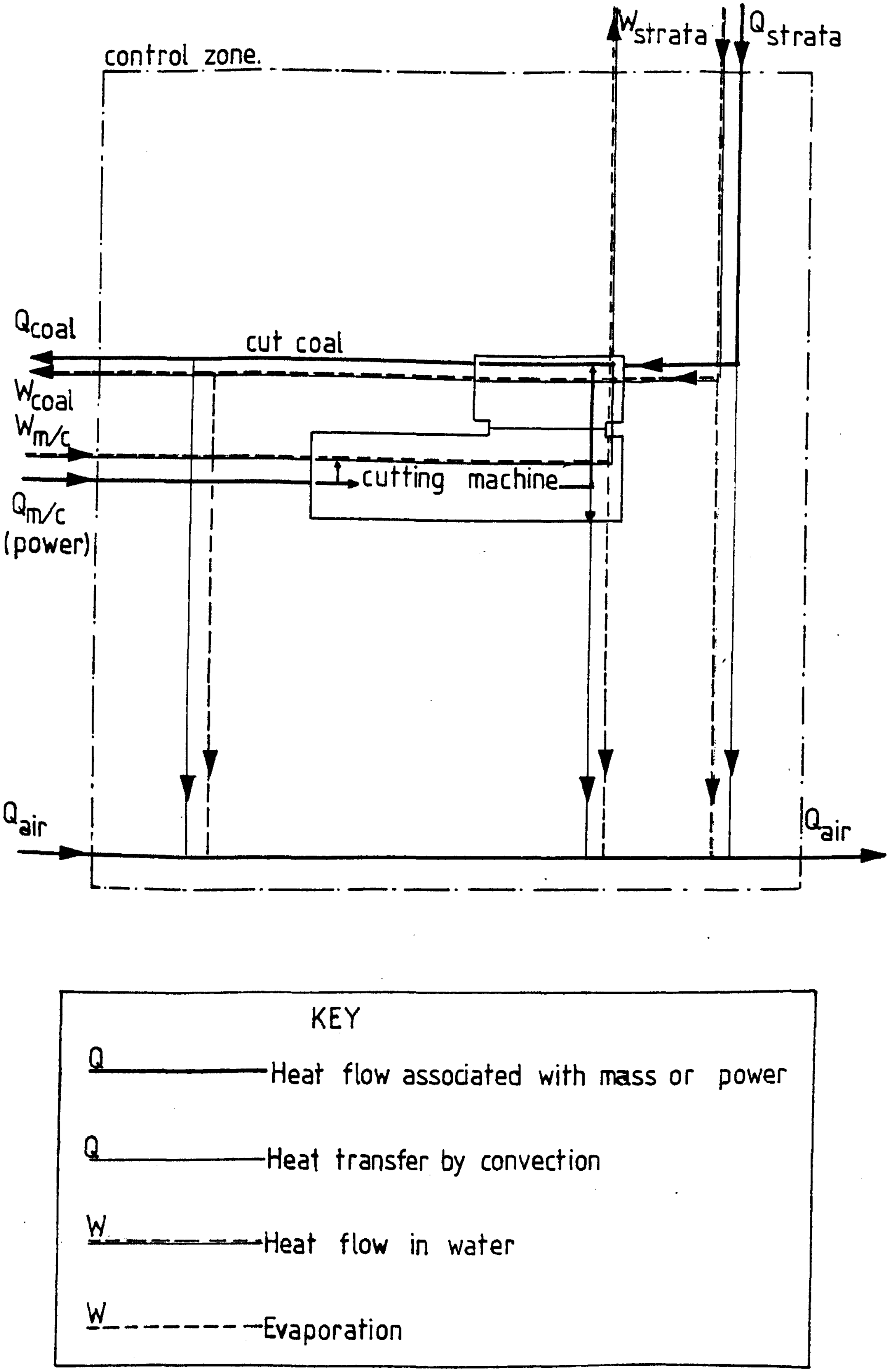


FIG 2.5b. HEAT FLOW PATHS AROUND COAL CUTTING MACHINE

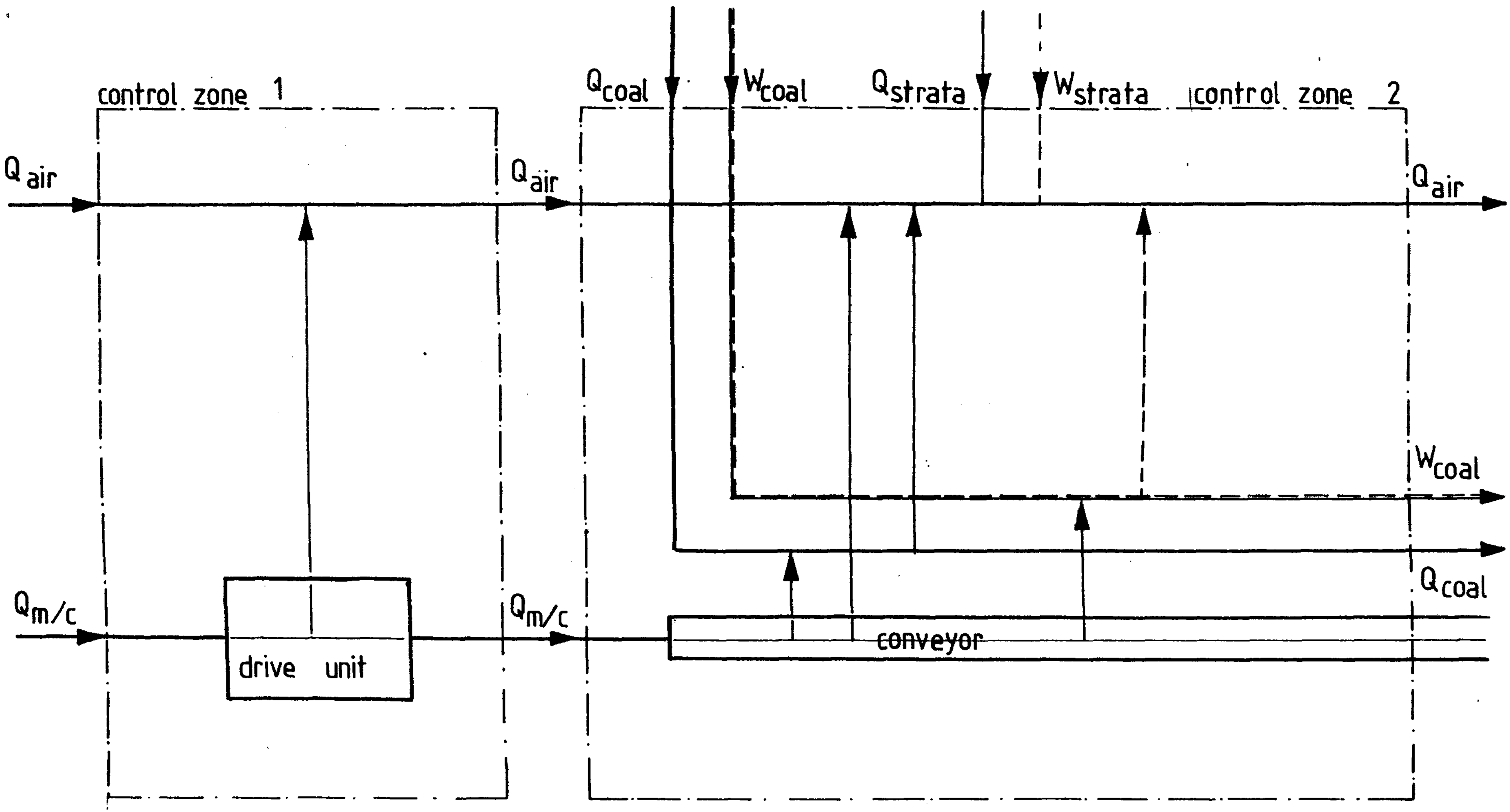


FIG 25c HEAT FLOW PATHS AROUND CONVEYOR SYSTEMS

2.6.2 Personnel

A man working hard can generate up to 500 W which is small compared with other sources. The only place body heat could be a problem is in a particularly labour intensive heading or stope.

2.6.3 Rock movement

Large amounts of potential energy are dissipated during caving of rock. The strata acts as a heat sink though so there is little effect on the ventilation air.

2.6.4 Explosives

Although explosives have the potential to release large amounts of heat, about 4 MJ/kg, no heat problems are caused. On mines where explosives are used in large quantities the workings are cleared during blasting.

2.6.5 Ambient climate

The condition of the air entering the mine naturally affects conditions underground. Not a problem in the UK.

2.7 STORAGE EFFECTS

If the heat emissions from all sources between two points on a ventilation circuit are added they should equal the heat gain by the air. Work or energy balance being achieved. Over recent years it has been detected in several cases, particularly districts in coal mines that balance is only attained over a period of time [9]. This is due to heat storage mainly in steelwork and the strata immediately around the airway. During peak production the heat made from all sources, but mainly machines escalates rapidly, raising the air temperature. Heat is transferred in turn to the steelwork over which the air passes. The high thermal conductivity and specific heat coupled with the vast quantities of steel on a district, possibly a few thousand tons, give a large storage potential. It should be noted that the rate of temperature change will determine the storage, and release, rates not the extent of the change.

The strata surrounding an airway stores heat on a similar principle, but the magnitudes of the variables are different. Conductivity is low, but the masses involved vast. There is an added complication of the temperature profiles round certain roadways changing to give a storage effect even if the heat flow is all to the airway but the net result is similar.

Thermal storage has an advantageous effect in that to a certain degree it reduces peaks in the overall heat make, but it has little effect in the very short term and close to a large spot source. A cutting machine for instance. With respect to this, ventilation should still be designed to cope with the worst possible conditions until more is known about thermal storage to allow confident accurate predictions.

2.8 SOURCES OF MOISTURE

To understand the various processes that occur in a mine airstream, humidity must be taken into account. Moisture has a large effect on the psychrometric condition of the air and hence heat flow from the various heat sources, most importantly the mine personnel. Air must be kept reasonably dry for two reasons:

(i) The human body cools mainly by evaporation, which is reduced if the humidity is high.

(ii) When water evaporates into air it reduces the dry bulb temperature, thereby inducing more heat transfer to the air.

Heat transfer is enhanced when a surface is wet due to latent heat transfer. In the case of strata heat this can have a considerable effect. Sources of moisture in mine air are:-

(i) Ambient atmospheric moisture.

(ii) Natural ground water.

(iii) Service water. Dust suppression water, hosing down.

(iv) Personnel.

(v) Heat exchangers Cooling towers, spray chambers.

(vi) Oxidation and chemical processes.

CHAPTER 3

PSYCHROMETRY

CHAPTER 3

PSYCHROMETRY

3.1 INTRODUCTION

Mine environmental engineers have a particular interest in psychrometry, the study of moisture and heat in air for two reasons. Firstly, humidity must be taken into account to understand fully the thermodynamic processes occurring in a mine ventilation circuit and psychrometric relationships provide a quantitative means of assessing climatic changes.

Secondly, heat and humidity have considerable effects on the human body. The physiological consequences of an unsuitable environment are the main reasons for efforts to control heat and humidity which would not otherwise be a problem.

3.2 COMPOSITION OF AIR

The atmosphere surrounding the earth is a mixture of several gases, the main constituents being oxygen, nitrogen and carbon dioxide. Other gases are present in small quantities. Under normal conditions the proportions of these gases do not vary to any significant extent. It is therefore convenient to regard all the gases comprising one gas only, commonly called 'air'. However, in addition to the above mentioned gases the air also contains a certain amount of water vapour. The amount of water vapour can vary considerably. For subsequent treatment the proportions of all constituents other than water vapour are assumed to be constant.

3.3 TERMINOLOGY

The description of the mixture of air and water vapour is not strictly correct. We speak of dry, humid or saturated air. Air is none of these. The atmosphere may be dry, humid or saturated; if dry it contains only air, if saturated it contains the maximum amount of water vapour besides air. This has given rise to terms such as vair (vapour air mixture) and mair (moist air) which reflect the true situation. However, to avoid repetition, terms such as vair velocity etc and allow a standard the term air can be taken to mean the vapour air mixture and if in context there could be any ambiguity it will be qualified, eg dry air, moist air.

3.4 PROPERTIES

An air vapour mixture can be imagined as two constituents mixed uniformly but each exerting its own partial pressure to form a total pressure (Daltons Law). For a given volume and temperature each constituent part of the mixture contribute to the pressure, energy, entropy and total heat of the mixture by the same amount as it would if alone in that space.

Two independent properties are needed to define the state of a pure substance. When the values of any two are fixed (defining a condition) the values of all other properties for that condition can be found. Since moist air consists of water vapour and air, four properties are needed but as air and water vapour have a common dry bulb temperature only three are required. In psychrometric work these three properties are generally the pressure, dry bulb temperature and wet bulb temperature. All the work in this project was carried out at atmospheric pressure.

3.5 BASIC UNIT

For this project the properties of the vapour air mixture are expressed per unit mass of dry air. The kilogram of vapour air mixture would not be satisfactory since the mass of water vapour in the vapour air mixture changes whereas dry air is a non condensable gas at normal temperatures and pressures so its mass remains constant.

3.6 BASIC PSYCHROMETRIC EQUATIONS

The basic psychrometric equations can be derived by applying the perfect gas equation to the air and water vapour separately, then combining them by means of Dalton's Law.

The perfect gas equation is

$$P.V = m.R.T \tag{1}$$

Thus for dry air

$$P_a.V = m_a.R_a.T_a \tag{2}$$

and water vapour

$$e.V = m_w.R_w.T_w \tag{3}$$

Assuming $V = 1\ m^3$ we obtain for dry air

$$m_a = \frac{P_a}{R_a.T_a} \tag{4}$$

and for water vapour

$$m_w = \frac{e}{R_w.T_w} \tag{5}$$

Combining these two we obtain

$$m_a + m_w = M \tag{6}$$

then

$$M = \frac{P_a}{R_a.T_a} + \frac{e}{R_w.T_w} \quad \text{kg of air} \tag{7}$$

According to Daltons Law $P = P_a + e$. Hence $P_a = P - e$. With $s = \frac{R_a}{R_w}$ and $T_a = T_w = T$ (air and water have same temperatures) we can write

$$M = \frac{P - e.(1 - s)}{R_a.T} \quad \text{kg of air} \tag{8}$$

Similarly we obtain

$$\frac{m_w}{m_a} = \frac{e}{R_w.T} \times \frac{R_a.T}{P_a} \tag{9}$$

$$= \frac{e.s}{P - e} \quad \text{kg of water vapour/kg dry air} \tag{10}$$

Also

$$\frac{m_w}{M} = \frac{e}{R_w.T} \times \frac{R_aT}{P - e.(1 - s)} \tag{11}$$

$$= \frac{e.s}{P - e.(1 - s)} \quad \text{kg of water vapour/kg of air} \tag{12}$$

Substituting the known values in the above equations:

$R_a = 0.287$ kJ/kg K

$R_w = 0.461$ kJ/kg K

$T = 273.15 + t_{db}$ K

$s = \frac{R_a}{R_w} = 0.622$

$1 - s = 0.378$

We obtain the following equations for any temperature and pressure:-

Specific volume of dry air

$$V_a = \frac{0.287045\ (273.15 + t_{db})}{P} \quad m^3/\text{kg of dry air} \qquad (13)$$

Density of dry air

$$\rho_a = \frac{P}{0.287045\ (273.15 + t_{db})} \quad \text{kg of dry air}/m^3 \qquad (14)$$

N.B. for dry air $e = 0$, so $P = P_a$

True specific volume of air

$$V = \frac{0.287045\ (273.15 + t_{db})}{P - 0.378\ e} \quad m^3/\text{kg of air} \qquad (15)$$

True density of air

$$\rho = \frac{P - 0.378\ e}{0.287045\ (273.15 + t_{db})} \quad \text{kg of air}/m^3 \qquad (16)$$

For many calculations the apparent specific volume or density based on 1 kg of dry air is more useful than the true specific volume.

Apparent specific volume of air

$$V_{sp} = \frac{0.287045\ (273.15 + t_{db})}{P - e} \quad m^3/\text{kg dry air} \qquad (17)$$

Apparent density of air

$$\rho_{sp} = \frac{P - e}{0.287045\ (273.15 + t_{db})} \quad \text{kg of dry air}/m^3 \qquad (18)$$

A more detailed derivation of equations (15) to (18) is given by McPherson [10].

Apparent specific humidity, or moisture content (from 10)

$$x = \frac{0.622\,e}{P - e} \text{ g of water vapour/kg dry air} \qquad (19)$$

Relative humidity

$$\phi = \frac{e}{e_{sdb}} \times 100\% \qquad (20)$$

3.7 VAPOUR PRESSURE AND WET BULB TEMPERATURES

Many of the above formulae include the factor of vapour pressure, 'e'. Although the vapour pressure is generally small compared with the total barometric pressure (P) it is important that it be known accurately because the amount of water vapour in the atmosphere is a function of the vapour pressure and since the heat content of the water vapour can be large compared with the sensible heat of the air, an accurate determination of the amount of water vapour present is important in order to obtain an accurate value of the content of air.

Calculation of 'e' is from measurement of the ambient wet and dry bulb temperatures. The wet bulb temperature is measured using an ordinary thermometer which has a cloth sleeve over the bulb which is wetted. Air passes over this becoming saturated as evaporation takes place from the wet cloth surrounding the bulb. The temperature and consequently the thermometer reading drop until the temperature reached is that at which there is equilibrium with heat transfer from the air to the thermometer just balancing the heat transferred by evaporation of the water. Thus the sensible heat lost by the air is equal to the product of the mass, specific heat of moist air (C_{pm}) and the temperature change.

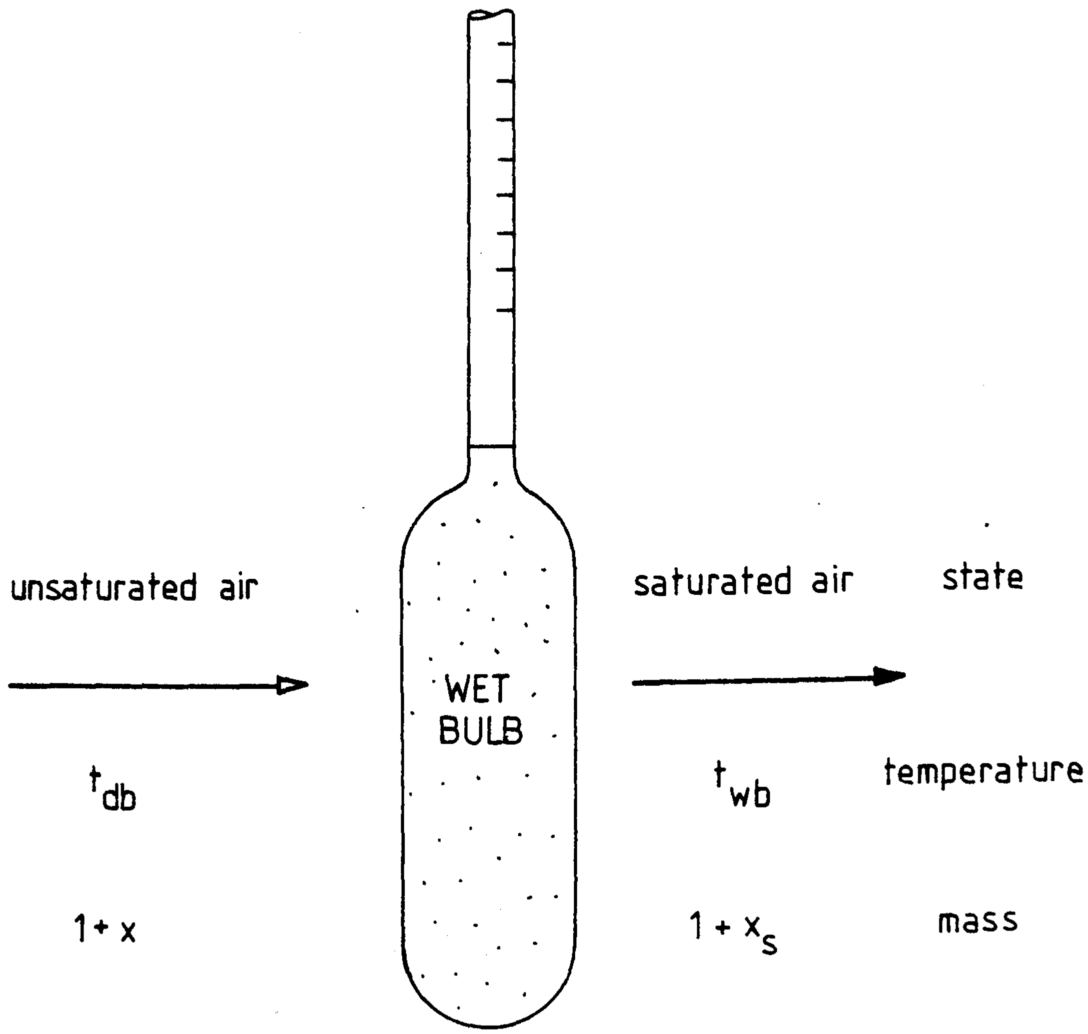


FIG 3.7a WET BULB THERMOMETER

If 1 kg of dry air has x kg of mositure it may be shown that

$$\text{Sensible heat loss} = (1 + x)\ C_{pm}\ (t_{db} - t_{wb})\ (x \text{ in kg/kg}) \tag{21}$$

From accepted relationships between specific heats found in a paper by Parkzewski and Hinsley on Hygrometry in Mines [11] it is shown that

$$C_{pm} = \frac{C_{pa} + C_{pw}.x}{1 + x} \tag{22}$$

where C_{pw} = Specific heat of water vapour

C_{pa} = Specific heat of dry air

Applying this to (22) we have

$$\text{Sensible heat loss} = (C_{pa} + C_{pw}.x)\ (t_{db} - t_{wb}) \tag{23}$$

The latent heat transfer from the wet bulb to the air = L $(x_s - x)$ where L = latent heat of evaporation of water and x_s = moisture content of air at saturation.

For equilibrium to be maintained

$$(C_{pa} + C_{pw}.x)\ (t_{db} - t_{wb}) = L\ (x_s - x) \tag{24}$$

From (19) in basic psychrometric equations

$$x = \frac{0.622.e}{P - e}\ \text{kg/kg} \tag{25}$$

at saturation equation (25) becomes

$$x_s = \frac{0.622.e_s}{P - e_s}$$

Therefore, substituting for x and x_s in the heat balance equation (24) it follows that

$$\left[C_{pa} + \frac{C_{pw}.0.622.e}{P - e}\right](t_{db} - t_{wb}) = 0.622.L.\left[\frac{e_s}{P - e_s} - \frac{e}{P - e}\right] \quad (26)$$

Re-arranging

$$e = (P - e).\left[\frac{e_s}{P - e_s} - \frac{1}{0.622.L}\left[C_{pa} + 0.622.\frac{e}{P - e}.C_{pw}\right].(t_{db} - t_{wb})\right] \quad (27)$$

assuming e and $e_s << P$ then

$$e = e_s - \frac{P.C_{pa}}{0.622.L} - \left[\frac{e}{L}.C_{pw}\ (t_{db} - t_{wb})\right] \quad (28)$$

assuming e and $C_{pw} << L$ we may approximate

$$e = e_s - \frac{P.C_{pa}}{0.622.L}(t_{db} - t_{wb}) \quad (29)$$

giving

$$\underline{e = e_s - P.a.(t_{db} - t_{wb})} \quad (30)$$

'a' is called the Psychrometric constant" although it is not constant but varies according to

$$a = \left[\frac{1.616}{L}\right]\left[\frac{1 - e_s}{P}\right]\left[1 + 0.14\frac{e}{P}\right]\ {}^oC^{-1} \quad (31)$$

For a more accurate determination of 'e' Barenbrug [12] gives

$$e = \frac{e_s\ (371.4 + 0.24\ t_{db} - 0.6\ t_{wb} - 0.24\ (t_{db} - t_{wb})\ P}{371.4 + 0.04\ t_{db} - 0.4\ t_{wb}}\ \text{kPa} \quad (32)$$

Within the range of temperature encountered in mining (0–70^{o}C) the saturated vapour pressure e_s may be calculated to within 0.1% using

$$e_s = 0.6105 \exp\left[\frac{17.27t}{237.3 + t}\right] \text{ kPa}$$

A table of saturated vapour pressures appears in the appendix (A.1).

3.8 HEAT CONTENT OF AIR

3.8.1 Enthalpy (total heat)

The enthalpy of the air is the sum of the enthalpy of the dry air and that of the water vapour.

$$H = C_{pa}.t_{db} + C_w.t_{wb}.x + L.x + C_{pw}\,(t_{db} - t_{wb})\,x$$

= Sensible heat of dry air + Sensible heat of water + Latent heat of evaporation + Superheat of water vapour

$$= 1.005\,t_{db} + \Big\{4.187\,t_{wb} + (2501 - 2.387\,t_{wb})$$

$$+ 1.884\,(t_{db} - t_{wb})\Big\}\,x \text{ kJ/kg} \qquad (33)$$

3.8.2 Sigma heat

Of particular importance is a property known as Sigma heat which is defined as the enthalpy minus the sensible heat of the water which has evaporated to vapour.

S = Enthalpy – Sensible heat of water evaporated

$$= 1.005\,t_{db} + \Big[(2501 - 2.387\,t_{wb}) + 1.884\,(t_{db} - t_{wb})\Big]\,x \text{ kJ/kg} \qquad (34)$$

NOTE x is here expressed in kg/kg.

Sigma heat differs only slightly in numerical value from enthalpy and is an important property of air when changes of moisture content occur. Its implications are best illustrated by example:

Consider a long level airway with no heat additions from any source and free water covering the floor. Unsaturated air enters one end and moves slowly through to allow the air to exit in a saturated condition. The liquid water surface will be at wet bulb temperature. At inlet the dry bulb temperature will be higher so sensible heat transfer from the air to the water occurs. Simultaneously mass transfer from the water to the air will take place as water evaporates,a latent heat gain by the air. As illustrated in figure 3.8a the dry bulb temperature falls but the wet bulb temperature remains constant. These heat exchanges continue until the air is saturated, wet bulb, dry bulb and water temperatures being equal. This is known as an adiabatic saturation process. There should be no net flow of heat into or out of the system for a truly adiabatic process so in this case a small correction must be made. Mass is added to the airstream in the form of water which already contains some sensible heat <u>before</u> evaporation. Thus if the process is to be considered truly adiabatic the sensible heat of the water evaporated must be subtracted from the enthalpy. Consequently in an adiabatic saturation process the sigma heat remains identically constant. The thermal changes involved take place internally as sensible heat is transformed to latent heat during evaporation

In a wet bulb thermometer the equilibrium temperature reached is that at which the rate of heat reception by the water from the

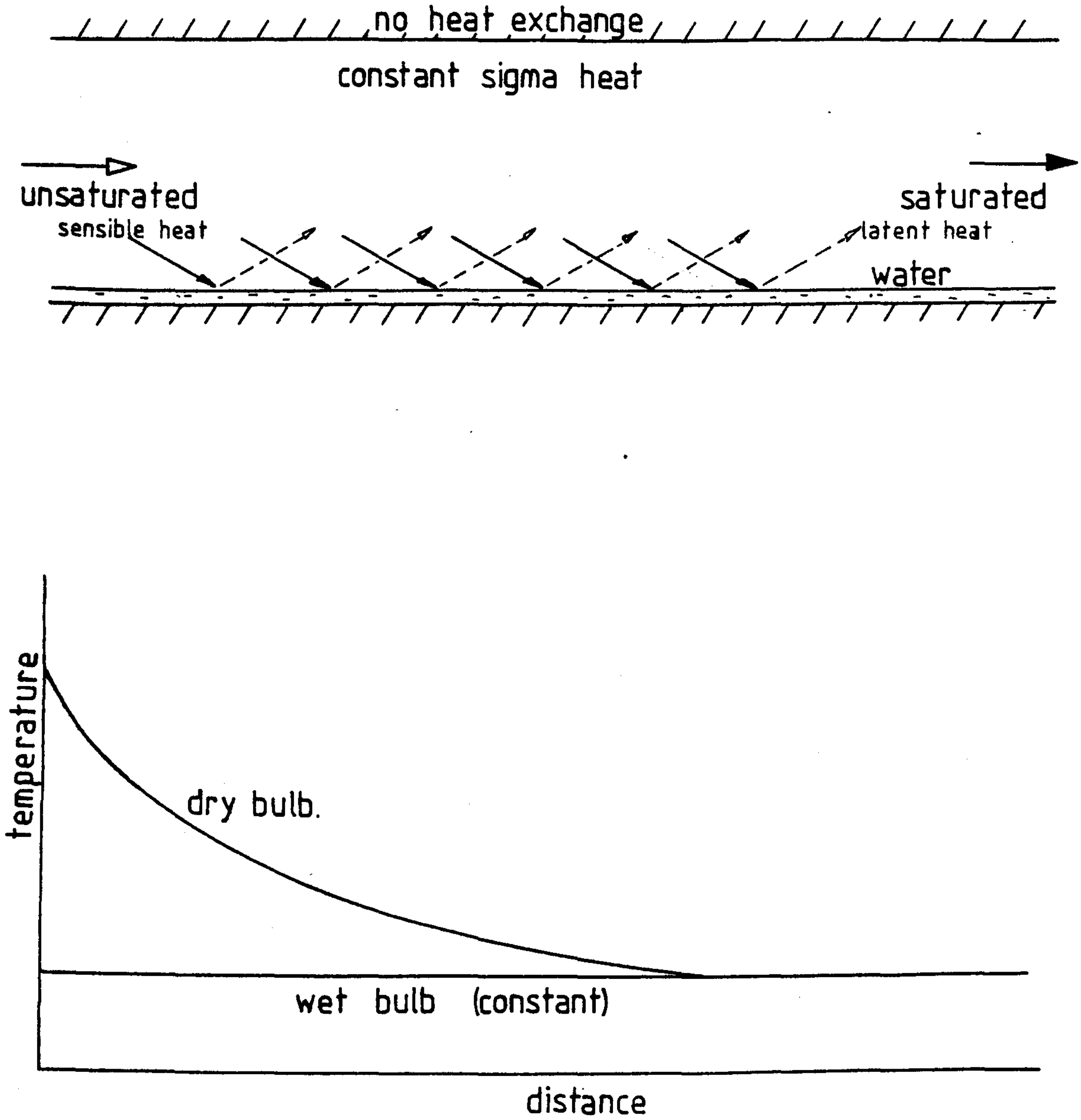


FIG 38a ADIABATIC SATURATION PROCESS (McPherson)

air just balances the loss of heat by evaporation. This equilibrium temperature differs negligibly from the adiabatic saturation temperature. Hence it is normally accepted that the sigma heat remains constant for a constant wet bulb temperature.

Using the concept of sigma heat, heat losses and additions to mine air can be determined from psychrometric observations. The behaviour of the wet bulb thermometer in traversing mine workings gives an immediate indication of the heat transferred to the air from all sources.

3.9 PSYCHROMETRIC PROGRAMS AND CHARTS

To avoid tedious and repeated calculations of psychrometric data two aids are available to the engineer.

3.9.1 Computer evaluation of psychrometric data

Psychrometric computer programs give instant display of relevant information for a given set of input conditions. The University of Nottingham, Department of Mining Engineering has a program on its mini-computer which was used frequently throughout this project. Given wet bulb and dry bulb temperatures and atmospheric pressure the program ("P.S.") gives:-

Psychrometric constant	-	$^{o}C^{-1}$
Vapour pressure	-	kPa
Relative humidity	-	%
Dew point temperature	-	^{o}C
Moisture content	-	g/kg dry air
Specific volume	-	m^3/kg dry air

True density	-	kg/m^3
Vapour enthalpy	-	kJ/kg
Mixture enthalpy	-	kJ/kg dry air
Sigma heat	-	kJ/kg dry air

3.9.2 Psychrometric charts

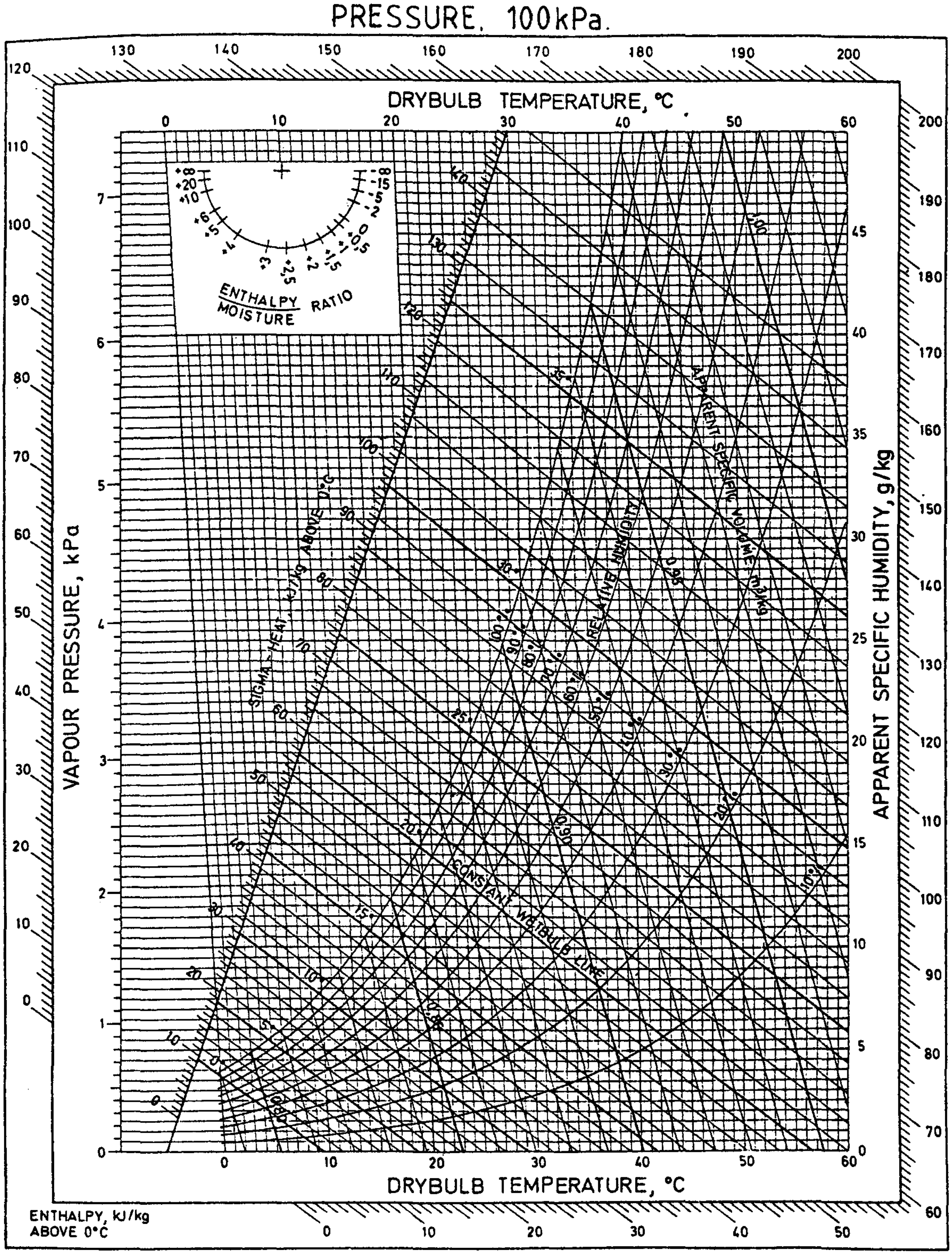
Psychrometric charts are produced in various forms for a range of air pressures. They allow evaluation of various psychrometric parameters rapidly and with little calculation. Such charts are necessarily less precise than the formulae on which they are based but their accuracy is adequate for most routine purposes. The charts are also useful in that processes on air may be followed graphically. The most comprehensive and useful set of charts for mine environmental engineers are those by AWT. Barenbrug. An example of a chart appears as figure 3.9a. Each chart is confined to a set of conditions at a given pressure, but sensitivity analyses show that pressure fluctuations have small effect compared with temperatures. More sophisticated, but unwieldy charts are produced which will take account of pressure changes. The next section illustrates some processes of interest to the mining engineer and relates them to a psychrometric charts.

3.10 PSYCHROMETRIC PROCESSES

3.10.1 Sensible heating and cooling

Sensible heating or cooling is a process where no moisture content variations take place. Therefore no latent heat is exchanged so the heat change is sensible, the temperature changes.

FIG 3·9a PSYCHROMETRIC CHART.



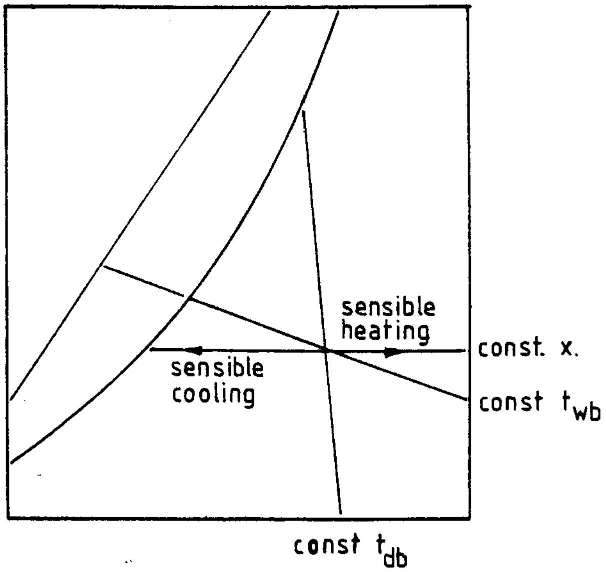


FIG 3·10a SENSIBLE HEATING AND COOLING

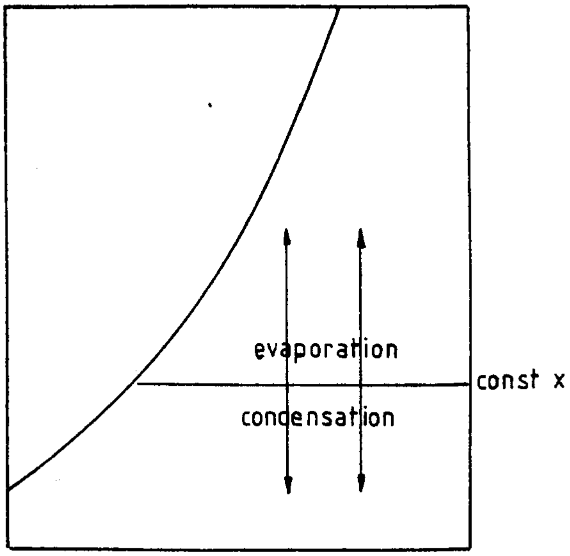


FIG 3·10b EVAPORATION AND CONDENSATION

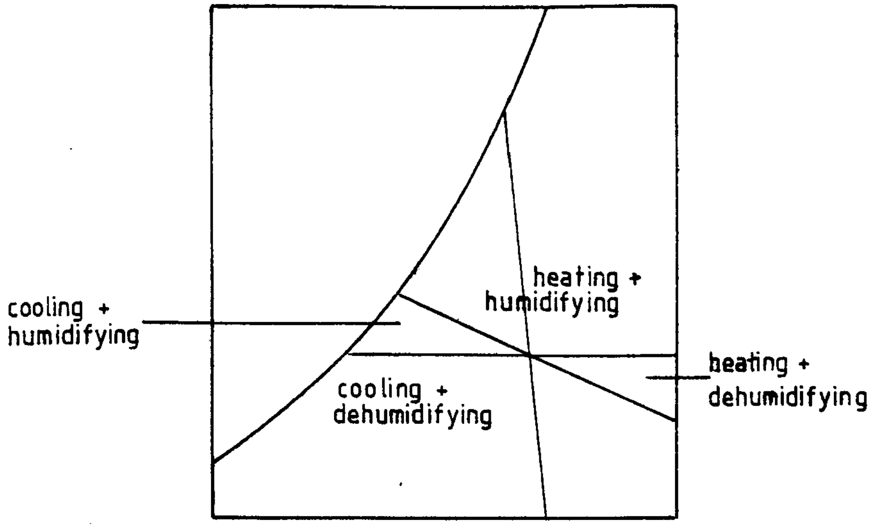


FIG 3·10c COMBINATIONS OF HEATING AND COOLING HUMIDIFYING AND DEHUMIDIFYING

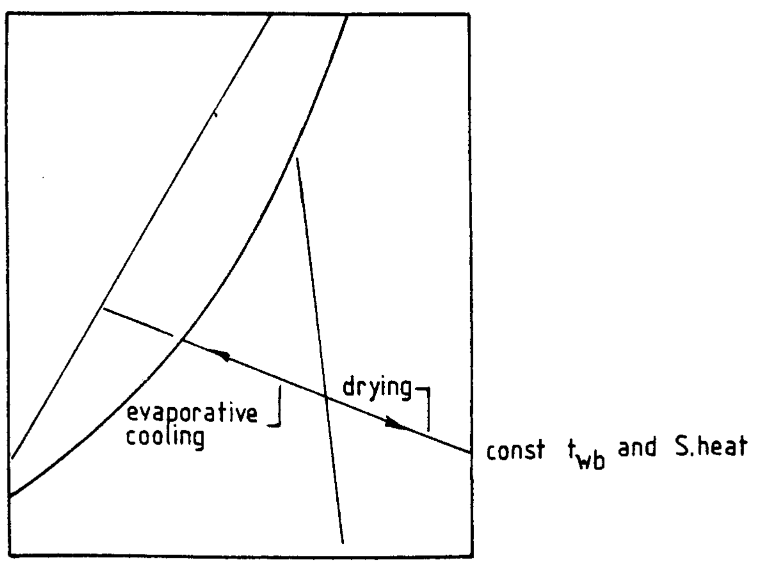


FIG 3·10d EVAPORATIVE COOLING AND DRYING

Such processes follow the horizontal moisture content line (figure 3.10a).

3.10.2 Evaporation and condensation

Evaporation and condensation are typified by changes in moisture content of the air. Movement along a constant dry bulb line indicates pure latent heating or cooling with evaporation or condensation respectively (figure 3.10b).

3.10.3 Combinations of processes

Figure 3.10c shows changes involving combinations of processes, two of which commonly occur in mines. Cooling and dehumidifying occurs in an air chiller. Heating and humidifying takes place at several heat sources. Wet roadways, cutting machines using water and coal conveyors are good examples.

3.10.4 Adiabatic process

An adiabatic process takes place at constant wet bulb temperature and hence constant sigma heat. Heat is neither added nor taken away, the only heat transfer being internal between air and water. An example of such a process is an evaporative cooler where water enters at air wet bulb temperature.

CHAPTER 4

PHYSIOLOGY AND HEAT STRESS

CHAPTER 4

PHYSIOLOGY AND HEAT STRESS

4.1 THE METABOLIC HEAT BALANCE

The heat exchange between any object and its environment takes place according to clearly defined physical laws. Heat is exchanged by combinations of processes, all occurring at the interface between the object and its environment. Although the same laws apply to all living bodies there are noteworthy differences applying to humans. These are of great significance in the ability to maintain thermal balance and a fixed body temperature ($36.8^{\circ}C$) despite widely varying environmental conditions.

Firstly, there is the continuous and variable production of heat from biochemical processes involved in metabolic activity. Secondly, physiologically the body can react dynamically, through cardiovascular and sweating mechanisms to changes in the overall heat load so as to modify the rate of heat transfer from the body core to the peripheral tissues and hence, by changes of surface characteristics to the environment.

The body can be seen as a heat engine taking in food and oxygen and by low temperature oxidation producing work and, due to its inefficiency, large amounts of heat. This metabolic heat generation varies according to muscular activity, physical and mental state and age of the person. It must equate to net heat loss to the environment for equilibrium to be maintained.

The heat balance between the body and the environment can be represented as a heat balance equation.

$$MHG \pm C \pm R - E = \pm S \qquad (1)$$

where

MHG = Metabolic heat generation

C = Heat lost or gained by convection

R = Heat lost or gained by radiation

E = Heat lost by evaporation

S = Heat gained or lost by storage.

C + R + E may be collected together into a single term, air cooling power (ACP). Although the body rarely achieves precise thermal equilibrium over a period storage (s) can be taken as negligible.

4.2 HEAT EXCHANGE EQUATIONS

With regard to the cooling terms Leithead [13] gives the numerical approximations with the following descriptions:-

4.2.1 Convection

If the air temperature is lower than the skin temperature the body loses heat by convection. Even in still air slight convection currents are caused by hot bodies which allow heat transfer. For a nude man

$$c = 8.16 \sqrt{U} \, (t_s - t_a) \qquad (2)$$

where

U = air velocity (m/s)

t_s = skin temperature (^oC)

t_a = air temperature (^oC)

c = convective heat exchange (W/m^2) can be positive or negative.

4.2.2 Radiation

If the body is hotter than any local surface radiant heat may be dissipated.

$$r = 6.63\ (t_s - t_g) \qquad (3)$$

where

t_g = globe temperature (^{o}C)

t_s = surface temperature (^{o}C)

r = radiant heat exchange (W/m^2)

r may be positive or negative. In the case of men near machines radiant heat transfer to the worker could cause special problems.

4.2.3 Evaporation

This is the most powerful cooling mechanism for the human body. The latent heat transfer of sweat evaporating is always away from the body. Evaporative cooling can only take place whilst the wet bulb temperature is above the skin temperature, hence the importance of wet bulb temperature and humidity. The determination of heat lost by sweating is more complicated than by other means. The present interpretation of the situation is that as the demand for heat loss by evaporation increases the amount of sweat production and area of body wetted increases. Eventually when the whole skin area is wetted further sweat production gives no increase in cooling but runs off as liquid. (Clifford et al 1959) give the rate of heat loss by sweating and evaporation as

$$e = 109\ U^{0.63}\ (p_s - p_a) \qquad (4)$$

where

p_s = skin vapour pressure (kPa)

p_a = air vapour pressure (kPa)

e = evaporative cooling (W/m^2)

U = air velocity (m/s)

Attention is drawn to the facts that:-

(i) The equations for heat loss are empirical.

(ii) The equations are converted to SI units from those used by the medical profession.

(iii) The air velocity has a large effect on heat transfer by convection and evaporation.

(iv) Vapour pressure is 'p' here rather than 'e' used elsewhere.

4.3 RESPONSE TO HOT CONDITIONS

When a man is forced to work in hot conditions and metabolic heat generation exceeds air cooling the body reacts as follows:-

4.3.1 Equilibrium

The heart rate increases, blood vessels dilate, especially near the surface and the sweat rate increases. This results in improved heat rejection to the environment with only mild strain of the regulatory mechanism.

4.3.2 Heart strain

If conditions are such that normal processes cannot maintain equilibrium,fluid loss due to sweating reduces the blood volume.

This results in circulatory instability as the heart rate increases dramatically in response to the reduced blood volume. This situation cannot be maintained and the blood pressure falls. At this stage the workman feels considerable discomfort, but if work is continued a critical stage is reached.

4.3.3 Heat stroke

The blood supply to the skin is cut drastically, sweating stops as the glands fail and even less cooling takes place. The body core temperature rises and collapse takes place. This is heat stroke.

Acclimatisation and higher states of fitness make the body much more able to contain the situation at the first stage. The Chamber of Mines of South Africa has refined acclimatisation procedures over the years to a high level. These are well documented by Wyndham [14] who also states that the risk of heat stroke is present above 27^{o}C wet bulb, a temperature often exceeded in British coal mines.

The symptoms of heat strain are in the primary stages psychological, lethargy and lack of care with work. This escalates to headaches and nausea and eventually coma and failure of the body mechanisms with possible irreversible damage. The next stage is death. No cases of heat stroke have yet been reported in the UK, but due to the fact that the older or unwell worker is more susceptible it is entirely possible that heat stroke has been reported as something else, or precipitated it, for example heart trouble. The primary symptoms should also be watched for in a hot environment to avoid worse problems. It should be noted that they are similar to a hangover.

It is accepted that environmental conditions have an effect on productivity. Several workers over the years have published work relationships between reduced productivity and hot conditions notably the Chamber of Mines of South Africa, Cooke et al 1968 [15]. Apart from the reduced physical performance there are also other intangible but costly disadvantages of working in hot conditions relating to mental state and performance, increased susceptibility to accidents and absenteeism.

4.4 HEAT STRESS INDICES

The heat stress of any given situation, is the combination of all those factors which result in heat gains to the body or which prevent the body from dissipating heat. Physiological strain as indicated by sweat rate, heart rate and body temperature has been studied in relation to certain heat stress parameters, singly and in combination. The relative effects of the various parameters differ in different circumstances and their interaction upon physiological and psychological reactions is complex. It is for this reason that so many different systems have been proposed to give a quantitative expression for the heat stress-strain relationship.

Many attempts have been made to devise instruments that would integrate all parameters into a single reading usually by imitating the human body. None has so far proved entirely satisfactory in all conditions mainly due to their inability to allow for all heat exchanges and in the correct proportions. For this reason the many systems based on formulae or on nomograms with several input parameters have been devised. Heat stress indices fall into three groups:-

(i) Indices based on subjective preference.

(ii) Indices based on physiological observations.

(iii) Indices based on analysis of heat exchange.

The following sections deal with heat stress indices used in the mining industry.

4.5 SPECIAL THERMOMETERS

Three systems have been used in the mining industry:-

4.5.1 Wet bulb thermometer

Due to the importance of evaporative cooling wet bulb temperature still remains the easiest to measure and widely acceptable single measure of heat stress. It is of limited value though in high air velocities and high radiant temperatures.

4.5.2 Kata thermometer

The kata is similar to a conventional thermometer, but has only two graduations. It is heated then allowed to cool so that the time may be taken as the fluid falls between the two graduations giving a reading linked to cooling rate. It has limitations due to its small size compared with the human body, but was used in South Africa in a wet bulb form until quite recently.

Plate 1

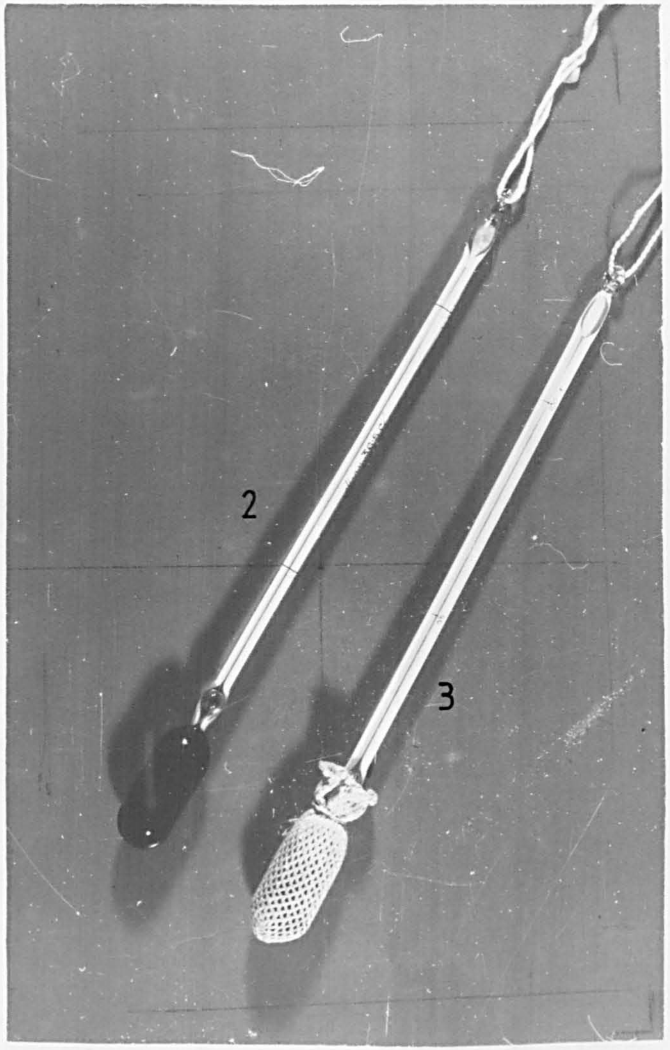
Globe and Kata Thermometers

1 Globe thermometer

2 Dry bulb Kata thermometer

3 Wet bulb Kata thermometer


1


2
3

4.5.3 Wet bulb globe thermometer

This device consists of a thermometer whose bulb is central in a matt black metal globe. It takes good account of radiant heat but is not reputed to be reliable in extremely severe conditions. It was devised by the US Army as a simple alternative to the ET system.

4.6 SYSTEMS USING NOMOGRAMS AND FORMULAE

4.6.1 Effective temperature scales

This system was devised by the American Society of Heating and Ventilation Engineers in 1923 to assess the subjective comfort of various combinations of wet and dry bulb temperature and air velocity. The scales were devised by moving men from one room to another at various conditions and recording subjective comparisons. As the system was devised as a comfort rather than heat stress index it has shortcomings in very hot conditions. It does not give sufficient weight to the deleterious effects of air velocities lower than 0.5 m/s or the harmful effects of high air velocities at high temperatures. Several derivatives have been designed over the years for use in heavy industrial situations but these do not only reduce its shortcomings due to use out of context. This system nevertheless is widely used in Europe and by the NCB.

4.6.2 Predicted 4 hour sweat rate

This system was devised empirically by the British Medical Research council in W.W.II specifically to evaluate physiological effects of working in severe conditions. The nomogram devised

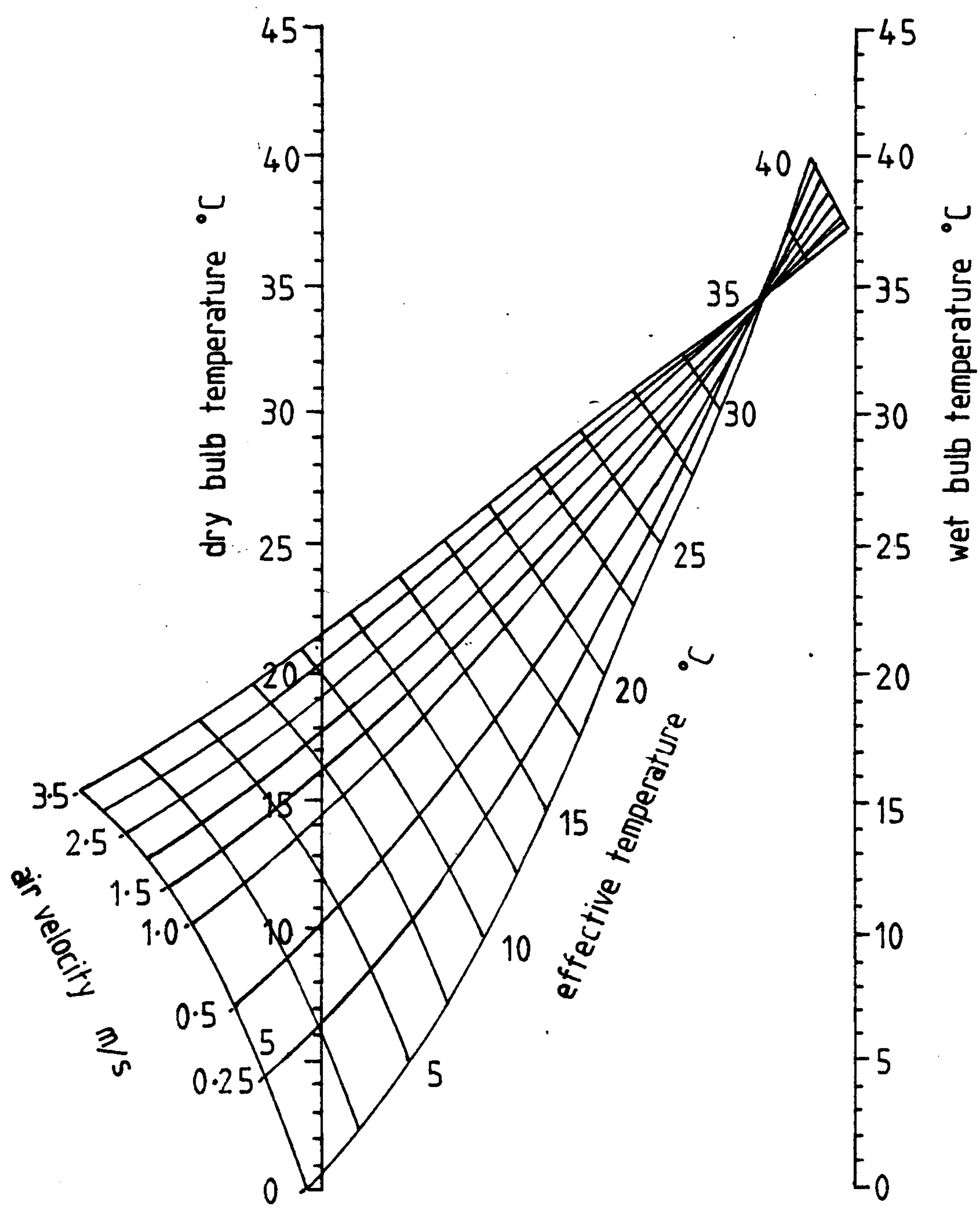


FIG 4.6a EFFECTIVE TEMPERATURE NOMOGRAM

gives the heat stress in sweat units and despite being accurate and reliable has not found favour in mining due to its complexity rendering it unsuitable for making spot checks. It is used in a simplified form at Mount Isa Mine, Australia.

4.6.3 Specific cooling power

In 1952-55 Haines, Belding and Hatch devised a system (HSI) for the assessment of heat stress based on sound engineering principles of heat transfer. This index though, had some shortcomings in the evaluation of heat transfer coefficients and the assumption of a skin temperature of 35°C. The Human Sciences Laboratory of the Chamber of Mines of South Africa recognised the basic soundness of HSI and modified it to avoid some shortcomings. Using wind tunnels and sophisticated instrumentation to provide a wide variety of conditions they derived heat transfer equations for men working in a variety of conditions. These equations are presented in a way which has meaning both in terms of the physiological reactions of men and the performance of ventilation systems.

The maximum heat emission q.(W/m^2) from a worker is given by the following equations:-

$$r = 17 \times 10^{-8} (t_r/2 + 290.7)^3 (t_s - t_r) \qquad (5)$$

$$c = 8.3 \times (p_a/101.3)^{0.6} \times U^{0.6} \times (t_s - t_a) \qquad (6)$$

$$e_{max} = (15100/p_a)(p_a/101.3)^{0.6} \times U^{0.6} \times (\phi p_{wa} - p_{ws}) \qquad (7)$$

$$q_{max} = r + c + e_{max} \qquad (8)$$

where

t_s = mean skin temperature (^{o}C)

t_r = mean radiant temperature (^{o}C)

t_a = mean air temperature (^{o}C)

p_a = air pressure (kPa)

p_{wa} = saturated vapour pressure at air temperature (kPa)

p_{ws} = saturated vapour pressure at skin temperature (kPa)

ϕ = relative humidity (%)

U = air velocity (m/s)

An assumption made is that 75% of the body area is involved in radiant heat exchange.

Studies have shown that the skin temperature of men in heat equilibrium exhibiting only mild strain is close to 35^{o}C in all conditions. Also for most underground environments one can introduce the following approximations without appreciable errors:-

(i) $p_a = 100$ kPa

(ii) $t_a = t_r = t_w + 2^{o}C$ (9)

This allows the construction of the nomogram, figure 4.6b, for specific cooling power at 100 kPa. The engineer can use calculations of SCP to determine most economically particular cooling rates for workers. Also values of cooling power can be used in conjunction with estimates of metabolic rate to assess the environmental conditions needed for a man to perform a certain job. Neglecting energy leaving the body in other forms a workman will attain equilibrium if the cooling power of the environment equals or exceeds the metabolic heat generation associated with his task. Figure 4.6c shows some examples.

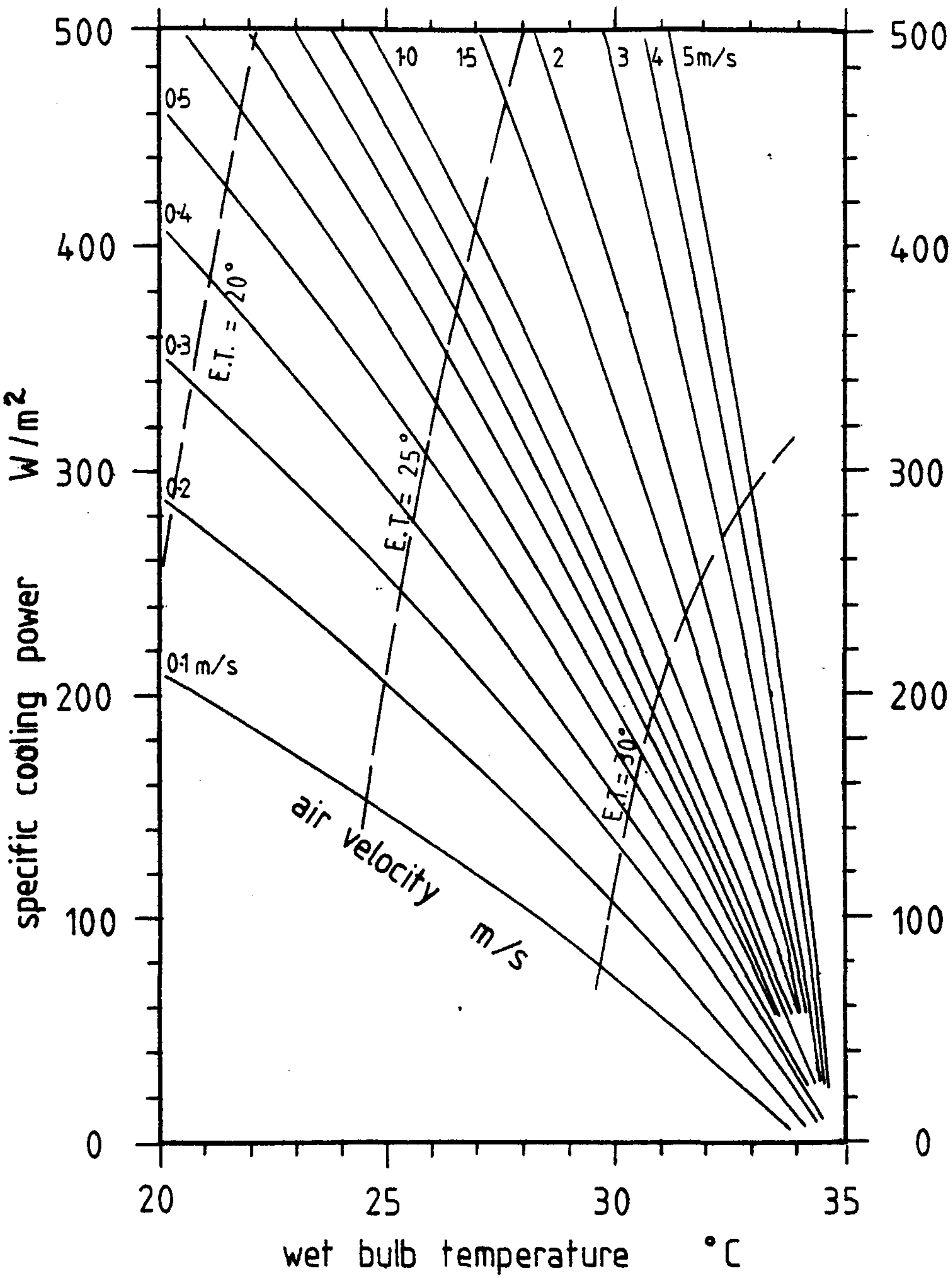
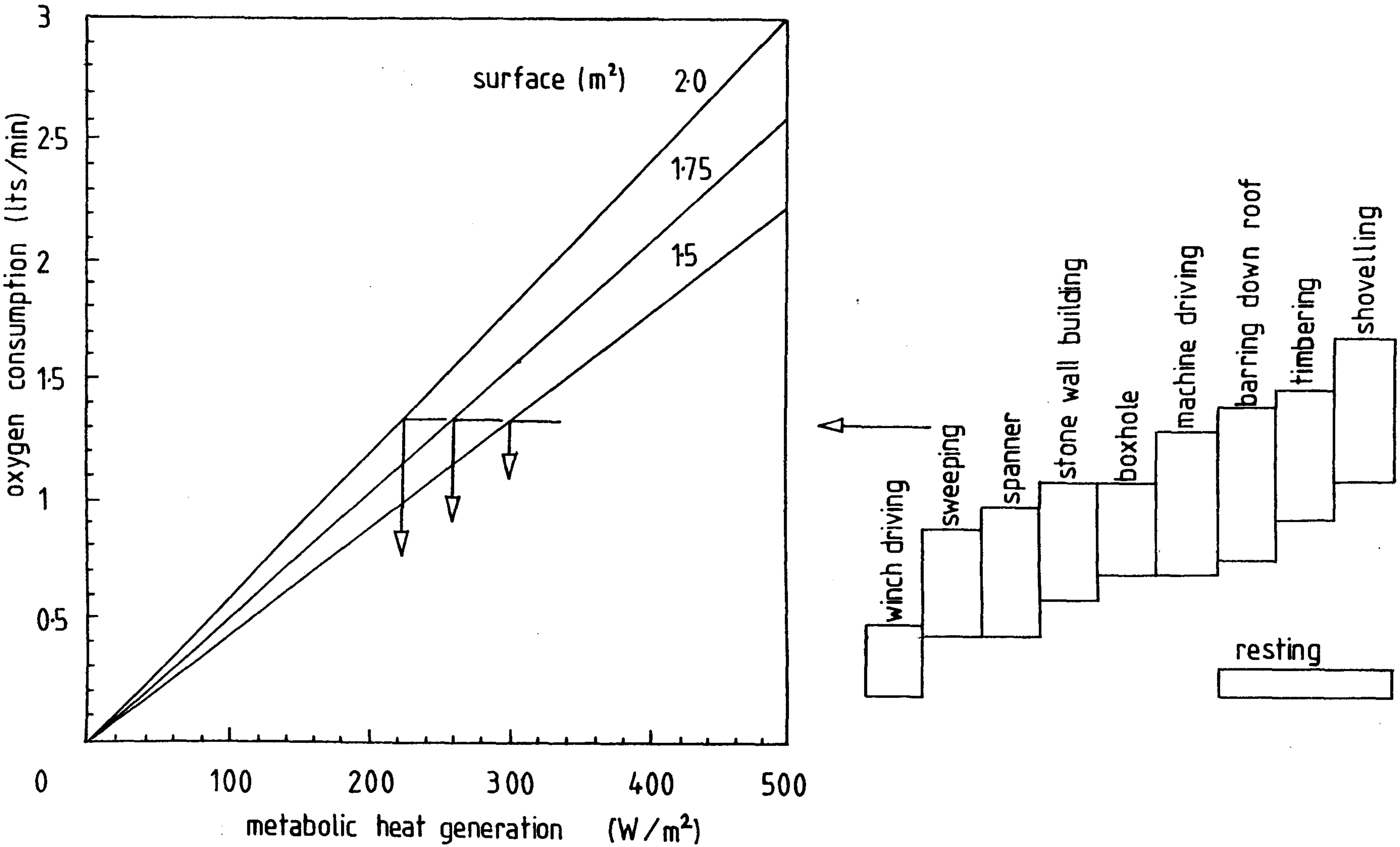


FIG 4·6b SPECIFIC COOLING POWER NOMOGRAM

FIG 4·6c METABOLIC HEAT GENERATION OF VARIOUS TASKS (Wyndham)



CHAPTER 5

HEAT TRANSFER

CHAPTER 5

HEAT TRANSFER

5.1 INTRODUCTION

In any substance having a temperature above absolute zero (0 K-273°C) the molecules are not stationary but vibrate. Energy is stored in the vibrating motion and when more energy is added, when it is heated, providing there is no phase change, the vibration and temperature increase. Hence heat may be regarded as a form of energy which passes from one substance to another at lower temperature, be it solid or fluid. The study of heat transfer deals with the mechanisms of heat exchange and the rate at which energy is transferred. Quantitative analysis allows prediction of temperatures due to heat flows and also allows assessment of measures by which heat flows may be enhanced or reduced.

Three modes of heat transfer may be distinguished. These are conduction, convection and radiation. Common to all types of heat transfer is the fact that a temperature difference is necessary and heat flows from the hotter body to the cooler one.

5.2 CONDUCTION

When temperature differences are present in any matter, heat flows from the hot to the cold regions until the temperatures are equalised. This heat will also flow across the boundary of two substances that are in contact. The heat flows without appreciable displacement of the molecules of the matter, hence it is normally associated with solids, however it can take place in fluids at rest and fluids in laminar flow. Steady state heat conduction is described by Fouriers law of conduction.

5.2.1 Fouriers Law

Fouriers law of conduction is based on the empirical observation of one dimensional steady heat flow through a solid. One dimensional implies that the temperature is uniform over surfaces perpendicular to the direction (x) of heat conduction and such surfaces are called isothermal surfaces. 'Steady' implies that the properties at any point, notably temperature, do not vary with time; also the flow of heat through successive surfaces is constant.

Considering a plane layer of thickness dx, with one face maintained at a temperature t, and the other at t + dt, the rate of heat flow Q, is found to be proportional to the area of flow A, and the temperature difference, dt, across the layer. It is inversely proportional to the thickness dx. This is Fouriers law and it can be expressed by the equation

$$Q = - kA \left(\frac{dt}{dx}\right) \qquad (1)$$

The minus sign arises from the fact that the heat flow is in the opposite direction to the temperature gradient. It follows from equation 1 that for steady flow (Q independent of x) the temperature will be constant if k is constant. Integrating between limits x_1 and x_2 to enable the heat flow to be expressed in terms of surface temperatures t_1 and t_2 gives

$$Q = - kA\left(\frac{t_2 - t_1}{x_2 - x_1}\right) \qquad (2)$$

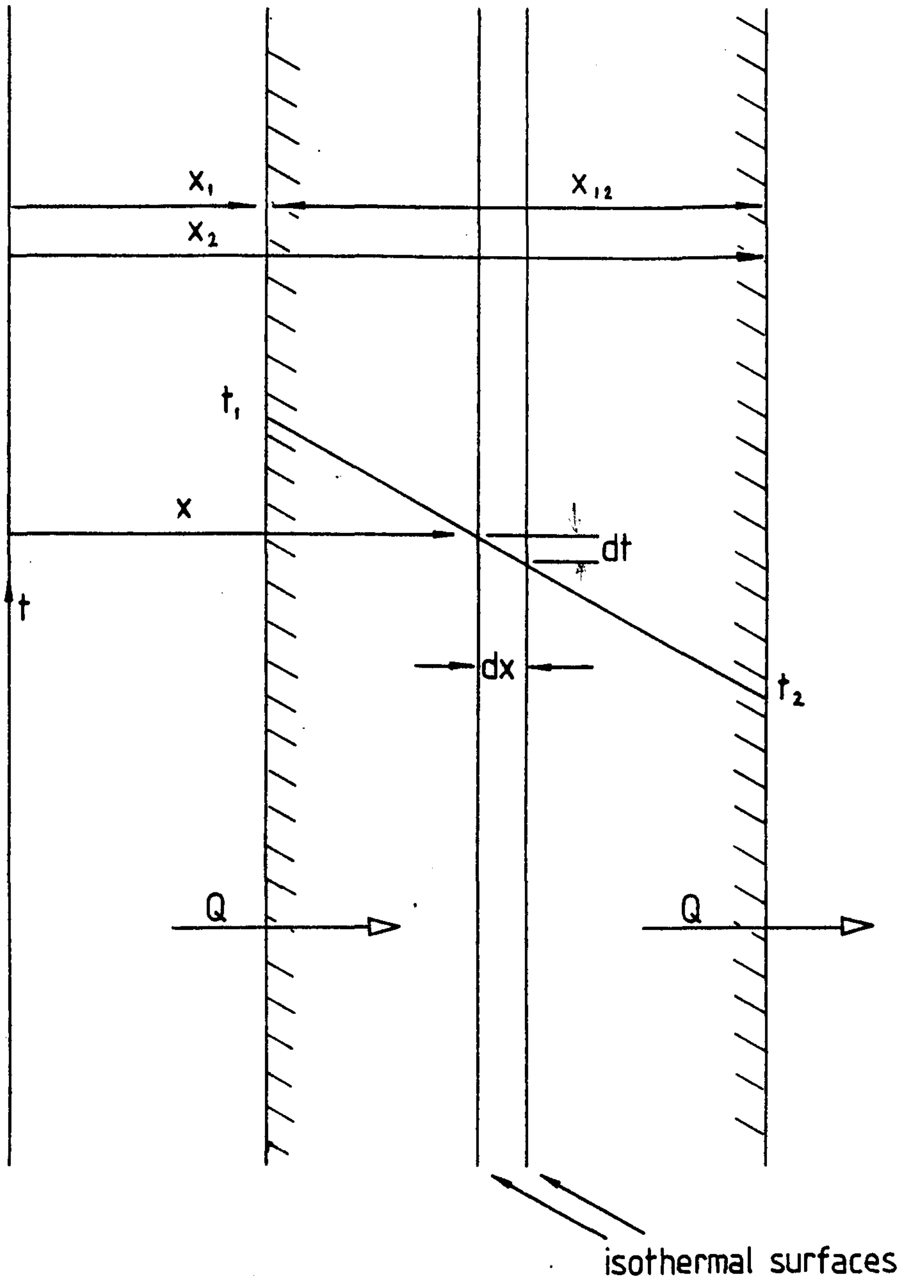


FIG 5·2a STEADY HEAT CONDUCTION THROUGH
A PLANE PARALLEL WALL

where

Q = Heat flow (W)

k = Thermal conductivity (W/m^oC)

t = Temperature (^oC)

x = Distance along x axis (m)

Figure 5.2b

Table of typical values of thermal conductivity, k, in $W/m^2\ ^oC$

Metals		Other Solids		Fluids	
Copper	380	Quartzite	6	Still Water	0.62
Aluminium	190	Concrete	1.7	Still Air	0.028
Brass	97	Wood	0.17		
Steel	45	Insulation	0.034		
Stainless Steel	16	Coal	0.3		

For many materials k varies with temperature but for the range of substances and temperatures found in mining k can be taken as constant. Typical k values are shown in table 5.2b.

5.2.2 Effect of shape

The above equation for evaluating one dimensional heat conduction may be expanded to provide solutions to problems in two or three dimensions and for various shapes. Carslaw [16] and Rogers and Mayhew [17] give several examples.

Cylindrical shapes merit description here as some commonly encountered situations in mining engineering involve radial heat flow. For example heat flow from the inside to outside of pipes or vice versa and radial heat flow into a mine roadway. The equation describing radial heat flow may be derived from the basic Fourier equation. See Figure 5.2c.

$$Q = -kA\,\frac{dt}{dr} \tag{3}$$

$$= -k\ 2\pi r\ \ell\,\frac{dt}{dr} \tag{4}$$

Rearranging 4 we have

$$Q\,\frac{dr}{r} = -2\pi\ k\ \ell\ dt \tag{5}$$

Integrating 5

$$Q\,\ell n\,\frac{r_2}{r_1} = -2\pi\ k\ \ell\ (t_2 - t_1) \tag{6}$$

$$Q = \frac{-2\ k\ \ell\ T_1\ (t_2 - t_1)}{\ell n\left(\frac{r_2}{r_1}\right)} \tag{7}$$

5.2.3 Non-steady flow

In section 5.2 a solution for steady conduction was presented. A more general case involves temperature at a point changing with time. For the case of one dimensional flow a solution can be derived from first principles similarly to the steady state situation. Referring to figure 5.2d the heat flow into a layer dx during a time interval $d\tau$, through an area A is

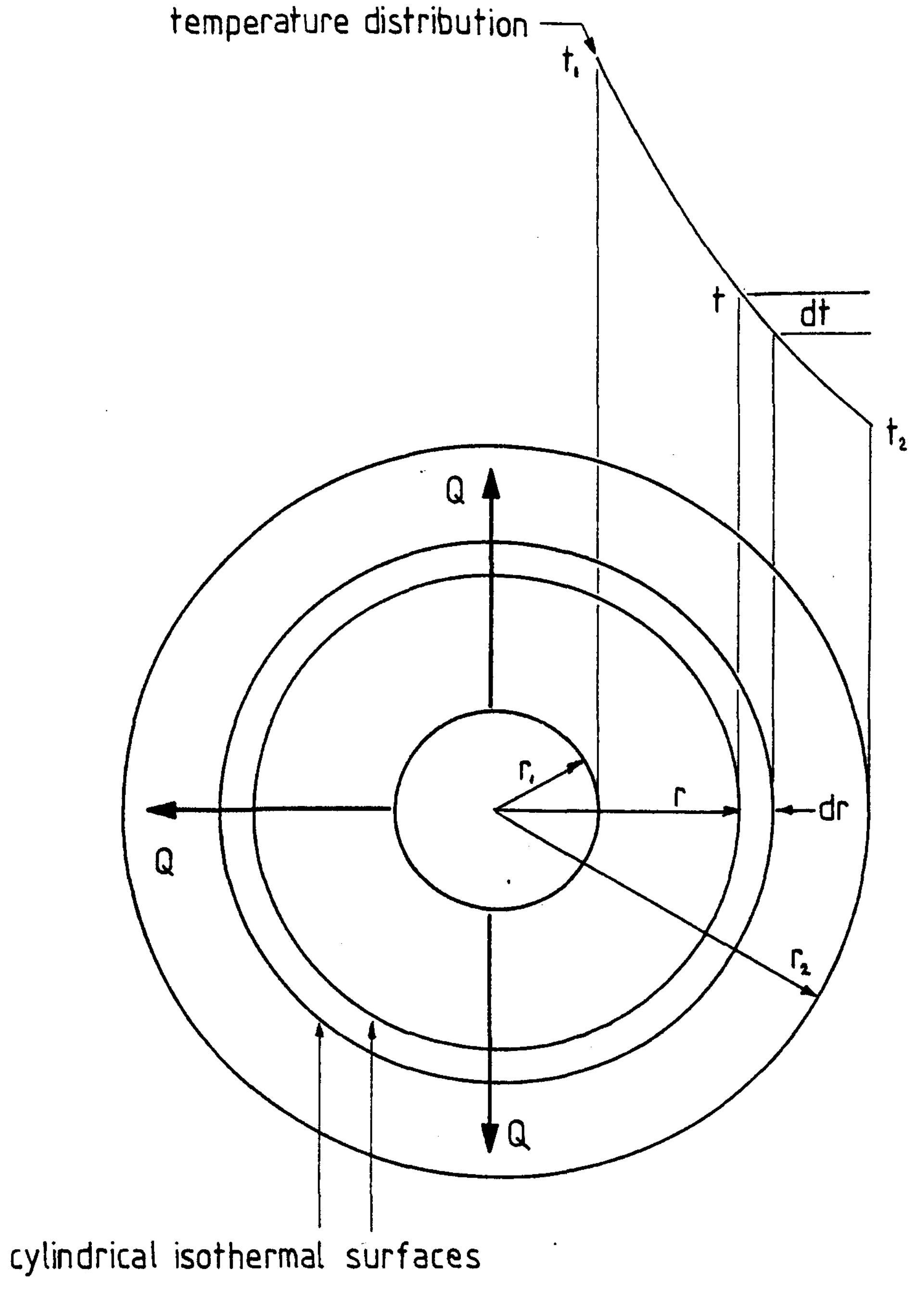


FIG 5·2c STEADY RADIAL HEAT CONDUCTION.

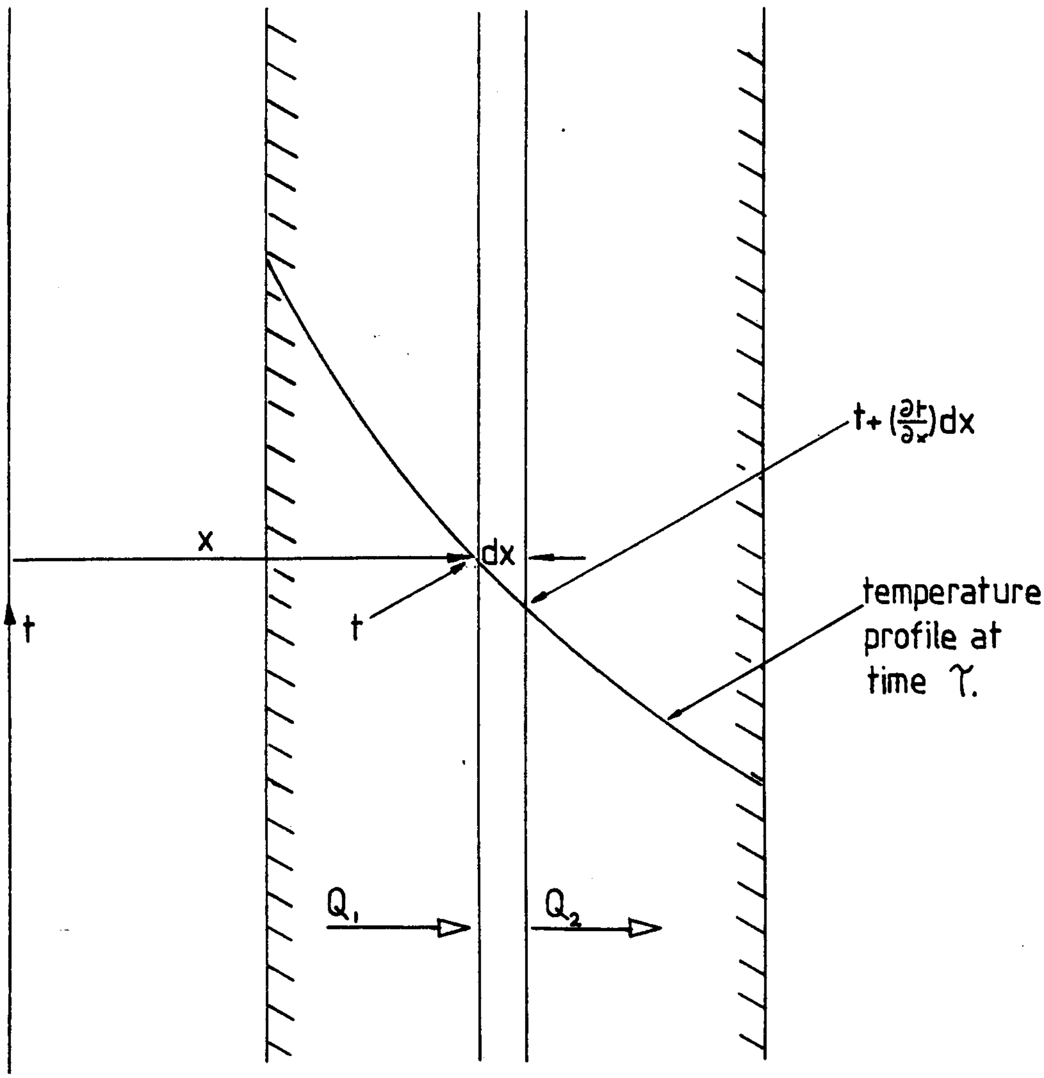


FIG 5·2d NON STEADY ONE DIMENSIONAL CONDUCTION

$$Q_1 = -kA \frac{\partial t}{\partial x} d\tau \tag{8}$$

Q is heat quantity (J) here rather than heat flow rate (W). The corresponding outflow is

$$Q_2 = -kA \frac{\partial}{\partial x} \left\{ t + \frac{\partial t}{\partial x} dx \right\} d\tau$$

$$= -kA \left\{ \frac{\partial t}{\partial x} + \left(\frac{\partial^2 t}{\partial x^2} \right) dx \right\} d\tau \tag{9}$$

The difference between the two heat flows must be equal to the heat stored by the conducting layer dx, during the time interval $\partial\tau$. The temperature rise of the layer during this time is $(\partial t/\partial\tau)\, d\tau$ and hence

$$Q_1 - Q_2 = \rho.c\ (A.dx) \left(\frac{\partial t}{\partial \tau} \right) \partial\tau \tag{10}$$

Thus

$$\rho.c\ (Adx) \left(\frac{\partial t}{\partial \tau} \right) d\tau = kA \left(\frac{\partial^2 t}{\partial x^2} \right) dx\ d\tau \tag{11}$$

If the thermal diffusivity $a = \frac{k}{\rho.c}$ then

$$\frac{\partial t}{\partial \tau} = a \frac{\partial^2 t}{\partial x^2} \tag{12}$$

The above equation may be used as a basis for finite difference methods to determine temperature profiles, or diffusivity, of conductivity of a material when the relevant parameters are known.

5.3 CONVECTION

The study of heat transfer by convection is concerned with the evaluation of heat exchange between fluids and solid boundaries at a different temperature. It is accepted that the differential equations describing convective heat transfer present considerable mathematical difficulties and exact solutions can rarely be obtained. This section describes some methods of giving approximate solutions to simple, but important cases of steady heat flow.

5.3.1 The boundary layer

The concept of the boundary layer is best illustrated by example. Consider the case of still fluid against a hot vertical surface. If the fluid temperature was taken at increasing but small distances away from the surface the temperature would fall linearly then stabilise as shown in figure 5.3a. The corresponding velocity measurements would indicate zero velocity against the wall and at some small distance from the wall. Between these two positions the fluid would be moving upwards. This narrow zone of movement is known as the boundary layer and it is this which exerts a dominating influence in convective heat flow. Measurements have shown that the thickness of the boundary layer as determined by temperature measurements is almost identical to that determined by velocity. Some methods of evaluating convective heat flow make use of this similarity first pointed out by Reynolds.

The above example is an illustration of natural or free convection where the movement of the fluid is caused by buoyancy effects, the hot fluid rising due to reduced density.

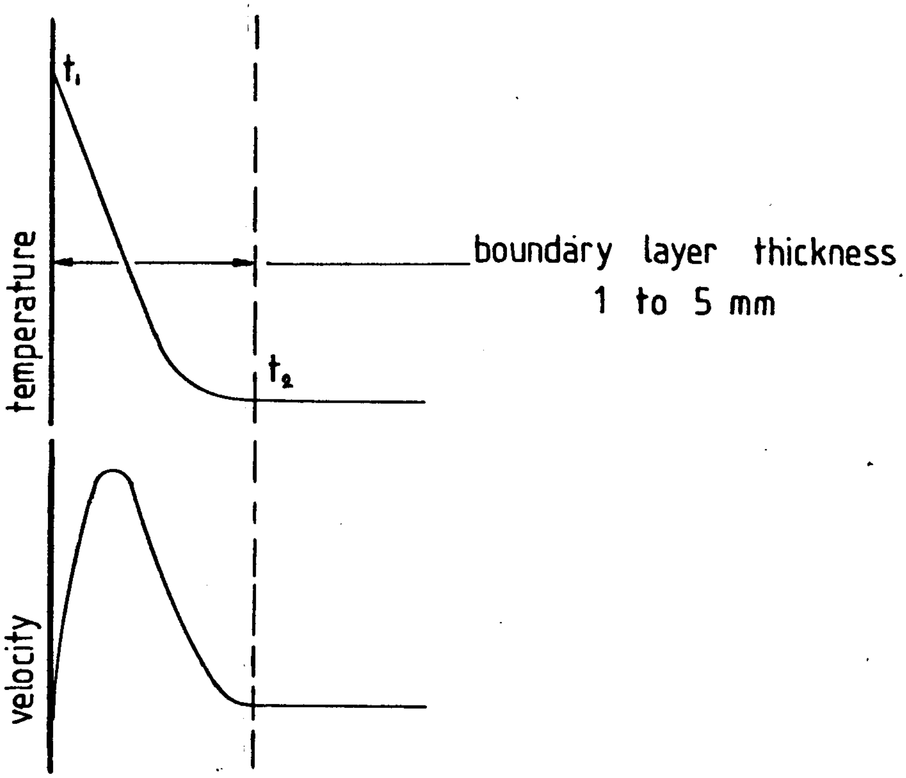


FIG 5·3a NATURAL CONVECTION

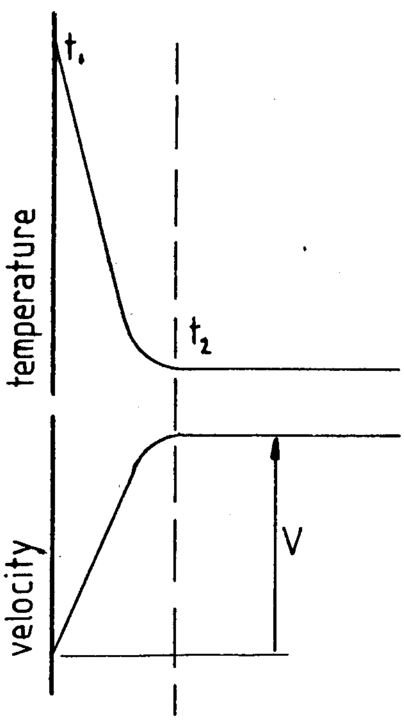


FIG 5·3b FORCED CONVECTION

In many situations the fluid is forced to move over a surface by a fan or pump giving the velocity and temperature profiles shown in figure 5.3b. It is known that the faster the flowing fluid the easier the heat transfer. This is because the heat is conducted across the boundary layer which decreases in thickness as the velocity increases. As shown in table 5.2b the conductivity of air and water the most common fluids encountered is very low.

The thickness of the boundary layer at any position depends on three factors. These are the shape of the solid surface relative to the moving fluid. The velocity and turbulence of the moving fluid and the distance along the surface for which the boundary layer has been able to grow. With regard to heat transfer from coal on a conveyor it can be seen that all these parameters will have high values.

The Reynolds number (R_e) is a dimensionless ratio that characterises the velocity and turbulence of the flow. It is the most important parameter in specifying the thickness of the boundary layer and hence the convective heat transfer coefficient. Reynolds number is calculated as follows

R_e = velocity x specified dimension x density/dynamic viscosity

5.3.2 Convective heat transfer coefficient

Because the thickness of the boundary layer is generally unknown the Fourier heat conduction equation cannot be used to derive a convective heat transfer coefficient. Instead the following equation is used

$$Q_c = h_c A (t_1 - t_2) \qquad (13)$$

where

Q_c = Heat flow between surface and fluid (W)

A = Area involved in heat transfer (m^2)

t = Temperature of surface or fluid (^oC)

h_c = Convective heat transfer coefficient ($W/m^2\ ^oC$)

The convective heat transfer coefficient h_c, depends mainly on the thickness of the boundary layer as mentioned earlier. This in turn depends on the Reynolds number for a given situation. Generally the relationship between the heat transfer coefficient and Reynolds number can be described by

$$h_c = (R_e)^n \times C_o \times k/D \qquad (14)$$

where

k = Thermal conductivity of fluid (W/m^oC)

D = Diameter of pipe or some other representative dimension (m)

C_o = A coefficient (depends on configuration and fluid)

n = An exponent (depends on configuration)

Whillier [18] gives some good examples of convective heat transfer calculation and has produced various nomograms for situations found in mining based on the following formulae

<u>For forced convection</u>

Water inside pipes

$$h_c = 0.023\ R_e^{0.8}\ (C_p\ \mu/k)^{0.4} \times \frac{k}{D} \qquad (15)$$

Air inside pipes or airways

$$h_c = 0.02\ R_e^{0.8} \times \frac{k}{D} \qquad (16)$$

Air outside pipes

$$h_c = 0.2\ R_e^{0.6} \times \frac{k}{D} \qquad (17)$$

<u>For natural convection</u>

Air

$$h_c = 1.4\ (\Delta t)^{\frac{1}{3}} \qquad (18)$$

Water

$$h_c = 190\ (1 + 0.012\ t_s)\ (\Delta t)^{\frac{1}{3}} \qquad (19)$$

where

h_c = Convective heat transfer coefficient ($W/m^2\ {}^oC$)

R_e = Reynolds number

C_p = Specific heat of fluid (kJ/kg K)

μ = Dynamic viscosity (kg/m s)

k = Thermal conductivity of fluid (W/m^oC)

D = Diameter of pipe (m)

Δt = Temperature difference between fluid and surface (^oC)

t_s = Surface temperature (^oC)

5.4 RADIATION

The amount of heat transferred by radiation is not generally large in most coal mining situations, however it must still be evaluated and is included here for completeness.

In the introduction to this chapter it was stated that the heat energy stored in a material above absolute zero is due to molecular vibration. This molecular motion in turn produces electromagnetic waves some of which are radiated in straight lines away from the body. Hence any body above absolute zero emits radiant heat energy.

When this energy impinges on matter it may be totally or partially reflected, transmitted through it or absorbed. Practically all substances encountered in engineering are opaque to thermal radiations, even glass is only transparent in a fairly narrow waveband, so we can assume thermal radiation is either reflected or absorbed at a surface. Thus is it possible to write

$$\rho + \alpha = 1$$

ρ = reflectivity

α = absorptivity

A material whose absorptivity is 1 is known as a black body. The total energy emitted by a black body per unit time by unit area of a black surface is

$$q_b = \delta\ T^4 \qquad (20)$$

where

T = Absolute temperature (K)

q_b = Heat emitted from black surfaces (W/m^2)

$\delta = 56.7 \times 10^{-9}$ (W/m^2 K^4)

δ is known as the Stefan-Boltzman constant

No real material emits and absorbs radiation according to the laws of the black body and few approach the black body condition closely. This has given rise to the term "grey body" which obeys similar laws of energy distribution and whose emissivity does not depend on wavelength. The emissivity, ε, is defined as the ratio of energy emitted by a surface to the energy emitted by a black surface at the same temperature. Some materials and emissivity values are listed in the appendix (A.4).

The table of emissivities shows how clean polished metallic surfaces have low emissivities. These are substantially proportional to the absolute temperature. Oxidised or greasy metal surfaces have emissivities which are several times higher than polished surfaces. The emissivity is a property of surface finish as well as of the material. Surface irregularities may result in rays striking a surface usually not being reflected away after one incidence but suffering multiple reflections first, raising the emissivity value.

When two surfaces can 'see' each other each absorbs some of the radiation emitted by the other. If the surfaces are at different temperatures the net result is a flow of energy from the hotter to the colder body. There are several equations available for different geometrical orientations [17]. Whillier [18] gives an equation for radiative heat transfer combining the effects of emissivity and orientation

$$Q_r = 5.67 \left[(T_1/100)^4 - (T_2/100)^4\right] \times A_1 \times F_{ev} \qquad (21)$$

where

Q_r = Net radiant heat flux (W)

T_{1-2} = Absolute temperature (K)

A_1 = Smaller of two surfaces involved (m^2)

F_{ev} = Emissivity and view factor

The emissivity and view factor is calculated

$$F_{ev} = \left[\frac{1}{\varepsilon_1} + \frac{A_1}{A_2}\left(\frac{1}{\varepsilon_2} - 1\right)\right]^{-1} \qquad (22)$$

See figure 5.4b

Equation 21 may be linearised to give a more easily used equation of a similar form to those used for forms of heat transfer

$$Q_r = h_r . A_1 \ (t_1 - t_2) \ F_{ev} \qquad (23)$$

where h_r is the radiative heat transfer coefficient. The radiative heat transfer coefficient may be determined by equating 21 and 23. Fortunately, h_r, remains fairly constant within the range of temperatures which exist in mining. Much more so than the convective heat transfer coefficient. Its numerical value depends only on the average temperature of the two surfaces involved and is calculated thus

$$h_r = 0.2268 \left(\frac{T_{av}}{100}\right)^3 \qquad (24)$$

In mining situations the temperatures involved are usually such that equation (24) is accurate to within 0.1%.

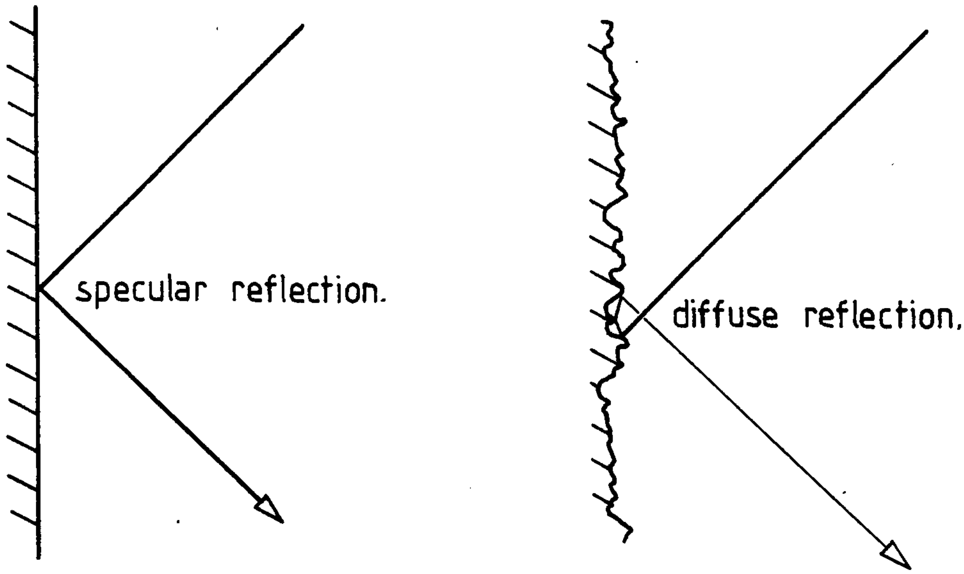


FIG 5·4a EFFECT OF SURFACE FINISH ON EMISSIVITY

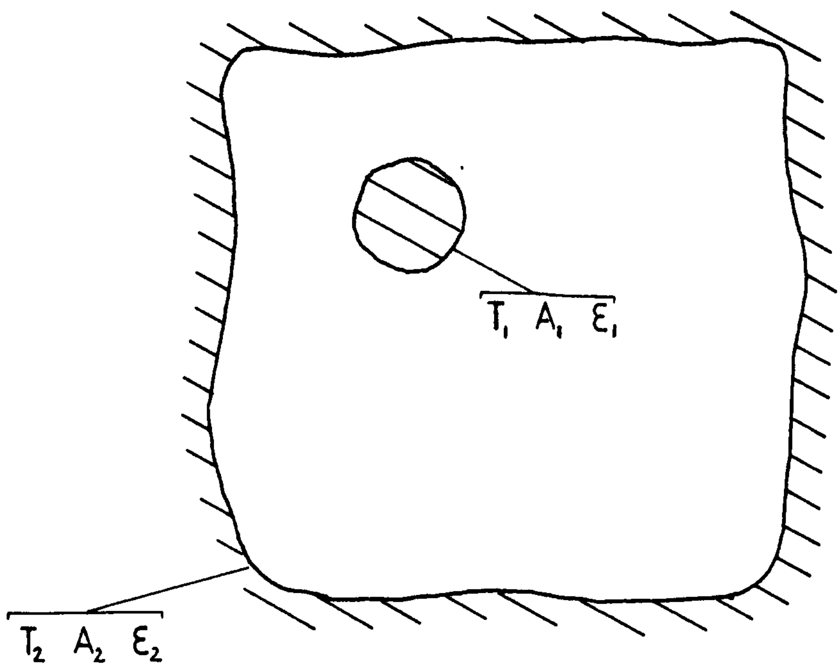


FIG 5·4b RADIATIVE HEAT EXCHANGE

5.5 HEAT TRANSFER AT WET SURFACES

Before describing heat transfer at a wet surface it is necessary to define the dew point temperature. The dew point temperature is the temperature to which an unsaturated atmosphere must be cooled at constant pressure for it to become saturated and for condensation to begin.

If a wet surface temperature is higher than dew point evaporation will take place and latent heat transfer will be away from the surface. If the surface temperature is lower condensation will take place and latent heat transfer is to the surface.

The latent heat transfer is not always in the same direction as the convective or radiative heat transfer. For instance on a wet bulb thermometer latent heat transfer is away from the surface whilst convective and radiative heat transfer is toward the surface.

The rate of latent heat transfer is simply the product of the evaporation or condensation rate and the latent heat of water. The latent heat of water is about 2430 kJ/kg within the range of conditions encountered in mining. It varies slightly with temperature as follows:-

$$L = 2501 - 2.378\ t \qquad (25)$$

where t is the temperature in ^{o}C at which evaporation or condensation takes place.

Fortunately condensation and evaporation behave with similar boundary considerations as in the case of convection. If careful measurements of air moisture content are made at increasing distance from a surface at which condensation or evaporation is taking place freely, variations in vapour pressure will be as shown in figure 5.5a.

The thickness of the boundary layer is almost equal to that of the thermal boundary layer. Also it has been verified experimentally that the rate of condensation or evaporation is directly proportional to the convective heat transfer coefficient (h_c). The driving force of latent heat transfer is vapour pressure just as temperature is the driving force for direct heat flow. Thus in the case of condensation and evaporation, when enough 'free' water is available

$$\text{Rate of condensation (g/s)} = h_c\ A\ [0.7\ (\phi\ e_{db} - e_s)/P] \qquad (26)$$

where

P = Barometric pressure (k Pa)

e_s = Saturated vapour pressure at surface temperature (k Pa)

e_{db} = Saturated vapour pressure at dry bulb temperature (k Pa)

ϕ = Relative humidity (%)

A = Area (m^2)

Multiplying by the latent heat gives the rate of heat transfer due to condensation

$$Q_L = A.h_c.L\ [0.7\ (\phi\ e_{db} - e_s)/P] \qquad (27)$$

This equation may also be reversed and used for evaluating evaporative heat transfer. It should be noted that $\phi.e_{db}$ being equal to the vapour pressure of the air e can be calculated using

$$\phi\ e_{db} = e = e_{wb} - a.P\ (t_{db} - t_{wb}) \qquad (28)$$

where a is the psychrometric constant (see Chapter 3.7).

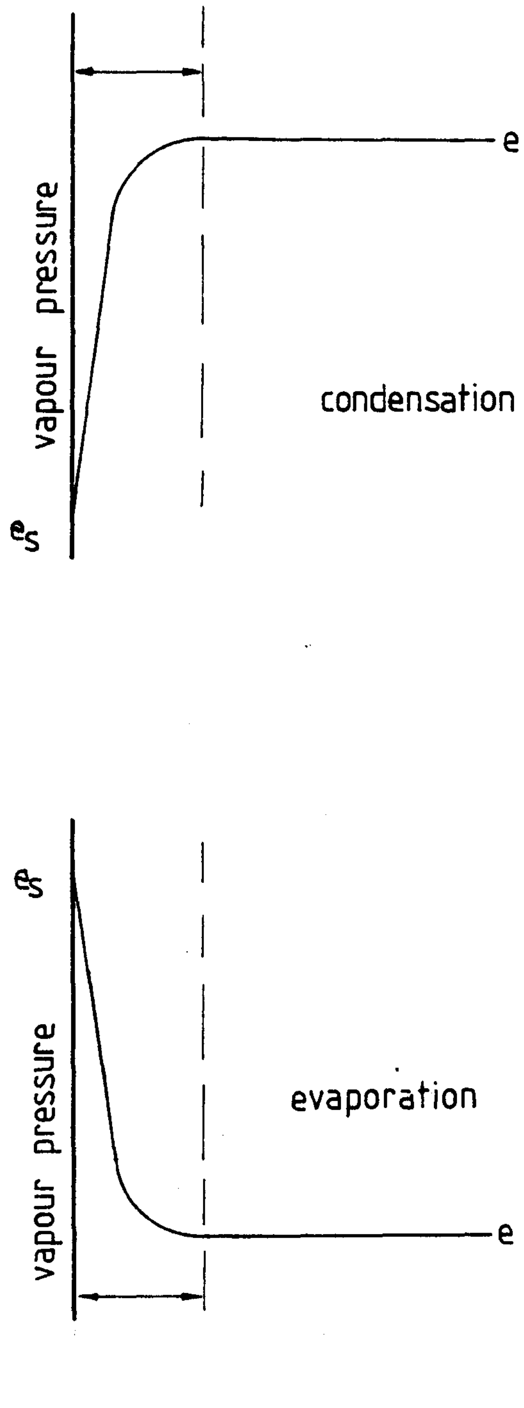


FIG 5·5a VAPOUR PRESSURE BOUNDARY LAYER.

CHAPTER 6

PRELIMINARY UNDERGROUND SURVEYS

CHAPTER 6

PRELIMINARY UNDERGROUND SURVEYS

6.1 INTRODUCTION

This chapter describes the underground work carried out at Bentinck and Hickleton collieries in the early stages of the project. It was not aimed specifically at evaluating the conveyed coal heat load, but was carried out to give experience in selecting equipment and making and interpreting measurements. It provided also the opportunity to gain a practical understanding of the magnitude and characteristics of the many interactive heat sources encountered underground.

6.2 BENTINK K76's TEMPERATURE SURVEY

6.2.1 Description of district

This site was chosen for investigation due to its potentially hot environmental conditions. At the time of the surveys Bentinck K76's was 6.7 km from the shafts and advancing into a previously unworked part of the Blackshale seam. The face was 2.1 m high, 210 m long and had a planned high advance rate of 640 m/year through 3 shift workings. The tailgate was 830 m long and dry, but badly crushed, probably due to old workings above. The main-gate was 790 m long and in good condition. The depth of cover was 635 m resulting in a VRT of 29°C taken from the local VRT borehole profile.

The installed power of machinery on the district was of the order of 1.2 MW comprising of the equipment listed in table 6.2a and positioned as is illustrated in figure 6.2b.

Table 6.2a

Electrical equipment installed on K76's district

Position	Description
Maingate, outbye	2 x 112 kW drive unit for 42" conveyor
Maingate, inbye	37 kW drive for tandem conveyor 49 kW stage loader motor 49 kW Eimco bucket 37 kW pump [500 kVA transformer]
Face	2 x 112 kW AFC motors 2 x 150 kW shearers
Tailgate, inbye	2 x 37 kW pumps 125 kW Dosco ripping machine [500 kVA transformer]

Note, panzer is the commonly used name for the armoured flexible conveyor, (AFC) on the face.

6.2.2 Temperature survey

A temperature survey was made of K76's district on the morning shift 30th November 1978. As the air temperatures on a high production district tend to rise through the week the day of the survey, Thursday, was chosen to evaluate what was expected to be as near as possible the worst conditions encountered. Wet and dry bulb temperatures were measured at points around the district shown in Figure 6.2b. The survey was carried out as quickly as possible to provide a quasi steady state picture of what is an

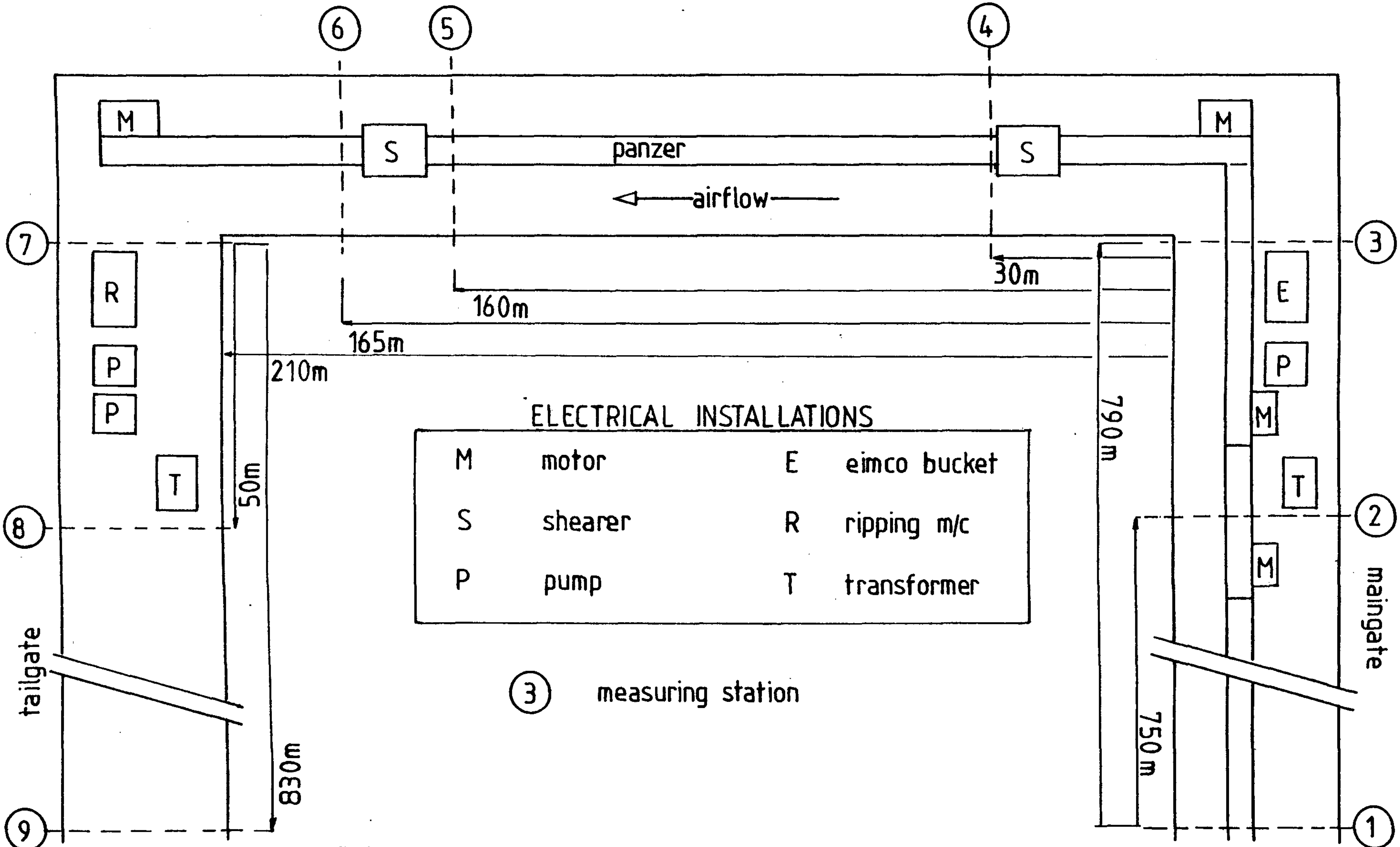


FIG 6·2b BENTINCK K76's DISTRICT TEMPERATURE SURVEY.

unsteady situation. Later work will be described that illustrates how the conditions at a point vary with time. As shown in the diagram the measuring stations were chosen to include longer lengths of airway where strata heat or a linear heat source such as a conveyor would be detected, or shorter distances to 'bracket' a spot source such as a machine or concentration of machines. The wet and dry bulb temperatures of the air were measured using an 'Assman' hygrometer. The air quantity flowing around the district was measured using a vane anemometer traversed using the approved procedure described in 'Ventilation in coal mines' [19]. A precision aneroid barometer was used to measure the air pressure. This was taken at the first and last stations of the survey. Using the mean value gave acceptable accuracy for this survey. All survey measurements were taken by two operators and repeated if any pair of readings did not agree. At the time of the survey all machinery was working.

Given a wet and dry bulb temperature and air pressure the state of the air is fully defined as described in Chapter 3 (Psychrometry). These three properties were entered as data into the pyschrometric program "PS" on the Nottingham University Mining Engineering Department minicomputer and hence, for each station in turn on the survey all psychrometic data was specified. To evaluate the heat and moisture changes as the air passed successive stations around the district the sigma heat and moisture content for each station were tabulated and the differences per kilogram of air noted. The air quantity measured by the anemometer traverse at station 1 multiplied by the simultaneous local true density of the air gives the mass flow rate of air into the district. No air leakage was detected so, assuming conditions

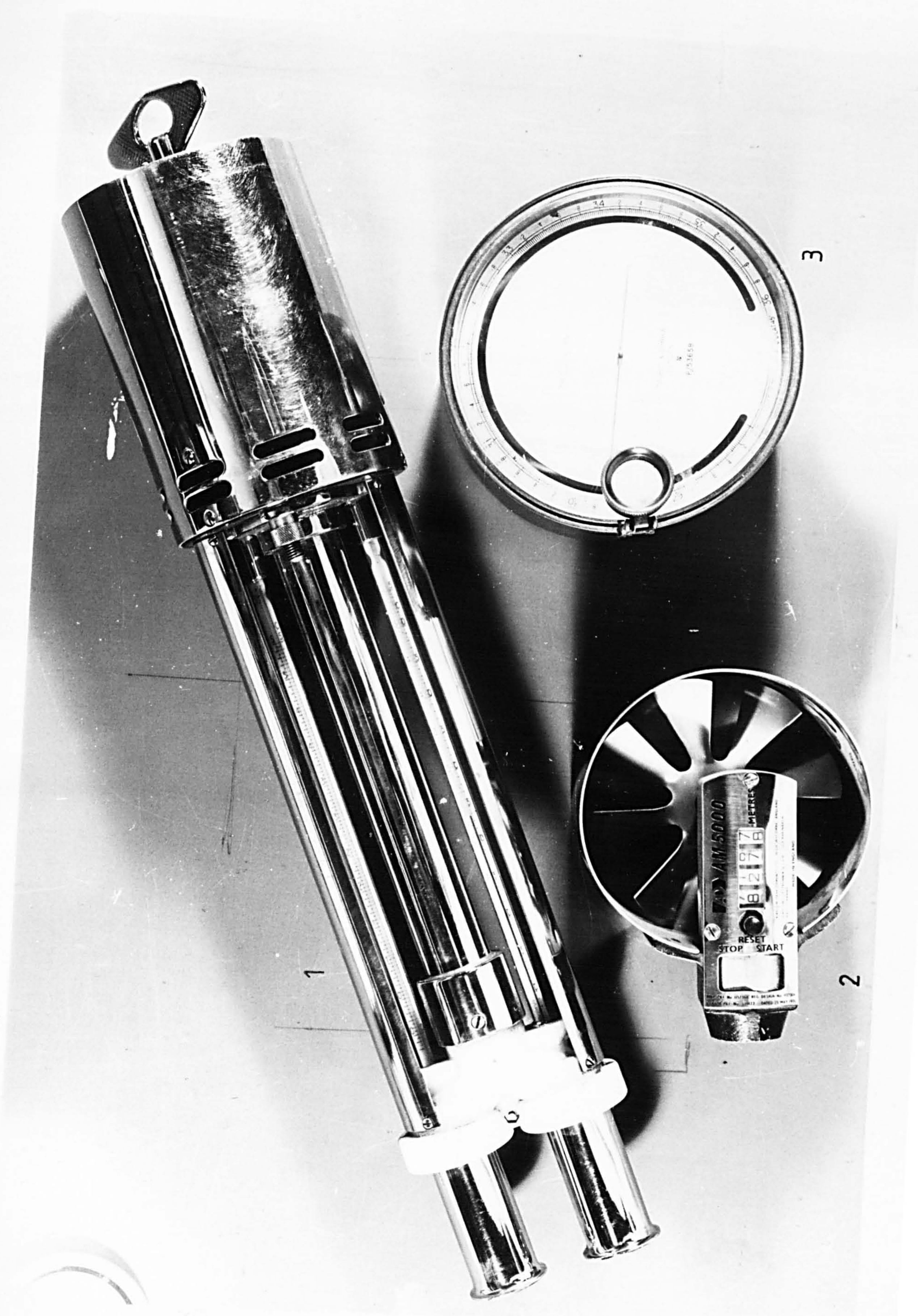
Plate 2

Ventilation survey instruments

1 Assman hygrometer

2 Vane anemometer

3 Precision aneroid barometer


1
2
3
AM 5000
METRES
RESET
STOP START

to be close to steady state there would be a constant mass flow rate at all stations.

Multiplying the mass flow by the differences per kilogram of air for each section gives the heat and moisture pick up around the district. The results are shown in table 6.2c. Figure 6.2d shows the heat and moisture pick ups superimposed on the district plan.

6.2.3 Sample calculation and results

Sample calculations

Station 1 Observed readings

Wet bulb temperature	23.7°C
Dry bulb temperature	27.9°C
Air pressure	104.77 k Pa
Air quantity	10.16 m^3/s

From computer program "PS":-

Air density	1.2004 kg/m^3
Sigma heat	67.55 kJ/kg
Moisture content	16.11 g/kg

To find air mass flow around district:-

Air volume flow x Air density = Air mass flow

$$10.16\ m^3/s \times 1.2004\ kg/m^3 = \underline{12.196}\ kg/s$$

Station 2 Observed readings

Wet bulb temperature	25.2^{o}C
Dry bulb temperature	28.3^{o}C
Air pressure (from 1)	104.77 k Pa

From computer program "PS":-

Moisture content	18.30 g/kg
Sigma heat	73.23 kJ/kg

Moisture gain between stations 1 and 2:-

(Moisture content at 2 - Moisture content at 1) x Air mass flow

(18.30 g/kg - 16.11 g/kg) x 12.196 kg/s

= 26.71 g/s

Heat gain between stations 1 and 2:-

(Sigma heat at 2 - Sigma heat at 1) x Air mass flow

(73.23 kJ/kg - 67.55 kJ/kg) x 12.196 kg/s

= 69.27 kW

Table 6.2c

Bentinck K76's temperature survey

Station	Wet bulb Temperature (°C)	Dry bulb Temperature (°C)	Sigma heat (kJ/kg)	Heat gain (kW)	Moisture content (g/kg)	Moisture gain (g/s)
1	23.7	27.9	67.55		16.11	
				69.27		26.71
2	25.2	28.3	73.23		18.30	
				38.66		11.46
3	26.0	29.2	76.40		19.24	
				9.87		3.05
4	26.2	29.4	77.21		19.49	
				60.86		25.49
5	27.4	29.4	82.20		21.58	
				12.81		3.29
6	27.6	29.6	83.05		21.85	
				31.95		13.41
7	28.2	29.6	85.67		22.95	
				43.66		8.90
8	29.0	31.4	89.25		23.68	
				-32.93		-11.83
9	28.4	31.0	86.55		22.71	
			TOTAL	234.15		80.48

Air pressure 104.77 k Pa

Air quantity 10.16 m^3/s

Air density 1.20039 kg/m^3

Air mass flow 12.196 kg/s

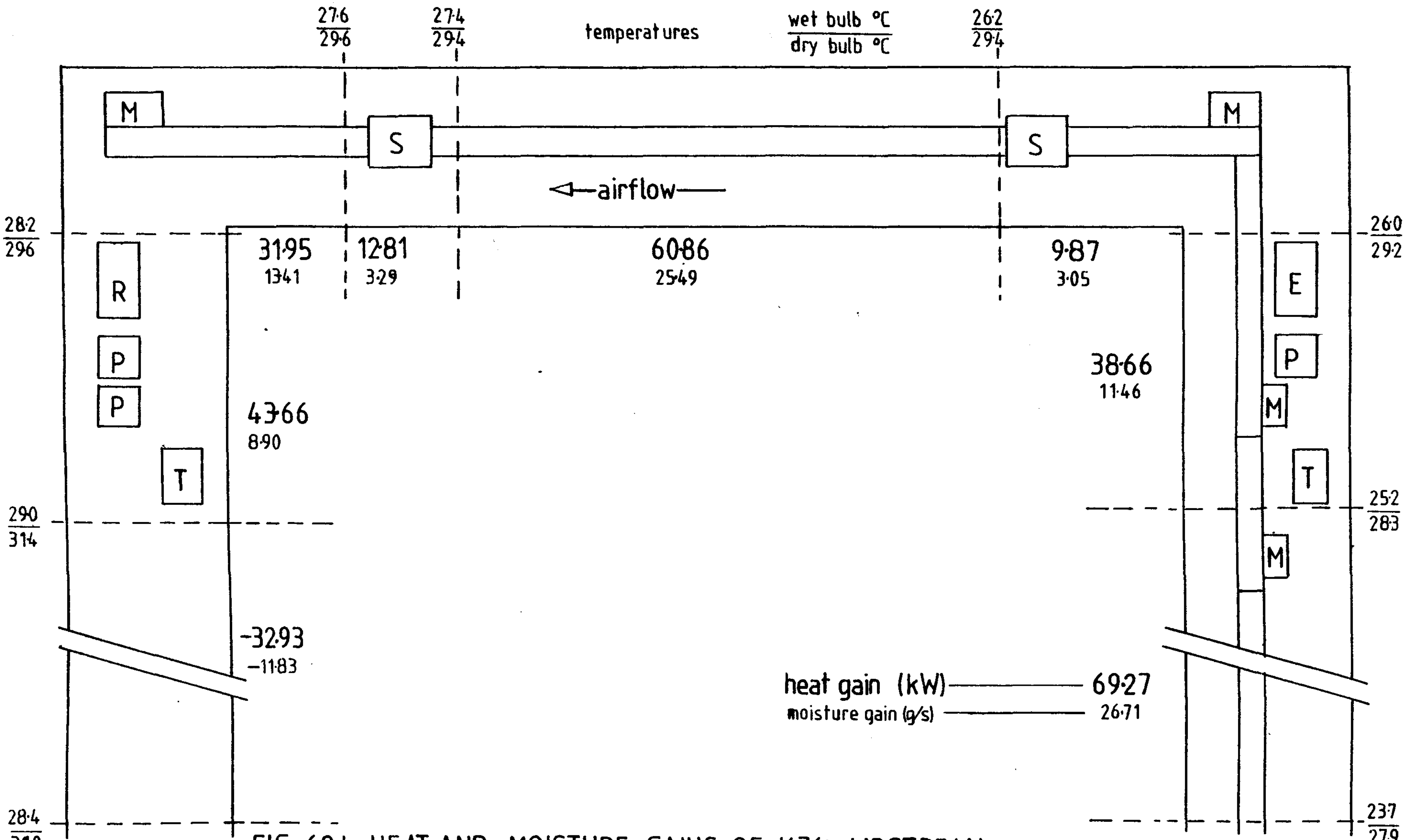


FIG. 6·2d HEAT AND MOISTURE GAINS OF K76's AIRSTREAM.

6.2.4 Discussion

This district certainly provided the hot conditions expected due to the reasons mentioned in section 6.2. At the time of the survey, late November, surface temperatures were hovering around $0^{o}C$. It is noticeable that the air entering the district at station 1 was already quite warm, $27.9^{o}C$. This could be due to strata heat or machine activity in the intake which is discussed in detail more fully in later sections.

The maingate may be split into two sections. The first 750 m from stations 1 to 2 containing only the conveyor and a small tandem conveyor motor produced heat at a relatively low rate of 69 kW and there was a correspondingly low moisture pick up. The main heat source along this length of road would be the strata and the already high dry bulb temperature of $27.9^{o}C$, near VRT would result in a low 'driving potential' for heat transfer. The importance of keeping intake air as dry as possible should be mentioned here. If the air has water evaporated into it due to say, excessive use of sprays, leaking pipes or uncovered drains the air dry bulb temperature falls. Naturally this will result in greater heat flow to the air due to enhanced heat transfer and a greater temperature gradient in the strata surrounding the airway. The moisture gained by the air would be from both the exposed strata and coal on the conveyor. Although the roadway appeared to be in a totally dry condition some moisture could have been evaporating from them. Due to the vast surface area available in 750 m of airway the net mositure contribution could be significant despite the small amount of evaporation per unit area. Moisture evaporation from coal on the conveyor was recognised as a source, but at this preliminary stage of the study its contribution could not be evaluated.

The section of maingate from stations 2 to 3 contained a concentration of machinery. The much higher heat gain per unit length can be clearly seen (figure 6.2d). This section being only 40 m long. From the previous section an indication of the strata heat flow can be obtained and by differences it may be shown that most of the heat in this section may be attributed to the machinery. It is noticeable that the heat gain of 39 kW is only a small fraction of the installed power of 135 kW (not including the transformer). This is a typical situation as the machines tend to run intermittently and rarely at full load. A similar situation exists on the first section of the face with the rated power of the shearer and AFC drive totally 262 kW, but the heat gain being only 10 kW. This machinery had only just started as the temperature measurements were taken here so despite the machinery producing more than 10 kW most of the heat produced would be absorbed internally in the machines thermal capacity.

Mining machinery is extremely substantial and the potential for large amounts of internal heat storage exists. This storage effect which currently is to a large extent unevaluated does provide a moderating effect by damping out the fluctuations in heat production of machinery by absorbing heat as a machine works and releasing it as the machine cools after working. Until these mechanisms are more fully understood they should not feature in design calculations for ventilation systems which should always consider the worst conditions that might arise. If a machine is run steadily for a long period, such as on a high production face it will when it gets sufficiently hot, dissipate heat at the rate it produces it.

The section along the face, stations 4 to 5 had a heat gain of 61 kW and a high moisture gain of 25 g/s. At first glance it would appear that the heat must all be from the strata as the section contains no motors, but machine heat is still dissipated here. Figure 2.4a shows how a machine dissipates heat from its motor-drive unit and along its working length as work against friction is performed. Two such cases exist here. The useful work performed by the AFC drive is used to overcome friction as it moves coal along the face. Also mechanical energy is degraded to heat through fluid friction and heat is transferred along and dissipated by the fluid in the hydraulic lines on the face.

The pumps in the face ends provide hydraulic power for the face chock supports via an extensive pipe system. The hydraulic fluid is heated in the pumps and by turbulence in valves and pipes. The pipework becomes hot and heat is dissipated from it even when no work is being done by the rams and chock legs. Heat is also dissipated as is a large amount of the moisture from the cut coal on the conveyor and that left temporarily in the machine track. The evaluation of heat from cut coal is the main objective of this study. At this stage its potential was recognised and plans were made for a more detailed investigation described later.

The amount of heat flowing into an airway from the strata is a function, amongst other things, of time since exposure. After an airway is driven the heat gain from the strata decays from a maximum level asymptotically to a base level over a period of up to 2 years. A coal face advancing into virgin strata may approach the VRT if its advance rate is high enough to give the coal body little time to cool. Also new ground is constantly being exposed, keeping heat emission at a high rate. Consequently rapidly advancing coal faces release large amounts of heat into the ventilating airstream. This section of face is a typical example.

The section from stations 5 to 6 contained the barrier shearer in 5 m of face. Only 12 kW of heat gain were measured over a 150 kW machine. The most likely explanation of this imbalance is the storage effects described earlier. Other relevant factors were that the machine was ploughing back which needs very little effort compared with cutting, also badly chosen measuring points did not give the air time to mix thoroughly downstream of the machine.

The last section of face 45 m between station 6 and 7 contained the tail motor unit for the AFC. This section had a heat and moisture gain per unit length slightly higher, but comparable with section 4-5. The higher figure probably arising from electric motor heat.

The first 50 m of tailgate, containing a ripping machine, pumps and transformer had a heat gain comparable to that containing the maingate machinery, and similar explanations apply.

As the air passed along the remaining 780 m of tailgate, between stations 8 and 9, it lost heat to the strata and the moisture content fell slightly. No evidence of condensation was seen. A similar but reversed situation to the maingate provides a likely explanation. Heat flow into the strata is due in this case to the machinery raising the dry bulb temperature above the

VRT. When the machinery is not used, eg at weekends, the air leaves the face at a temperature below VRT resulting in heat flow from the strata into the air. Conditions encountered in British collieries usually result in a net heat flow from the strata into the air, although if for short periods in very hot conditions the situation is reversed. In some mines the very low VRTs heat flow is usually from the air to the strata.

The survey was timed to observe what was likely to be the worst possible conditions and produced a wet bulb temperature of over 27^{o}C half way along the face, 28^{o}C at the tailgate rip and 29^{o}C at 50 m outbye. Reference to Chapter 4 (Heat Stress) will illustrate the risks involved.

This survey was a basic temperature traverse at almost a fixed point in time. To extract more information a survey may be designed to highlight a particular aspect or heat source by measuring changes with time at selected locations. Examples of this will be described.

The temperatures around a longwall district vary considerably with time and the variation of temperature with time can reveal information about the dynamic interaction of the various heat sources. This aspect is described in the next section.

6.3 CONTINUOUS TEMPERATURE RECORDING ON BENTINCK K76s DISTRICT

6.3.1 Introduction and method

It is known that the temperature at a fixed place in a mine is rarely steady. Near the shafts it may be expected to vary with changing surface conditions. Further inbye the mines heat sources have more effect. The characteristics of the mining system and amount of activity have a large effect on some of these heat sources and hence cause climatic variations.

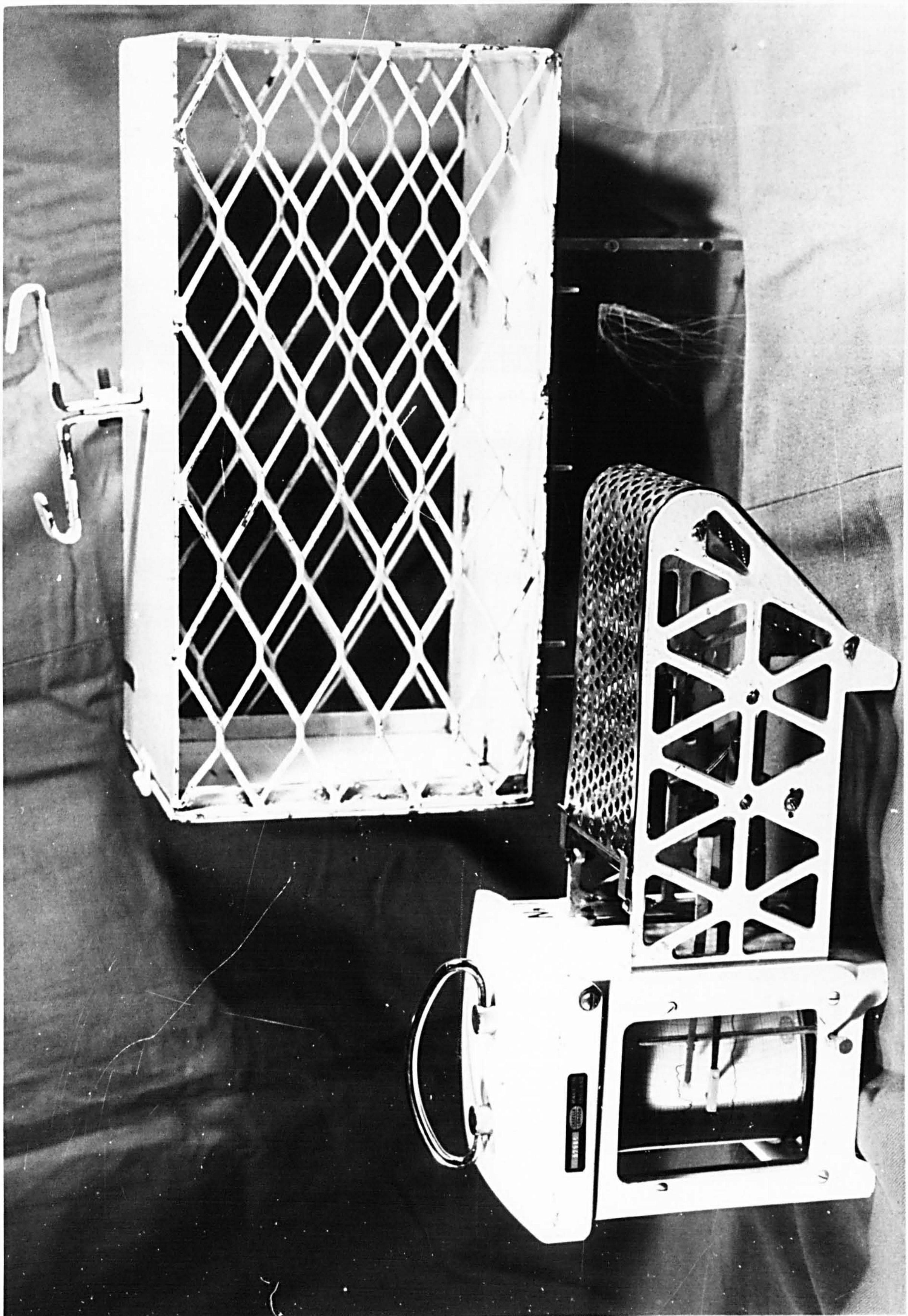
A continuous record of the mine air conditions over a period is usually made using an instrument known as a thermohygrograph. The thermohygrograph continuously records dry bulb temperature and relative humidity onto a moving chart.

As stated in Chapter 3 (Psychrometry) the state of the air is specified fully by three independent characteristic properties, usually wet and dry bulb temperatures and pressure. Fortunately in many places in mines the pressure varies by a small enough degree, within about 5%, to assume it is constant otherwise continuous pressure record would need to be made. It also has to be assumed that the air flow rate is steady if heat gain calculations are to be made or a recording anemometer must also be installed.

The dry bulb temperature is measured using a bimetallic strip driving a recording pen via a linkage which allows compensation for non linearity of the strip and adjustments of range. Although a wet bulb temperature record would be desirable, continuous recording of this is extremely difficult due to problems of water supply and bulb aspiration. Relative humidity though is easily recorded by using a pen connected by a linkage to some material sensitive to humidity. In the case of this instrument human hair. From the known relative humidity and dry bulb temperature the wet bulb temperature may easily be evaluated or read from the slide rule supplied with the instrument.

Plate 3

Thermohygrograph and protective case



The records of temperature and humidity are made onto a chart carried on a clockwork revolving drum whose gearing may be changed to give one revolution in either one day or one week. For this study all records were made onto one week charts.

Due to the delicate linkages thermohygrographs have given trouble in the past due to ingress of dust. Recognising this problem Middleton [20] constructed perspex covers which did not affect the performance of the instrument but gave a high degree of protection from dust. Also a steel cage to allow mounting in a roadway and protect from tampering was used on each instrument. With dismantling and cleaning before installation, careful calibration and adjustment, and checks using an Assman hygrometer whilst underground, the instruments could be relied upon to give readings within 0.2°C and 1% relative humidity.

6.3.2 Installation at Bentinck

At the time of this survey three thermohygrographs were available. To gain the most useful information the placing of the instruments was considered and to correlate with the first temperature survey the following sites were chosen (figure 6.2b).

(i) Instrument 1, maingate outbye at station 1.

(ii) Instrument 2, maingate inbye at station 2 before maingate electrical equipment.

(iii) Instrument 3, tailgate 50 m outbye of rip at station 8.

The instruments were installed in November 1978 and removed in January 1979. This allowed observation of ordinary production conditions and also the effect of a break in production over the

Christmas holiday. The instruments were visited once a week for checking and chart changing. Occasionally a weeks record was lost on an instrument when failures occurred due to jamming of pens or vibration, but generally they performed well. Figure 6.3a shows a typical set of charts. The instrument charts were changed on Wednesdays as this coincided with the Colliery Ventilation Officers visit to the district. Consecutive sets gave a continuous record and for presentation a Monday to Monday cycle was adopted as this illustrates more clearly a working week. As mentioned in 6.3.1, providing pressure and air flow rate are known, any other psychrometric data, including heat and moisture gains, may be derived from the dry and wet bulb temperatures.

Many different charts were drawn to show variations in heat and moisture content and gain etc. However wet bulb temperature records as shown in figure 6.3b combined with the original instrument charts give an excellent portrayal of the situation without presenting too much information simultaneously.

6.3.3 Discussion

The instrument charts and wet bulb temperature record are typical of all the working weeks observed apart from those during and after holidays. They show clearly a daily and weekly cycle and provide some indications of the contributions of various heat sources.

The dry bulb temperature varied by only a small amount at the two maingate stations, 3.5^{o}C outbye and 3.0^{o}C inbye. This illustrates the stabilising influence of the strata. The airstream to K76s district had to travel about 7 km from the surface. Most of

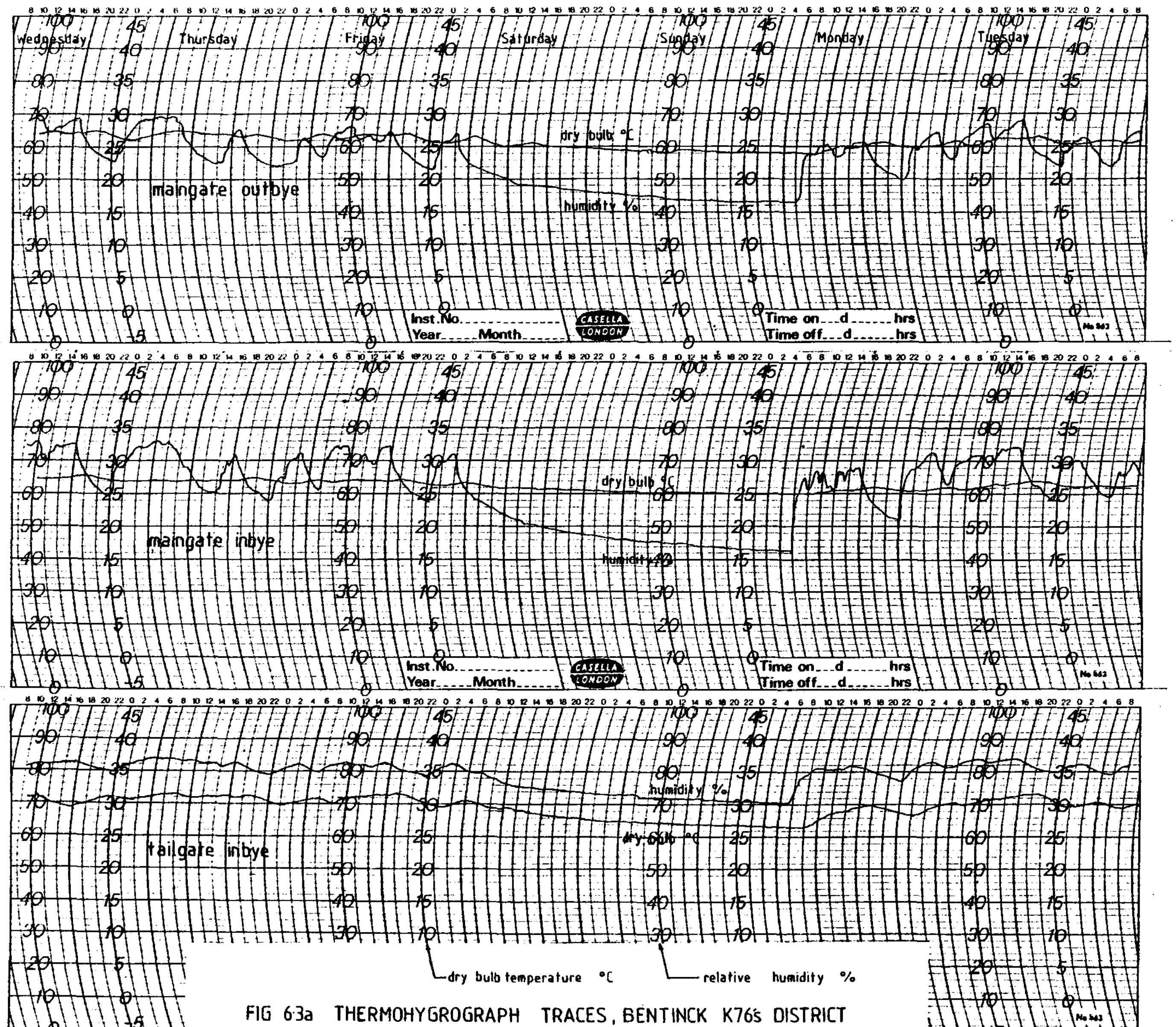


FIG 6·3a THERMOHYGROGRAPH TRACES, BENTINCK K76s DISTRICT

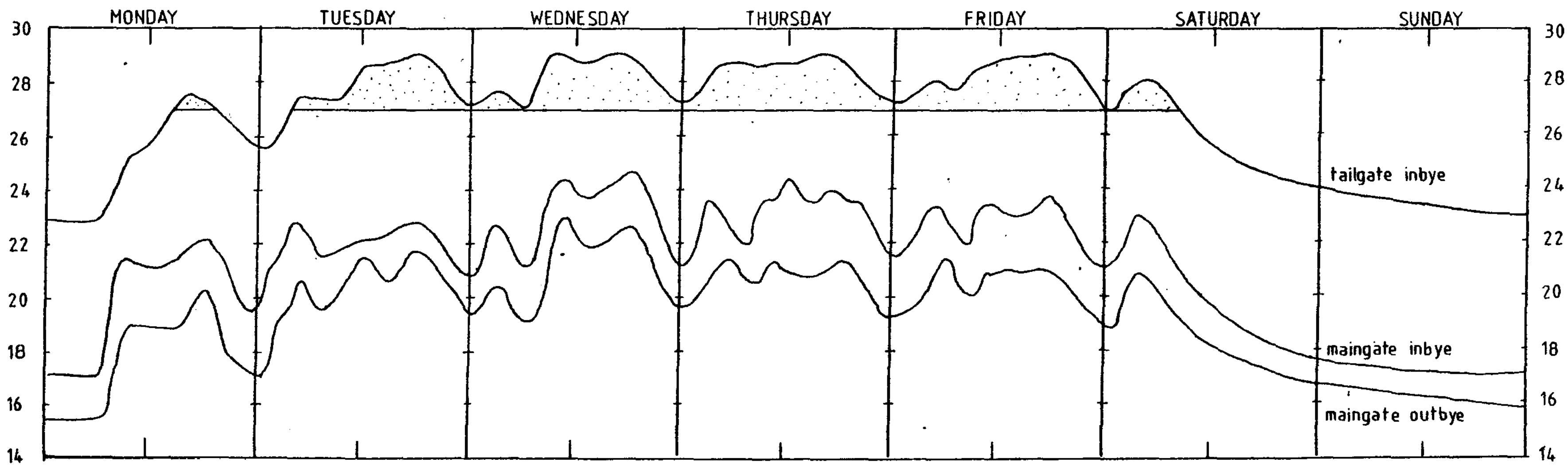


FIG 6·3b WET BULB TEMPERATURE RECORD , BENTINCK K76s DISTRICT this distance was through dry roadways which were well established. The air then passed through a large mass of rock with a very high thermal capacity and at an almost fixed temperature. There were fluctuations though which would be caused by machinery and moisture evaporation. After the air had traversed the face passing over machinery operating intermittently its dry bulb temperature was less steady with a $6^{o}C$ fluctuation over the working week.

The humidity and wet bulb records are of similar shape. This is because for a constant dry bulb temperature the wet bulb temperature is proportional to the relative humidity. Figure 6.3b illustrates the wet bulb temperature record of all three stations for a typical week.

Starting on Monday morning the district had settled almost to a steady wet bulb temperature and hence heat load. The first shift started at 6.00 a.m. and the instant sharp rise in humidity, temperature and heat in the air can be easily identified. The increased heat make over the base level would be almost entirely due to machinery as the strata responds much more slowly. Each shift of the week can be picked out on the weekly chart. The fact that the three shifts were not evenly spaced over 24 hours is also detectable. After a working shift the temperature begins to drop back to a base level until the machinery starts for the next shift. The mining activity was such that the temperatures could not return to base level between shifts. This indicates a storage effect which is thought to be due to the large quantities and thermal capacities of steelwork particularly on the face and also that of thin skin of strata around an airway.

The storage and hence damping effect produces a slightly smoother temperature profile for the instrument after the face, (tailgate inbye). The storage effect and continued activity result in a gradual worsening of conditions as the working week progresses. Conditions can be seen to get gradually hotter each day until equilibrium is reached, Thursday and Friday. At this stage the machinery, steelwork and possibly small amount of strata involved in storage are in equilibrium over a 24 hour period. If daily production was held at that level one would expect the conditions to be bad, but stable. On figure 6.3b a line has been marked at the 27^{o}C wet bulb to show the possible heat strain risk periods. With wet bulb temperatures consistently above 27^{o}C after Tuesday morning at the tailgate the workers, particularly those on heavy tasks could have been at risk.

When mining stopped at 6.00 p.m. on Saturday conditions could be seen to decay back towards the base heat load. Given that the machine heat source ended abruptly at 6.00 a.m. and that over a period strata heat is fairly stable the net heat release from then to steady state on Sunday would also be due to storage. The effects of storage have only recently been recognised and following the MRDE study of mine climate [9] more research is envisaged. The base heat load reached on Sunday would of course be due only to the strata.

Although the heat sources on the face produce the final escalation to intolerable conditions it is noticeable on figures 6.3a and 6.3b that even well outbye at the start of the maingate that the effects of production are recorded. It has been accepted in the past that the massive concentrations of machines on high production faces responsible for the hot conditions, but this effect is seen in intake air 800 m outbye. This short term fluctuation corresponding to production periods could only be due to machinery in the intake airways. Mainly conveyors and coal on the conveyors which would be an ideal moisture source to give the fluctuations in humidity. To assess the potential of the heat made from coal on conveyors a separate investigation was necessary.

6.4 SURVEY OF CONVEYED COAL TEMPERATURES

6.4.1 Method

Many attempts have been made to measure the temperature of coal on conveyors by various workers eg Middleton [20], MRDE [9], Voss [21]. Inherent measurement problems are caused by the nature of conveyor systems, intermittent operation, and the solid granular coal which is not at uniform temperature such as fluid in a pipe.

The temperature measurements should ideally be taken on one batch of coal on its journey outbye to remove the possibility of measuring the temperature of coal which started at different temperatures. Due to the speed of conveyors and the number of transfer points this would mean many personnel stationed at various points on the chosen conveyor system. If one length of conveyor was chosen for study the numbers could be reduced, but two teams and sets of equipment would be needed. Workers from MRDE have attempted this [9].

The temperature within the coal bed on a conveyor is known to be non uniform. It cools quickly at the surface and retains heat inside. Also size variations, particularly large lumps, cause non-uniformity of temperature. Problems of choosing the sample are apparent as are those of collecting the sample chosen on a fast moving conveyor.

Several measuring vessels have been used in the past ranging from thermos flasks [9],[20],to buckets. Ideally the measuring vessel should be robust, easy to fill, well insulated and make no heat contribution itself. The speed of filling could be crucial and this is the main pitfall of a thermos flask. The actual instrument used to take the temperature should be accurate and responsive. It should also be robust or protected during filling.

Due to the small numbers of helpers available and the tight budget the following system was adopted.

The investigation was timed to take place as the machine was cutting before 'snap time' in a period of stable production. This gave the best opportunity for uniformity of size and starting temperature. Communications with the face were maintained as first stations were visited so it was confirmed that production was steady. It was noted on previous visits that the coal produced during cutting was of a fairly small and even size whereas whilst the ploughing back was in progress the size of coal varied more and its temperature was not uniform due to lying on the face for an unspecified period.

To overcome the problems of actually collecting a sample measurements were made at transfer points. At a transfer point the coal is well mixed and a sample taken from the receiving belt by sliding a scoop into the edge of the coal bed provided a sample more safely than trying to catch coal in the chute. Some unpleasant surprises occurred trying this when large lumps came over.

The actual sample vessel consisted of a polystyrene lined wooden box as shown in figure 6.4a. It was cheap, robust and met the thermal requirements stated earlier. In use the box, which held about 3 kg of coal, was quickly filled using a flat scoop slightly narrower than the box. The metal tube normally used to carry the thermometer being inserted down the centre during filling. The lid was then fitted and the thermometer inserted into the hole left by the tube which was carefully removed. The box was then jarred slightly to close the hole and give thermal contact between the coal and the thermometer. The thermometer was then read at 1 minute intervals until it started to fall from a maximum which occurred within 10 minutes.

Tests in controlled conditions before the survey showed this combination of insulated box and (0-50 x 0.1°C) thermometer could be relied upon to give a temperature reading with -1°C 5 minutes and -0.5°C 10 minutes after filling.

At each measuring station shown on figure 6.4b, 5 samples were taken and the mean value recorded. Filling each sample into the box took about 10 seconds and temperature measurement 10 minutes. The first 4 stations starting at the shearer on K76's face, K76's stage loader, K76's transfer and K75's transfer were all on coal only from K76's cut before 'snap time'. Subsequent stations K74's transfer to the surface were on coal from several sources.

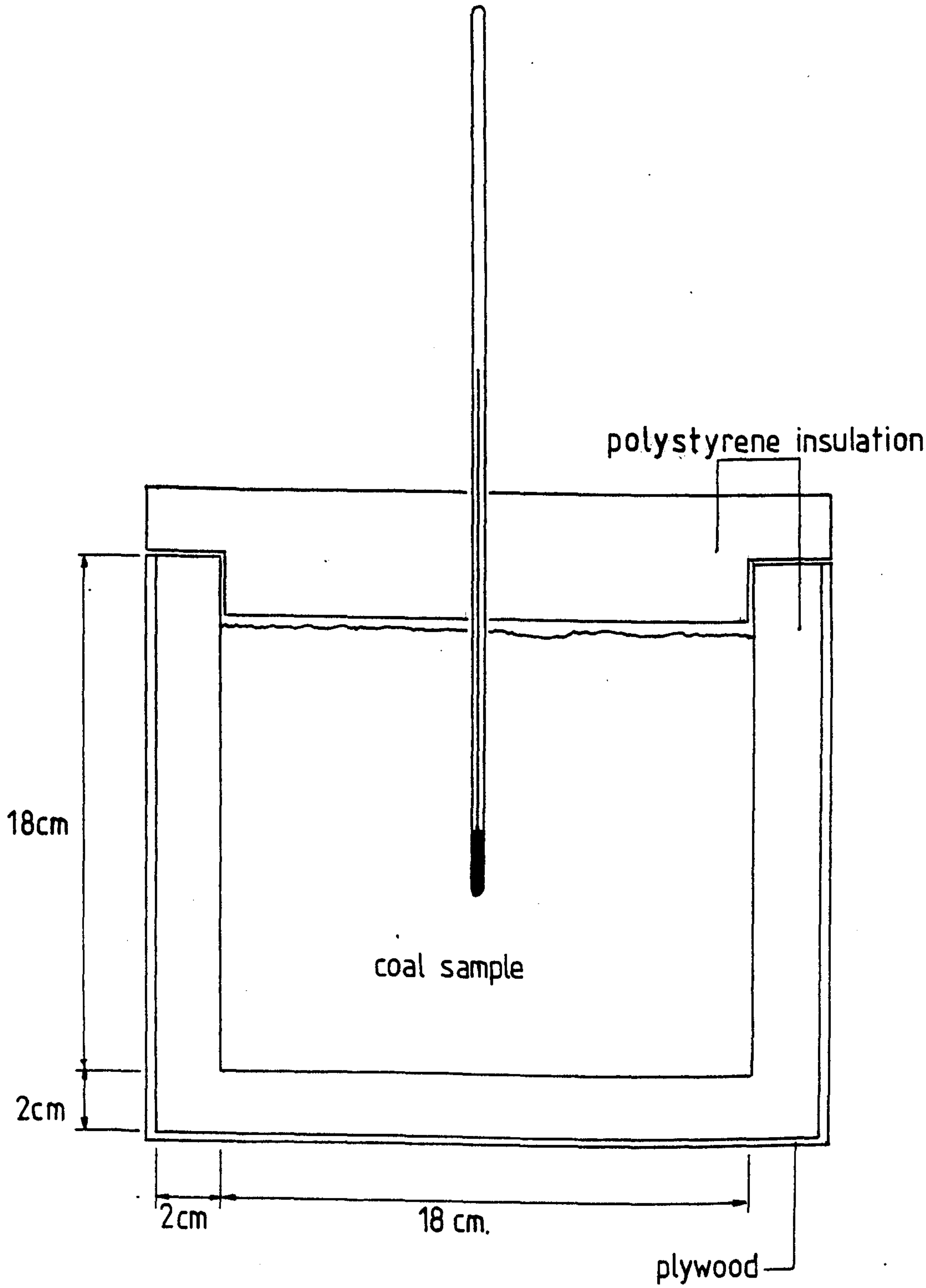


FIG 6·4a COAL TEMPERATURE SAMPLING VESSEL

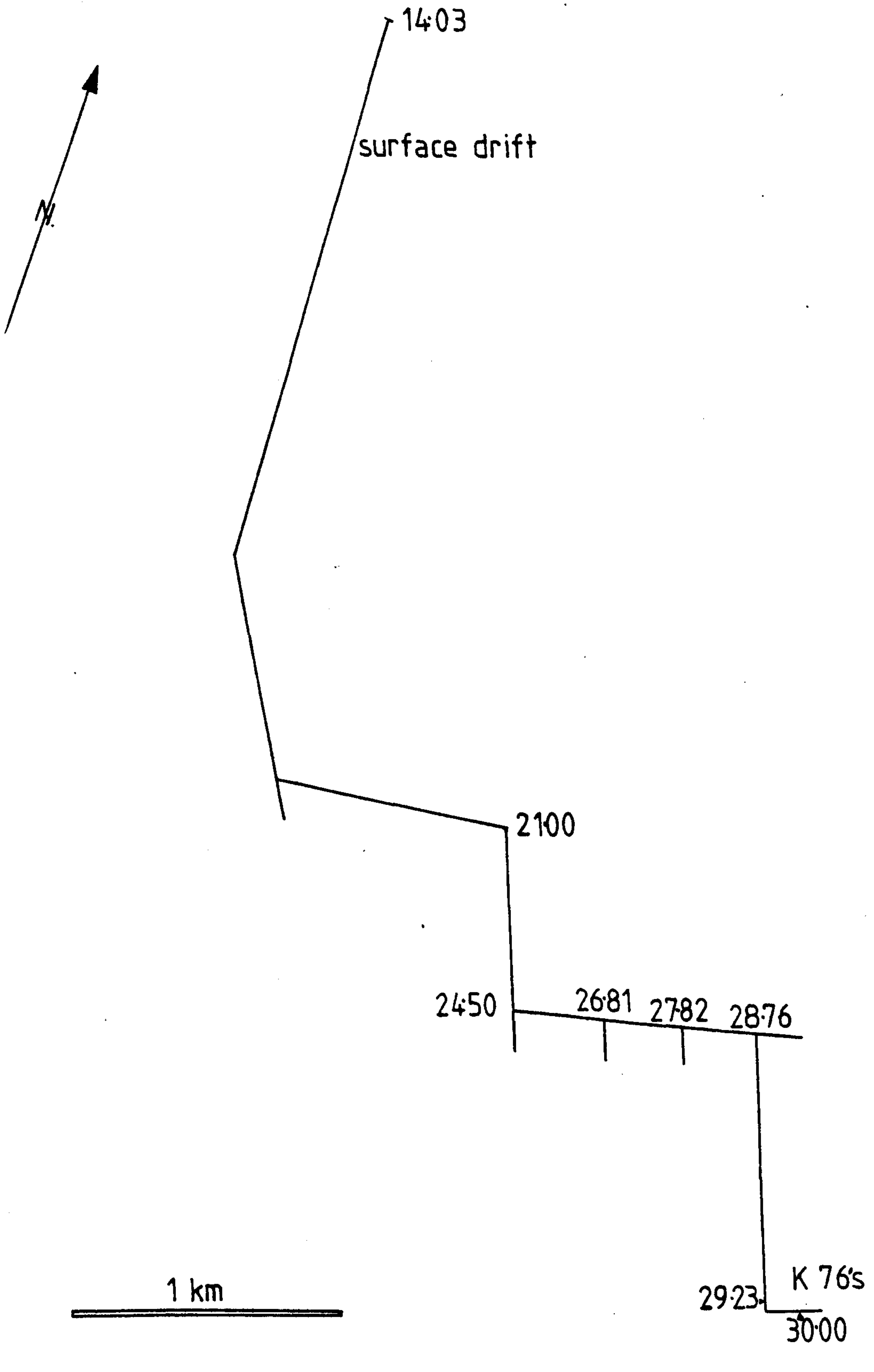


FIG. 64b CONVEYED COAL TEMPERATURES (BENTINCK)

6.4.2 Discussion and estimation of heat load

The results of a full shifts work shown on figure 6.4b give an indication of the time consuming nature of this work. Whilst it is accepted that the results of this survey are not of the highest accuracy obtainable, with so many unpredictable variables affecting the flow of coal a more careful and accurate measuring method would have taken too long.

The results were satisfactory for evaluating likely magnitude of the contribution of conveyed coal heat source. In a previous attempt at a coal temperature survey the moisture content of the coal varied considerably and the belts stopped so frequently that the survey results were meaningless. The moisture content of the coal was not checked in this survey, although its consistent appearance and nature suggested it was satisfactorily constant for a preliminary survey of this nature.

The total heat contribution of conveyed coal to an airstream is the product of its temperature fall, specific heat, and production rate.

For an average figure for the whole mine the temperature fall of the coal can be taken as $15^{o}C$. That is, face coal temperature minus temperature at drift exit. The specific heat of coal varies around 1.0 kJ/kg K to 1.15 kJ/kg K. A figure of 1.1 kJ/kg K is sufficiently accurate for this estimate. The production rate at the time of this survey was 17,000 tonnes per week run of mine (28.1 kg/s). The average heat load then over the whole week would be

$$15 \times 1.1 \times 28.1 = 464 \text{ kW}$$

The coal is not run out of the mine continuously though and the heat load whilst the conveyors are running is probably several times this. 464 kW gives the minimum figure. During peak production one face producing 4 tonnes per minute (66.7 kg/s) could give a heat load to the whole mine of

$$15 \times 1.1 \times 66.7 = 1100 \text{ kW}$$

It must be remembered that heat sources interact and the net effect after some other heat source was reduced could be much smaller. The coal produced on K76s at the peak production of 66.7 kg/s and the temperature falls measured indicate production of 56 kW on the face and 34 kW in the maingate. Reference to figure 6.2d heat and moisture gains on K76s district show these figures to be quite credible.

The uneven temperature distribution in the coal could be detected by sampling coal from different depths in the coal bed. An electronic indicating thermometer probe borrowed from MRDE showed that a thin skin of less than 2.5 cm thickness at the surface was 1.5-2.5^{o}C cooler than the interior. If the coal did not cross a transfer point no doubt this skin would provide an insulating layer to heat and moisture flow. This was investigated more fully later in the study.

6.5 HICKLETON T01's DISTRICT HEAT SURVEY

6.5.1 Description

In August 1979 an investigation was undertaken at Hickleton Colliery, NCB Doncaster Area. The purpose of the surveys and associated predictions was to assess the likely deterioration of climatic conditions on the T01s face in the Thorncliffe seam as it advanced.

On the East side of the pit, faces in the Thorncliffe seam had a history of heat problems. T01's was to be the first face in the Thorncliffe seam on the west side. At the time of the survey the face had advanced about 100 m and climatic conditions were already causing concern.

Two main surveys of temperature and airflow were carried out. One started at the surface and traversed the whole ventilation circuit to T01's and P46's districts then back to surface and the second survey was from pit bottom to pit bottom. The first survey was carried out in the summer holiday week to assess base heat conditions, the second during peak production.

At each measuring station the wet and dry bulb temperatures were taken using an Assman hygrometer and the pressure was measured using a precision aneroid barometer. The colliery ventilation officer measured the airflows using vane anemometers, usually at designated airflow measuring stations. A continuous recording of temperatures was also made using the instruments and techniques described in section 6.3 as at Bentinck Colliery. Thermohygrographs were sited at either end of T01's face, 50 m outbye and the bottom of the main west intake as shown in figure 6.5a. The temperature traces with the calculated wet bulb temperature superimposed are shown for comparison with those from Bentinck Colliery.

This survey was carried out on a more complicated ventilation circuit than at Bentinck Colliery where there was a steady airflow round a single airway. At Hickleton many airways were involved with leakage and mixing giving varying airflows around the circuit. After station 6, shown on figure 6.5a, leakage of air direct from intake to return and from P46's return resulted in much mixing and this complicated the processing of the observations.

The results of the whole survey are given in the appendix (A.2). Only a section of particular interest is given here. The intake from the pit bottom to T01's face and in the first section of return as far as T01's tailgate end were all free of mixing effects as air flows from the high pressure side of a ventilation circuit. This part of the ventilation circuit, with stations shown 1 to 6 on figure 6.5a contained the face where worst conditions were expected to occur. Also between stations 1 to 4 was a powerful conveyor system with an installed drive power of 750 kW which extended as far as the pit bottom area. The positions and ratings of the drives are shown on figure 6.5a. Face machinery rated power totalled 770 kW.

The results of the surveys carried out in static conditions at the end of holiday week and peak production conditions are summarised in tables 6.5b and c. The temperatures measured are also displayed on a psychrometric chart in figure 6.5d.

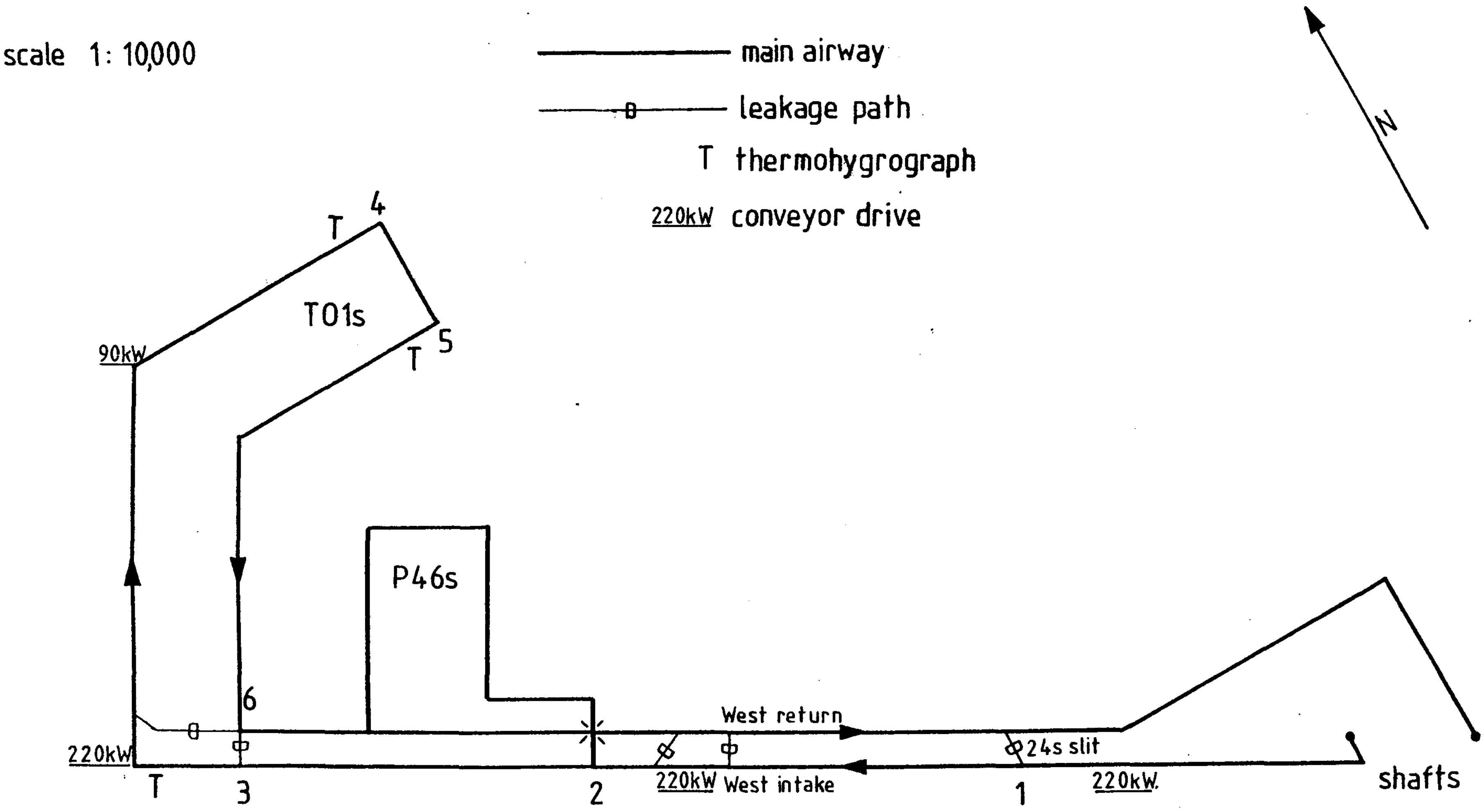


FIG 6.5a HICKLETON COLLIERY WEST SIDE

It is fortunate that conditions at station 1 were almost identical for each survey as this enables direct comparison to be made between the non operating, static conditions and production conditions.

6.5.2 Sample calculation and results

Sample calculation

Station 1, 5 m inbye of 24s slit

Observed readings

Wet bulb temperature	$16.3^{o}C$
Dry bulb temperature	$22.3^{o}C$
Pressure	108.15 k Pa
Air quantity	$31.07\ m^3/s$

From computer program "PS"

Air density	$1.268\ kg/m^3$
Sigma heat	43.07 kJ/kg
Moisture content	8.35 g/kg

Air mass flow in section

Air volume flow x Air density = Air mass flow

$31.07\ m^3/s \times 1.268 = \underline{39.4\ kg/s}$

Station 2, 5 m inbye of 46s intake

Observed readings

Wet bulb temperature	17.1°C
Dry bulb temperature	24.4°C
Pressure	107.75 k Pa
Air quantity	19.12 m^3/s

From computer program "PS"

Air density	1.255 kg/m^3
Sigma heat	45.37 kJ/kg
Moisture content	8.43 g/kg

Moisture gain between stations 1 and 2

(Moisture content at 2 - Moisture content at 1) x Air mass flow

(8.43 g/kg - 8.35 g/kg) x 39.4 kg/s

= 3.15 g/s

Heat gain between stations 1 and 2

(Sigma heat at 2 - Sigma heat at 1) x Air mass flow

(45.37 kJ/kg - 43.07 kJ/kg) x 39.4 kg/s

= 90.62 kW

Table 6.5b

Ventilation survey, Hickleton, T01s district (static conditions)

Station	Airflow	Temperature		Sigma heat	Moisture content	Heat gain	Moisture gain
	kg/s	WB°C	DB°C	kJ/kg	g/kg	kW	g/s
1		16.3	22.3	43.07	8.35		
	39.4					90.62	3.15
2		17.1	24.4	45.37	8.43		
	24.0					36.00	0.0
3		17.7	25.9	46.87	8.43		
	14.1					85.16	12.97
4		19.3	28.9	52.91	9.35		
	14.1					94.33	36.66
5		21.8	30.0	59.60	11.95		
	14.1					93.34	27.37
6		23.8	31.9	66.22	13.89		

Table 6.5c

Ventilation Survey, Hickleton, T01's district (production conditions)

Station	Airflow	Temperature		Sigma heat	Moisture content	Heat gain	Moisture gain
	kg/s	WB°C	DB°C	kJ/kg	g/kg	kW	g/s
1		16.5	22.5	43.61	8.49		
	39.4					468.46	146.9
2		20.5	25.3	55.50	12.22		
	24.0					71.76	22.56
3		21.5	26.0	58.49	13.16		
	14.1					157.36	46.95
4		24.7	29.1	69.65	16.49		
	14.1					268.18	92.64
5		29.3	32.3	88.67	23.06		
	14.1					-49.49	-20.02
6		28.6	32.2	85.16	21.64		

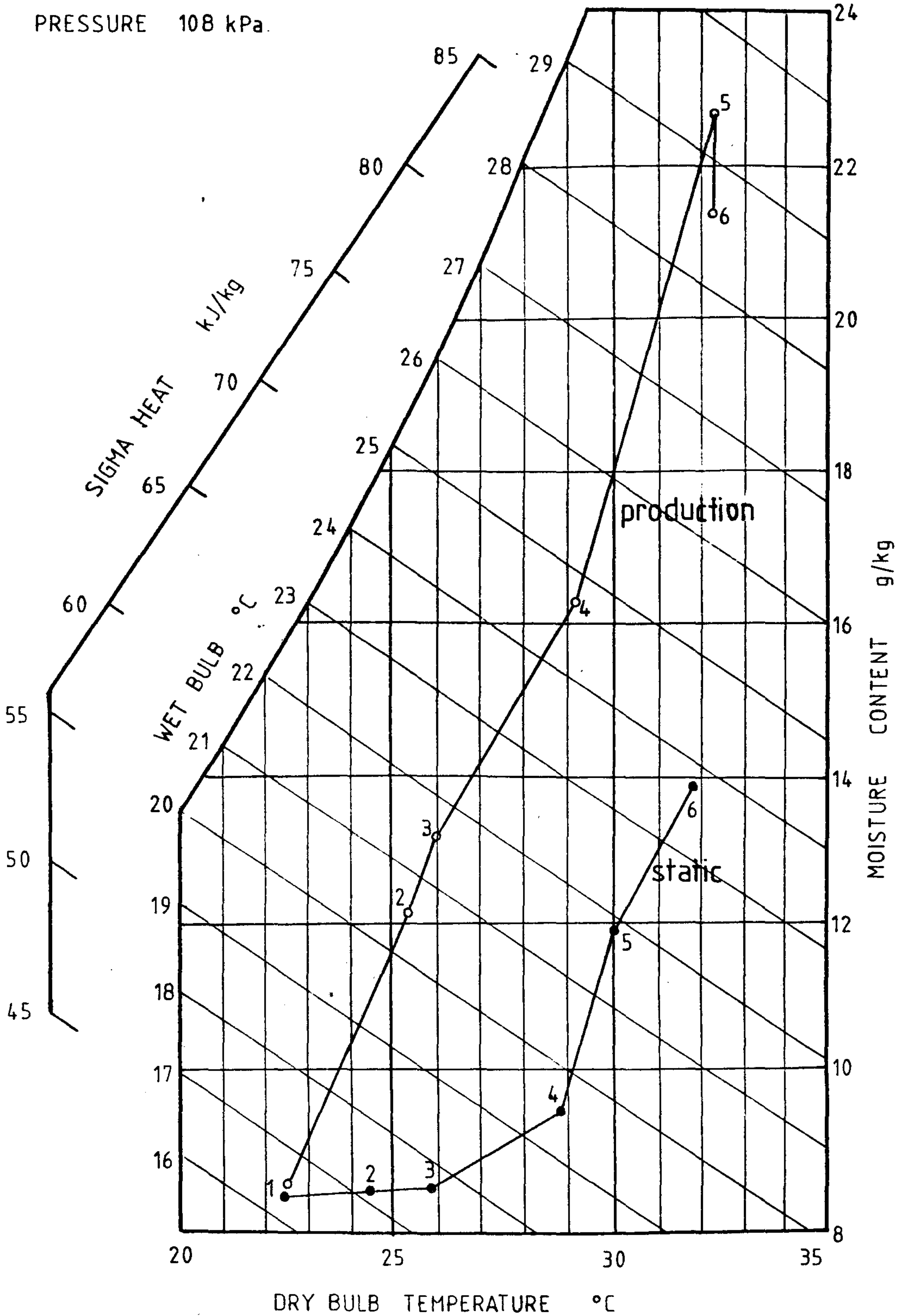


FIG 6·5d AIR CONDITIONS IN HICKLETON T01s INTAKE

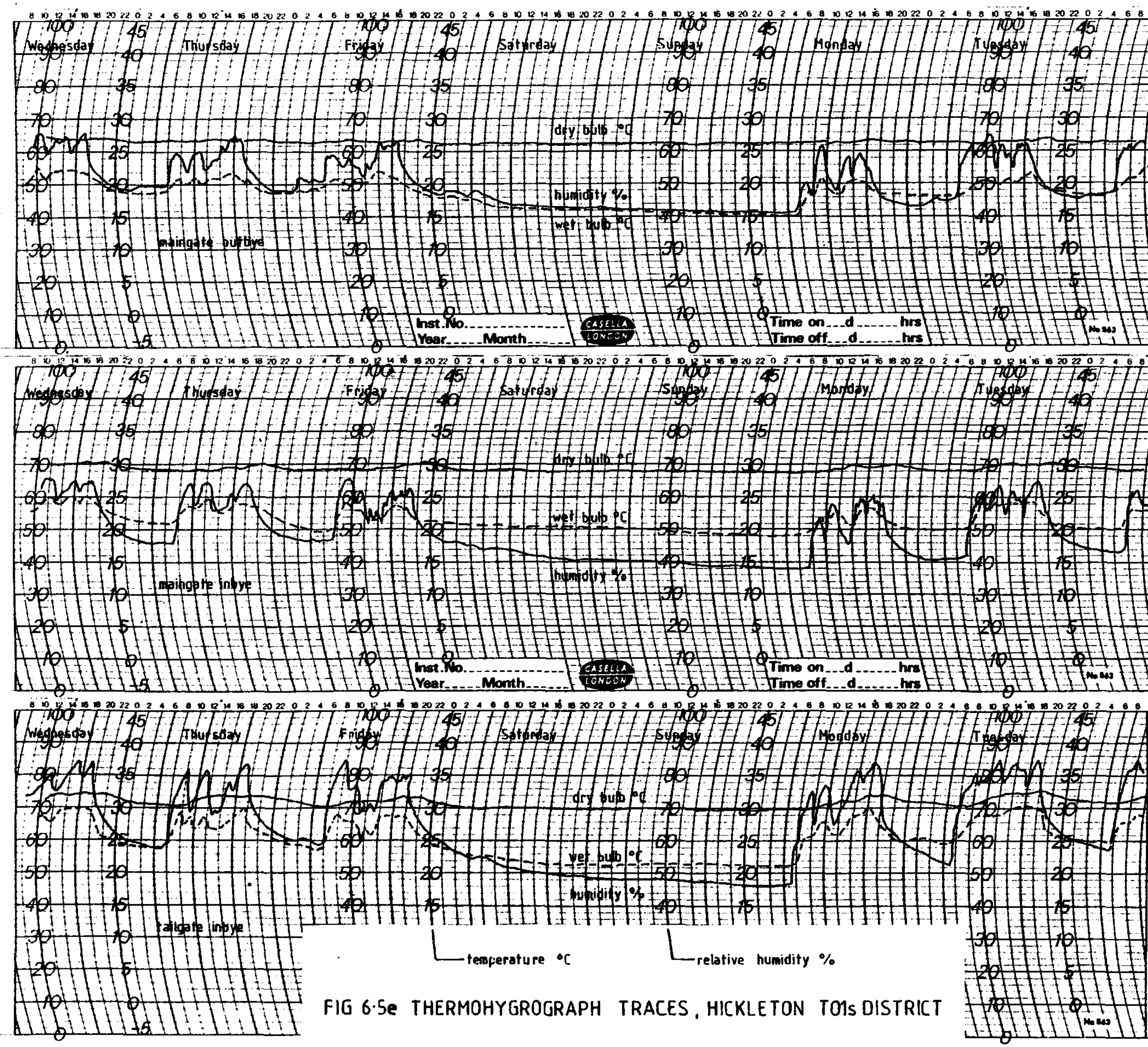


FIG 6·5e THERMOHYGROGRAPH TRACES, HICKLETON T01s DISTRICT

6.5.3 Discussion. Static conditions

Along the west intake, in static conditions the only heat source is the strata. From stations 1 to 3 a distance of 1350 m the heat gain was only 126 kW (93 W per m length). At the time of the survey the airway was seen to be dry and this is confirmed by the negligible moisture gain along this well established intake. Reference to the psychrometric chart figure 6.5d shows graphically the sensible, dry, heat transfer to the air as the air passed along the intake. Note that stations 1, 2 and 3 are almost on a horizontal line of constant moisture content.

Between stations 3 and 4 the heat and moisture transfer from the strata increased due to the airway being more recently driven. Although the air quantity here was about one third that in the main intake the heat gain was still 85 kW. The face between stations 4 and 5 had a heat gain of 94 kW in only 200 m showing an even larger strata heat load for newly exposed strata. Along the tailgate to station 6 the heat and moisture gain was of the same order as the maingate which was in similar good condition.

6.5.4 Discussion. Production conditions

The survey made in peak production conditions shows much greater heat and moisture gains around the ventilation circuit apart from in the tailgate. Along the west intake this effect is well illustrated with a fourfold increase in the heat gain and a massive increase in the moisture pickup. The differences between the production and static figures are due principally to the conveyor. With motor-drive units working at aroung 50% of rated power, estimated from meter readings in the sub station, this produces figures of about 210 kW from the motor-drive units and 210 kW from the coal and the whole 166 g/s moisture gain from the coal. The coal flow from T01's and P46's totals about 70 kg/s average.

A similar situation existed in the maingate with an increase in the heat and moisture gains of 72 kW and 34 g/s to 157 kW and 47 g/s. These increases are not as pronounced as in the intake because of the already high temperature and moisture content of the air on entering the maingate at station 3. The heat and moisture gains along the face were about 3 times those in static conditions. The 268 kW heat gain would be mostly attributable to the machinery since at the high temperatures reached in peak production the strata heat emission is temporarily reduced resulting in storage as described earlier. It is notable that the wet bulb temperature reached 29.3^{o}C in the tailgate face end. High enough for a considerable heat strain risk to workers.

As was the case at Bentinck K76's district the machinery on the face raised the air temperature above strata temperature and a slight cooling of the air took place in the tailgate in section 5-6.

CHAPTER 7

HEAT EMISSION FROM CONVEYED COAL

CHAPTER 7

HEAT FROM CONVEYED COAL

7.1 INTRODUCTION

The underground investigations at Bentinck and Hickleton Collieries described in Chapter 6 have illustrated the potential of conveyed coal as a heat source and given an indication that its magnitude is comparable to that of the mining machinery. Estimates by the Mine Ventilation Research Department of the Federal German Coal Industry attribute in the order of 50% of the heat make in maingate conveyor roads and about 10% of the total mine heat load to conveyed coal [21]. Recent work by the Mining Research and Development Establishment [9] suggest similar figures. These are comparable to those measured at Bentinck and Hickleton Collieries.

This chapter provides a qualitative assessment of the situation, followed by consideration of some factors which could affect the conveyed coal heat contribution and concludes with a simple theoretical treatment.

7.2 QUALITATIVE ASSESSMENT

7.2.1 Coal in the strata

Before the approach of any nearby face or airway the coal is at virgin rock temperature. As a coal face approaches the temperature falls at an increasing rate as heat is conducted more rapidly to the interface with the ventilating air due to the steepening temperature gradient. Despite the fact that the strata gives large quantities of heat to the airstream the temperature fall is small due to the vast thermal capacity of the mass of strata involved. The face temperature is typically within 1-2^{o}C of VRT on a fast advancing face.

7.2.2 The cutting process

During the cutting process the machine expends a large amount of mechanical energy to cut and break the coal. Most of this energy is dissipated through friction and heat to the ventilating air, machine dust suppression water, and the coal which may be expected to heat up about 0.5-2.0^{o}C depending on cutting efficiency. At Bentinck K76's a temperature rise of 1^{o}C was measured which is of the expected order. The shearer would be expending about 50 kW at the cutting drum with a coal production rate of 50 kg/s. Dividing the power input by the rate of production and specific heat of coal of 1 kJ/kg K gives a predicted temperature rise of 1^{o}C. The temperature fall of the coal before mining and the rise during cutting are roughly equal. This results in the temperature of the cut coal leaving the face on the AFC being very close to VRT.

Coal does not have a large specific heat capacity (1.0 - 1.3 kJ/kg K) or, in the solid, thermal conductivity (0.1 - 0.5 W/m^{o}C) but after mining its potential as a heat source is increased orders of magnitude.

During cutting the coal is heated, broken and crushed, wetted thoroughly with (usually hot) dust suppression water and well mixed. The airflow across, and possibly within, the coal is turbulent particularly on the face. This provides ideal conditions for convective and evaporative heat transfer.

7.2.3 On the conveyors

The coal is repeatedly mixed and wetted at the face end, stage loader and further outbye on the conveyors. Normal practice is to convey the coal outbye against the airstream (antitropal ventilation) and this results in a large relative velocity between coal and air, once again enhancing heat transfer. In this counter-flow situation as the coal cools and dries it meets progressively cooler and drier air on its journey outbye. This situation could be alleviated to a certain extent by conveying coal in return air (homotropal ventilation) as is practised in some mines. The reason this system is not commonly adopted is due to firefighting safety. Conveyors can catch fire and a conveyor fire in an intake sends smoke inbye. This allows firefighting teams access from fresh air without having to traverse a face. Also a water supply is more easily arranged.

As the moist coal travels outbye, particularly on the longer lengths of conveyor it cools and dries near the surface and a layer has been observed to form which insulates the main bed of coal on the conveyor belt. Dry coal has a low thermal conductivity as stated earlier and still air also has an extremely low conductivity (0.028 W/m^oC) so the benefits of this relatively cool dry skin as heat and moisture transfer are reduced are apparent.

Unfortunately at a transfer point the coal is sometimes sprayed and always well mixed so the coal surface temperature and wetness returns to a mean value approaching that for the whole coal bed. Consequently the heat emission from the coal rises to a new maximum to decay back to a lower level as the insulating surface forms once more. Consideration of the number of transfer points between any face and pit bottom illustrates how often the heat emission is revitalised in this way.

By the time the coal reaches the surface, possibly after spending time in bunkers and losing more heat, it is considerably cooler than when it started its journey outbye from the face. Some estimates of its potential were given in Chapter 6.4.

7.3 FACTORS AFFECTING HEAT AND MOISTURE TRANSFER

There are many variables which could affect the heat and moisture transfer within and from the surface of a bed of moist broken coal on a conveyor to an airstream. The following sections provide a brief summary of the principal parameters affecting the situation although these are not arranged in any order of importance or magnitude.

7.3.1 Psychrometric condition of the air

The wet and dry bulb temperatures and pressure of the air specify its condition and heat content. Dry bulb temperature affects the convective heat transfer (Chapter 5.3). Wet bulb temperature and vapour pressure figure prominently in the latent heat transfer (Chapter 5.5). Naturally lower air temperatures result in a greater driving potential and hence increase the heat and moisture transfer from the coal to the air.

7.3.2 Airflow, quantity and velocity

The quantity of the air absorbing a given amount of heat affects the psychrometric condition of the air. The relative velocity of the air to the coal is particularly important in evaluating convective heat transfer (Chapter 5.5).

7.3.3 Belt speed

This must be considered due to the relative velocity of the coal and air as above.

For coal transported at a certain rate the belt velocity dictates the load per unit length and the time a certain amount of coal has to transfer its heat.

7.3.4 Shape of coal bed

Obviously a thin flat bed of coal will cool faster than a narrower, deeper one having the same volume. A greater surface area is exposed to the airstream and the percentage of the coal cooled to form the insulating layer described in Chapter 7.1.3 is greater.

7.3.5 Temperature and wetness of coal loaded

Coal loaded with a high temperature and moisture content has a higher potential to lose heat and moisture to the air. Also the driving potential for heat and moisture emission is high.

7.3.6 Sprays and subsequent wetting

Evaporative heat transfer is enhanced by increased wetting of the conveyed coal. As little dust suppression water as is possible for the required effect should be used. If chilled service water systems are eventually installed in British coal mines there is a possibility of using the water to cool the coal on the conveyors and suppress dust. If cold enough water were used it could use the coal clearance system to remove a certain amount of heat, as heat would flow from the air to the coal.

7.3.7 Machine heat distribution

Although conveyor systems are the most efficient method currently available to move coal they still require a large drive power. Of the useful work done by the motor-drive unit all but that used to lift the coal against gravity is dissipated along the operating length of the machine overcoming roller friction. It therefore appears as heat which may be conducted through the conveyor structure to be transferred to the air or transferred to the belt and coal.

Tests using an infra red thermometer at Pye Hill Colliery, an electronic indicating thermometer at Bentinck Colliery and a sensitive infrared thermal imaging device at Daw Mill Colliery failed to detect any appreciably warm structure even on heavily loaded conveyors. Only failing bearings could be found to be hotter than the air dry bulb temperature. None of the instruments were accurate to more than $1^{o}C$ so the situation is probably that the large area of highly conductive steel structure only needs to warm slightly to dissipate a relatively large quantity of heat.

7.3.8 Radiant heat exchange with roadway walls

Radiant heat exchange with the conveyor roadway walls may be in either direction. The wall temperature of most of the relatively dry roadways in British collieries may be taken as being equal to the dry bulb temperature. The temperature of the belt bottom is also close to dry bulb temperature so radiant heat exchange at the bottom surface is small. The temperature of the coal surface though could vary from almost VRT down to air wet bulb temperature so the coal could have a higher or lower temperature than the walls resulting in a heat flow which could be in either direction.

7.3.9 Thermal characteristics of coal

The thermal characteristics which must be considered are those of a mix of coal,air and water in varying ratios. The specific heat of the mix may easily be evaluated providing the fractions of each constituent are known. The conductivity and diffusivity which affect the flow and distribution of heat in the coal bed are much more difficult to quantify. Chapter 8 describes attempts to evaluate some of these parameters.

7.3.10 Size distribution of coal

This affects the thermal characteristics of the coal bed due to packing differences changing the ratios of air, water and coal. The size distribution will also dictate the shape and roughness of the coal surface and hence the convective and latent heat transfer. The layer of coal near the surface also could be affected due to different size distributions allowing air penetration of the coal bed to different extents.

Due to its low conductivity and smaller surface area in the solid it is apparent that a single large lump of coal will take much longer to cool than the equivalent mass of fine coal.

7.4 RELATIONSHIP BETWEEN HEAT AND MOISTURE EMISSION

Early attempts to quantify the heat emission from conveyors were unjustifiably based on the principles of dry heat exchange. This enabled the temperature rise in the air to be calculated from the heat balance shown below.

$$\Delta t_c \cdot m_c \cdot C_c = \Delta t_a \cdot m_a \cdot C_{pa} \qquad (1)$$

where:-

Δt_c = Temperature change of coal (^oC)

m_c = Mass flow rate of coal (kg/s)

C_c = Specific heat of coal (kJ/kg K)

Δt_a = Temperature change of air (^oC)

m_a = Mass flow rate of air (kg/s)

C_{pa} = Specific heat of air (kJ/kg K)

This resulted in the prediction of air temperature increases which were many times greater than those actually encountered. In fact a large amount of heat is transferred from the coal to the air by the evaporation of moisture. Reference to the thermohygrograph traces (figures 6.3a and 6.5e) and the psychrometric chart of conditions in the conveyor road at Hickleton (figure 6.5d) will illustrate this.

Latent heat transfer is such that moisture may be evaporated from a surface into the air without necessarily raising the dry bulb temperature. Other moisture is evaporated from conveyed coal in significant quantities resulting in large amounts of heat being transferred to the air. Thus direct, sensible heat transfer and latent heat transfer due to moisture evaporation must both be considered when evaluating the total heat emitted. The term heat emission may now be taken to include latent heat transfer and moisture evaporation.

7.5 CHOICE OF ANALYTICAL METHOD

The ultimate object of this exercise is to predict the rate of heat flow from the cut coal on a conveyor to the ventilating airstream of a mine. The accepted approach for such a study would be to start from simple theoretical first principles and build up a more representative and complex mathematical model in stages. The validity being verified by experiment in different conditions as the model was developed and refined. Ideally the theoretical model would be based on an analysis of the heat exchange between each individual piece of coal in the coal bed and the airstream. However the complex interactions taking place would result in extremely cumbersome mathematics. Such an analysis would be impractical for other reasons, for example assessment of the correlation between actual boundary conditions and those postulated in theory.

In an attempt to simplify analysis the coal on the conveyor will be treated as a continuous, possibly multi-layered, plane parallel slab. Using this composite model would necessitate the measurement of bulk thermal characteristics of a coal-air mixture.

Information regarding the boundary conditions would still be difficult to obtain but the problems were not thought to be insurmountable.

A theoretical approach to the problem was needed which could also be reproduced in and correlated with a laboratory model. It was decided that the configurations of the theoretical and laboratory models should be such that the mathematics required was simple and already well proven. Also the practical experiment would be easy to build and control.

The theoretical model was designed to use, where possible, parameters which were already known, easy to measure directly or found from other properties which were easy to measure.

7.6 EQUATIONS FOR MASS AND HEAT EXCHANGE

Sections 7.7 and 7.8 state the basic principles of mass and heat transfer through and from the conveyed coal. The equations used are for steady state conditions. Naturally conditions would vary as the coal cooled and dried and different air temperatures were encountered, but equations for non steady conditions would be extremely complex when gathered together to represent all the heat transfer at the surface. It would also be difficult to relate these to practical situations.

It is envisaged that the steady state solutions could be used over a short period of time of at present unknown length. After such a period and knowing how much heat had left the coal, the variables could be reset to provide a new set of input conditions.

This process repeated a number of times would provide a series of step values which would ideally follow the curve describing the actual conditions. Such a method would also be very suitable for integration into a computer program. The following equations apply to the exchange of heat and water vapour between the warm moist coal and surroundings. In most cases they are based on the equations in Chapter 5.

The heat exchange at the upper and lower surfaces is initially dealt with in separate sections.

7.7 HEAT EXCHANGE AT COAL UPPER SURFACE

7.7.1 Heat exchange by convection

$$q_{cu} = h_{cu}\ (t_u - t_{db}) \tag{2}$$

where:-

q_{cu} = Convective heat emission from coal upper surface (W/m^2)
h_{cu} = Convective heat transfer coefficient ($W/m^{2}{}^{o}C$)
t_u = Temperature of coal upper surface (^oC)
t_{db} = Air dry bulb temperature (^oC)

7.7.2 Heat exchange by evaporation

$$q_{eu} = h_{cu}\ .\ 0.7\ .\ L\ \frac{(e_u - e_a)}{P} \tag{3}$$

where:-

q_{eu} = Evaporative heat transfer from upper surface (W/m^2)
L = Latent heat of evaporation of water (J/kg)
e_u = Vapour pressure at surface (kPa)

e_a = Vapour pressure of surrounding air (kPa)

P = Ambient air pressure (kPa)

7.7.3 Heat exchange by radiation

$$q_{ru} = h_r \cdot (t_u - t_r) \cdot F_{ev} \qquad (4)$$

where:-

q_{ru} = Radiative heat exchange (W/m^2)

h_r = Radiative heat transfer coefficient ($W/m^{2\,o}C$)

t_r = Temperature of roadway walls (^oC)

F_{ev} = Emissivity and view factor - N.B. This may be either positive from the coal, or negative to the coal from the roadway walls.

7.7.4 Conduction to the surface

$$q_{ku} = -k_u \cdot \frac{dt}{dx} \qquad (5)$$

where:-

q_{ku} = Heat conduction to the surface (W/m^2)

$-k_u$ = Bulk thermal conductivity of coal (W/m^oC)

$\frac{dt}{dx}$ = Temperature gradient in coal ($^oC/m$)

It is envisaged that the bulk thermal conductivity of the broken coal, air and water mix would change as the coal dried.

7.7.5 Thermal balance

For heat balance the heat conducted to the surface would equal the heat exchange at the surface due to convection evaporation and radiation giving:-

$$q_{ku} = q_{cu} + q_{eu} + q_{ru} \qquad (6)$$

$$-k_u \frac{dt}{dx} = h_{cu} (t_u - t_{db}) + h_{cu}\ 0.7.L \frac{(e_u - e_a)}{P} + h_r (t_u - t_r) \cdot F_{ev} \qquad (7)$$

7.8 HEAT EXCHANGE AT THE LOWER SURFACE

A different set of conditions affect heat transfer through the lower side of the conveyor. The existence of the belt forming an impervious barrier results in no latent heat transfer. The belt material also forms an extra thermal barrier.

7.8.1 Heat exchange by convection

$$q_{c\ell} = h_{c\ell} \cdot (t_\ell - t_{db}) \qquad (8)$$

where:-

$q_{c\ell}$ = Convective heat emission from belt bottom (W/m^2)

$h_{c\ell}$ = Convective heat transfer coefficient for belt lower surface ($W/m^{2o}C$)

t_ℓ = Temperature of lower surface (^oC)

7.8.2 Heat exchange by radiation

$$q_{r\ell} = h_{r\ell}\,(t_\ell - t_r)\,.\,F_{ev} \qquad (9)$$

where:-

$q_{r\ell}$ = Radiative heat exchange with roadway (W/m^2)

$h_{r\ell}$ = Radiative heat transfer coefficient (W/m$^{2\,o}$C)

F_{ev} = Emissivity and view factor

7.8.3 Conduction through the coal

$$q_{k\ell} = -k_\ell \frac{dt}{dx} \qquad (10)$$

where:-

$q_{k\ell}$ = Heat conduction through lower layers of coal (W/m$^{2\,o}$C)

$-k_\ell$ = Bulk thermal conductivity of coal (W/m^{o}C)

$\frac{dt}{dx}$ = Temperature gradient in lower layers or coal (^{o}C/m)

It is expected that the thermal conductivity of coal near the bottom of the coal bed would be different to that near the surface and more stable as no drying would take place here.

7.8.4 Conduction through the belt

$$q_{kb} = -k_b \frac{(t_\ell - t_{b\ell})}{x_b} \qquad (11)$$

where:-

q_{kb} = Heat conduction through the belt (W/m^2)

k_b = Thermal conductivity of belt material (W/m^{o}C)

t_ℓ = Temperature of lower surface of coal (^oC)

$t_{b\ell}$ = Temperature of lower surface of belt (^oC)

7.8.5 Thermal balance

For thermal balance heat conducted through the bottom layers of coal will be equal to the heat conducted through the belt material which in turn should equal the heat exchange at the surface giving:-

$$q_{c\ell} + q_{r\ell} = q_{k\ell} = q_{kb} \tag{12}$$

$$h_{c\ell}\ (t_\ell - t_{db}) + h_{r\ell}\ (t_\ell - t_r) \,.\, F_{ev} = -k_\ell \frac{dt}{dx} = -k_b \frac{(t_\ell - t_{b\ell})}{x_b} \tag{13}$$

7.9 CONSOLIDATION OF EQUATIONS AND ESTIMATES OF HEAT TRANSFER COEFFICIENTS

7.9.1 Upper surface

Equation 7.9 (7) summarizes the heat exchange at the top surface in a single heat balance equation. Some collection of terms and estimation of coefficients is now attempted so that the equation may be simplified and important parameters which will need experimental evaluation are identified.

$$-k_u \frac{dt}{dx} = h_{cu}\ (t_u - t_{db}) + h_{cu} \,.\, 0.7 \,.\, L \,.\, \frac{(e_u - e_a)}{P}$$

$$+ h_r\ (t_u - t_r) \,.\, F_{ev} \tag{14}$$

7.9.2 Radiative heat transfer

The section of the equation dealing with radiative heat exchange may be simplified and matched to that dealing with convection by making two assumptions:

(i) The emissivity and view factor may be approximated to 1 with only small errors. Referring to equation 5.4 (25):-

$$F_{ev} = \left(\frac{1}{\varepsilon_1} + \frac{A_1}{A_2}\left(\frac{1}{\varepsilon_2} - 1\right)\right)^{-1} \qquad (15)$$

where

F_{ev} = Emissivity and view factor
ε_1 = Emissivity of coal surface
ε_2 = Emissivity of roadway walls
A_1 = Area of coal surface (m^2)
A_2 = Area of roadway walls (m^2)

As both the coal surfaces and the roadway wall are rough, dull and black an emissivity value of 0.98 is inserted. This figure has been checked and found correct as is described later.

For each 1 m^2 of coal surface there exists about 10 m^2 of roadway in the average roadway.

Inserting these values in equation (15):-

$$F_{ev} = \left(\frac{1}{.98} + \frac{1}{10}\left(\frac{1}{.98} - 1\right)\right)^{-1}$$

$$= 0.9780$$

Thus by approximating F_{ev} = 1 will give 2.2% error.

(ii) The skin temperature of a dry roadway wall has been measured by several workers and found to equate very closely to air dry bulb temperature (Chapter 10.7.8). Insertion of dry bulb temperature for the roadway skin temperature is therefore justified. This allows the radiative heat transfer equation to use the same temperatures as the convective case.

i.e.

$$q_{ru} = h_{ru} \; (t_u - t_{db}) \qquad (16)$$

For an estimate of the radiative heat coefficient we may evaluate h_r using equation 5.4 (27)

$$h_r = 0.2268 \left(\frac{T_{av}}{100}\right)^3 \qquad (17)$$

T_{av} = Average temperature of the two surfaces involved (K).

Inserting a value of 300 K we obtain:-

$$h_r = 0.2268 \left(\frac{300}{100}\right)^3$$

$$= \underline{2.0412 \text{ W/m}^{2\,o}\text{C}}$$

This is the heat transfer from the coal to the roadway walls. When the roadway walls are hotter than the coal the emissivity and view factor must be calculated. It reduces the heat transfer coefficient by about 20%.

7.9.3 Latent heat transfer

Referring to equation 7.7(3) we find that the latent heat transfer coefficient is linked to the convective heat transfer coefficient. The justification for this is given in Chapter 5.5

$$q_{\ell u} = h_{cu} \cdot 0.7 \cdot L \frac{(e_u - e)}{P} \qquad (17)$$

To simplify equation (17) and remove the pressure, vapour pressure and latent heat terms on "equivalent temperature difference" is used. Whillier [18] gives an equivalent temperature:-

$$\Delta t_L = 17\ (e_u - e) \qquad (18)$$

where

Δt_L = Equivalent temperature difference (^oC)

e_u = Vapour pressure at upper surface (kPa)

e = Vapour pressure of the air (kPa)

The vapour pressures are still needed as input data but we now express the latent heat transfer thus:

$$q_{\ell u} = h_{cu} \cdot \Delta t_L \qquad (19)$$

7.9.4 Collection of terms

Equation 14 may now be expressed

$$k_u \frac{dt}{dx} = (h_{cu} + h_r)(t_u - t_{db}) + h_{cu} \Delta t_L \qquad (20)$$

Similarly for the bottom surface

$$k_u \frac{dt}{dx} = k_b \frac{t_\ell - t_{b\ell}}{x_b} = (h_{c\ell} + h_{r\ell}) \cdot (t_\ell - t_{db}) \qquad (21)$$

Equations (20) and (21) describe the heat exchange at the surface of the coal and the belt bottom. The right hand side is particularly dependent on the convective heat transfer coefficient which as stated in Chapter 5.2 is particularly difficult to predict accurately. Work was carried out later in the study to evaluate this important coefficient and also to check that the latent heat transfer coefficient could be based on this coefficient.

These equations also illustrate the importance of the surface temperature. Accurate surface temperature measurement is particularly difficult since in most cases the instrument used to measure the surface temperature makes a contribution by disturbing the conditions on attempting to measure. The next section describes a possible method for its prediction.

7.10 THE TEMPERATURE PROFILE WITHIN THE COAL BED

The left hand sides of equations describing the conduction of heat to the surface from the nearby coal allows a heat balance to be attempted as a check.

The temperature distribution in the interior of the coal on the conveyor may be described by the Fourier differential equation of conduction. The one dimensional case is illustrated here as it is assumed that the coal bed forms a plane parallel faced slab. This equation describes the temperature profile perpendicular to the surface.

$$\frac{\partial t}{\partial \tau} = \frac{k}{\rho C} \cdot \frac{\partial^2 t}{\partial x^2} = a \frac{\partial^2 t}{\partial x^2} \qquad (22)$$

where:-

t = Temperature (^{o}C)

τ = Time (s)

x = Distance (m)

k = Thermal conductivity ($W/m^{o}C$)

ρ = Density (kg/m^3)

C = Specific heat (kJ/kg K)

a = Thermal diffusivity (m^2/s)

This equation does not describe a steady state situation but specifies temperature in the coal at a given time and position. It could therefore be used to evaluate the temperature gradient near the surface and hence the heat conduction to the surface in equations (20) and (21).

It must be stressed that the thermal conductivity and hence diffusivity are bulk values for the coal air water mix and that they would change as the coal dried. The rates of change being at present unknown.

7.11 REQUIREMENT FOR EXPERIMENTAL INVESTIGATION

The theoretical treatment described in this chapter may not provide an answer in itself to the evaluation of heat transfer from the coal on a conveyor, but it has allowed identification of important coefficients and characteristics and shows which must be evaluated to find a solution to the problem. The main tasks of the experimental work were therefore to quantify the convective heat transfer coefficient of a rough coal surface and the bulk thermal conductivity of broken coal. Chapter 8 describes an attempt at theoretical prediction and experimental evaluation of the thermal conductivity of broken coal.

CHAPTER 8

EVALUATION OF THE THERMAL CONDUCTIVITY OF BROKEN COAL

CHAPTER 8

THE THERMAL CONDUCTIVITY OF BROKEN COAL

8.1 Introduction

This chapter describes the evaluation of the bulk thermal conductivity of a mixture of broken coal, air and water, initial work being concerned only with a dry coal air mixture. Very little work has been carried out in the past either experimentally or theoretically to assess this characteristic. Davis and Byrne (1922) and Schuman and Voss (1934) made some measurements of thermal conductivity of granulated coal. Their work is not well documented and the units used not clearly stated.

The only other information available about heat transfer in granular coal is concerned with the coking and gasification processes and is concerned with very high temperatures where the heat transfer processes are vastly different.

Two sets of experiments are described. The object of the first was to measure the thermal conductivity directly, the second to measure the thermal diffusivity from which the conductivity may be evaluated.

In the last section a theoretical method of calculating the bulk thermal conductivity is briefly described and demonstrated.

8.2 DIRECT MEASUREMENT OF THERMAL CONDUCTIVITY

8.2.1 Design of apparatus

Chapter 5.2 describes the conduction of heat and gives Fourier's equation of steady one dimensional heat conduction through a plane parallel body.

$$Q = -kA\left(\frac{t_2 - t_1}{x_2 - x_1}\right) \qquad (1)$$

where

Q = Heat transferred by conduction (W)

k = Thermal conductivity (W/m^oC)

t = Temperature (^oC)

x = Distance (m)

(Also see figures 5.2a and 8.2a.)

There are several methods for evaluating thermal conductivity which are based on equation (1) or use it in some slightly modified form, eg divided bar apparatus, but few are suitable for use with granular materials. In the interests of simplicity equipment was devised which would use the Fourier equation and model the situation shown in figure 8.2a.

The configuration of the apparatus was such that it was a model of part of the infinite flat plate on which the Fourier equation is based. The temperatures of the end planes being maintained constant by a heater at one end and a cold water tank at the other. Insulation around the sample would restrict radial heat flow to very low levels so heat could be considered to flow

FIG 8·2a STEADY STATE HEAT CONDUCTION

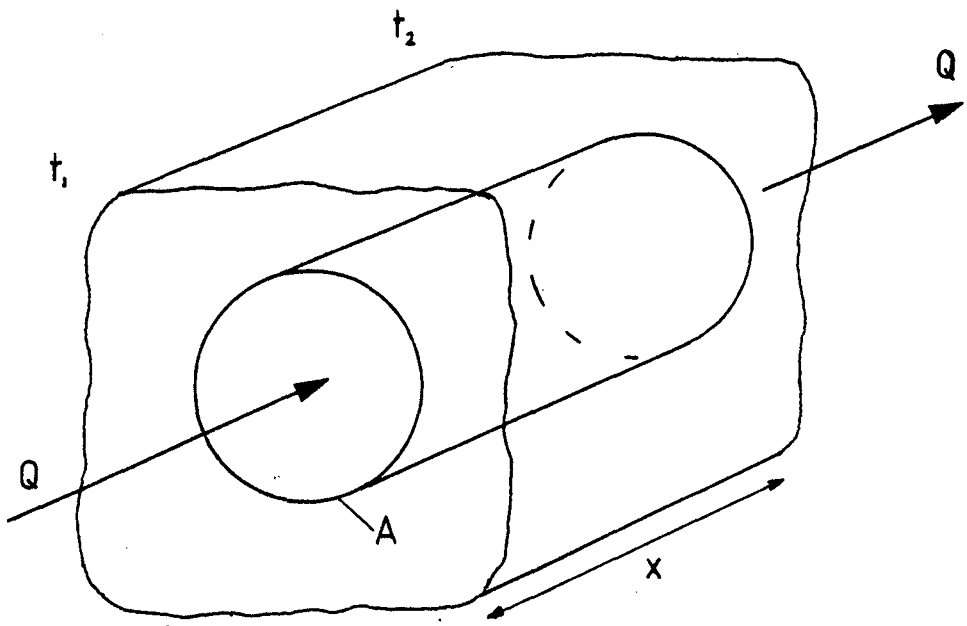
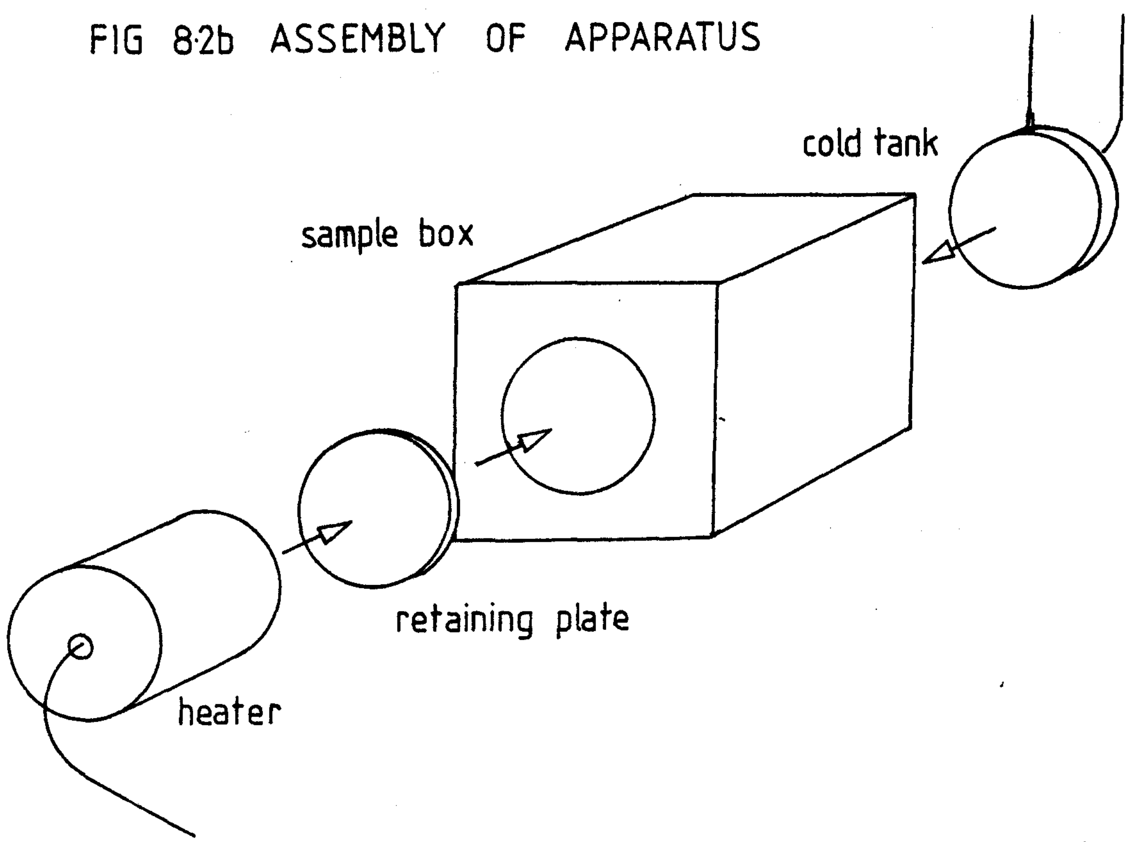


FIG 8·2b ASSEMBLY OF APPARATUS



only axially. Temperature difference across the sample was measured using thermocouples and the amount of heat flowing through the sample by flow rate and temperature difference through the cold water tank. A more detailed description of the main components follows.

8.2.2 Sample box

This component was cast in the form of a 400 mm cube with a 150 mm diameter hole through the centre. The material used was an insulating foam. This was formed from two liquid resin constituents which foamed and expanded rapidly on mixing. Solidification of the resultant aerated block took about 5 minutes. The thermal conductivity of the solid foam was 0.035 W/m^{o}C (manufacturers specification). The central cylindrical hole was lined with polythene prior to casting and this bonded strongly to provide a strong air and water tight liner.

Thermocouples in two concentric rings around the cylinder were also cast in. These were to check for radial heat flow which was found to be negligible in use. This was due to the conductivity of the coal being much higher than that of the insulation and the mean sample temperature being near ambient. The sample of coal was retained in the central core at the 'hot end' by a steel plate which was painted matt black to enhance heat transfer. The other end of the sample was retained by the cold water tank.

8.2.3 Cold water tank

To keep the 'cold end' of the sample at a constant temperature and allow measurement of the amount of heat passing through the sample a cold water tank was used. It was constructed in the form of a flat cylinder the same diameter as the sample cylinder and held in position by interference fit in a retaining ring. The tank was constructed of tinplate with connections for water inlet and outlet. Inside the cylinder was a long spiral fin which was soldered to the inside surface of the face of the cylinder in contact with the coal sample. This fin improved heat transfer to the water by increasing the water velocity as it flowed in a long spiral path from inlet to outlet and also increased the effective area of the cylinder face.

8.2.4 Chilled water supply

The water supply to the cold tank needed to be at constant low temperature and have a constant flow. The cold end was maintained at a constant temperature of just a few degrees centigrade and this avoided the hot end having to be dangerously hot in order to produce the desired temperature difference across the sample. This would allow the mean sample temperature to be kept near the room ambient temperature which would reduce radial heat flow to a minimum level. Also any heat flow outwards through the insulation at the hot end would be compensated for by the reverse at the cold end. To provide a steady temperature near zero two beer chillers were acquired. These took the form of a small refrigeration unit which chilled a cold reservoir to a thermostatically controlled temperature. When the flow rate through the cooling coil in the reservoir was low the outlet temperature of the fluid passing through was extremely stable.

To provide the necessary constant flow rate the water supply was taken from a large constant head tank. The flow rate was controlled by a needle valve between the tank and the beer chillers. Flow rate was measured by timed collection into a graduated container.

8.2.5 Heater

The heater was constructed from tin plate the same diameter as the sample cylinder with a 60 W light bulb as the heat source. In order to minimise radiant heat flow in all directions other than axially through the sample the heater container was provided with a double lining except in the direction towards the sample. The heater was held in place at the sample hot end by interference fit in a retaining ring. The mains supply voltage to the bulb passed through a rheostat which enabled the heat output to be regulated and provide a reduced operating voltage to avoid overheating and failure of the bulb. No attempt was made to contain and measure the heat at this end of the apparatus. No insulation was used and excess heat was allowed to escape.

8.2.6 Sample temperature thermocouple

To measure the temperature difference across the sample a 2 x 6 junction thermocouple was constructed. The materials used were copper and eureka wire assembled in the configuration shown in figure 8.2c. The thermocouple was calibrated to 0.1^{o}C accuracy for the temperature range 0-80^{o}C. Each junction was coated in

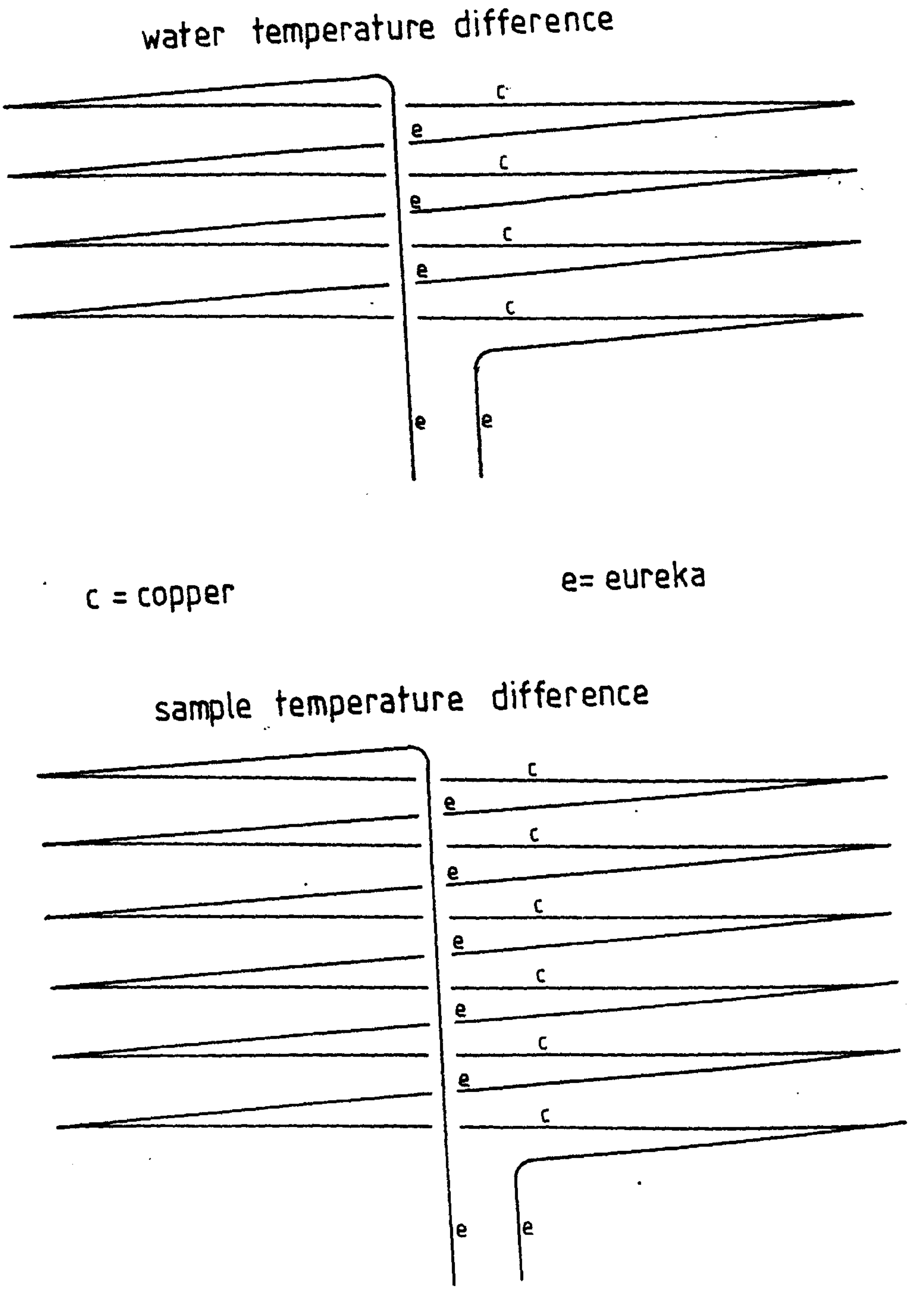
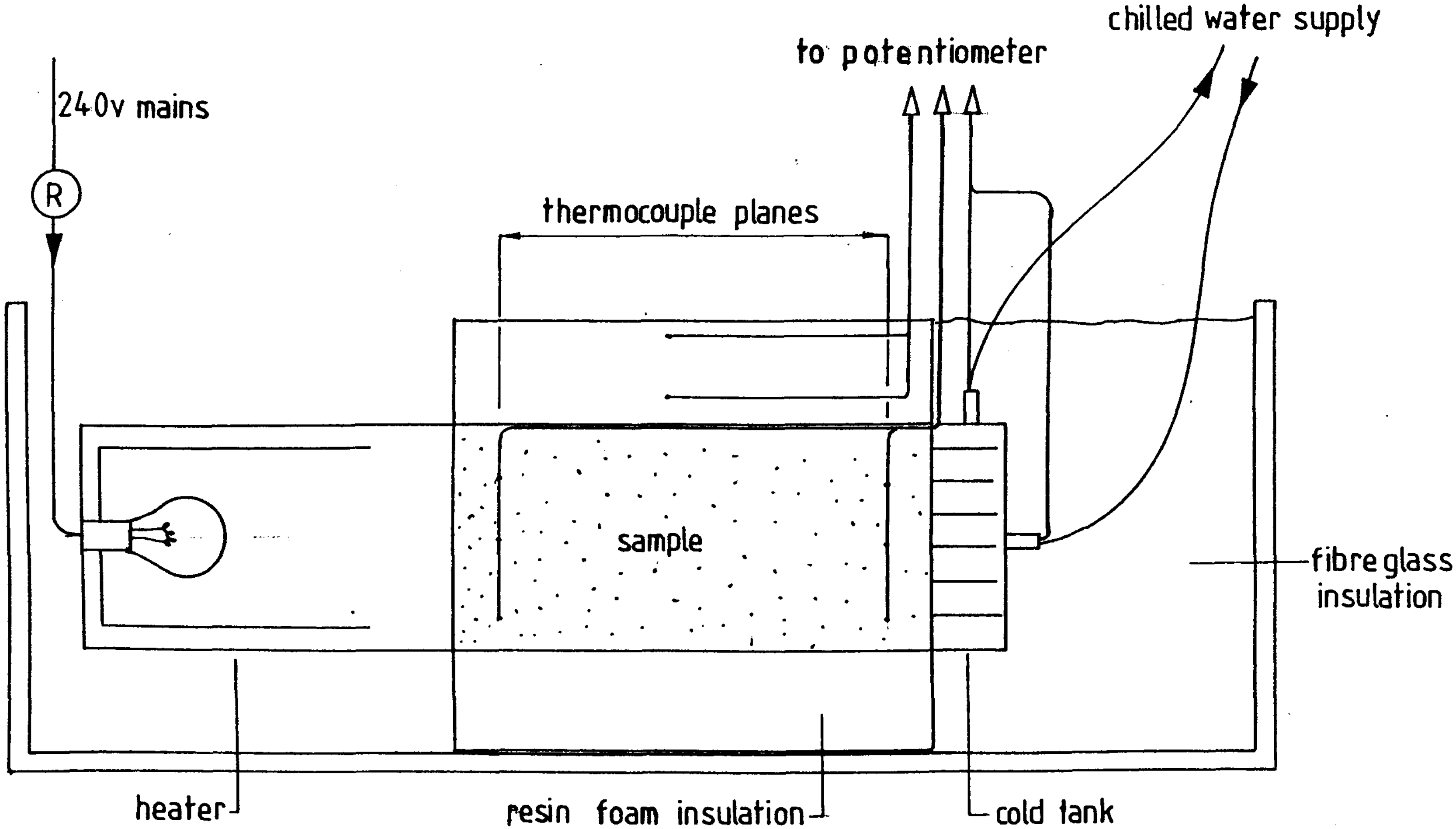


FIG 8·2c CONFIGURATION OF THERMOCOUPLE SETS

FIG 8.2d APPARATUS FOR STEADY STATE MEASUREMENT OF CONDUCTIVITY



epoxy resin to protect it and the whole circuit was liberally varnished to waterproof it.

To position the thermocouples within the sample the six junctions at each end were placed in a ring 80 mm in diameter concentric with the cylindrical case as it was filled. The two rings of thermocouples were arranged 50 mm from each end giving two thermocouple planes 300 mm apart within the sample. The 50 mm of sample left at each end was to reduce end effects.

The voltage of the thermocouple circuit was measured using a Pye potentiometer and light spot galvanometer. The system balanced the voltage of the thermocouple using a bridge circuit against a measured proportion of that from a cell to give voltage readings in 1 μV increments.

8.2.7 Water temperature thermocouples

The water temperature at the inlet and outlet of the water tank had to be accurately measured to allow calculation of the heat flow through the sample which was expected to be low.

A thermocouple was constructed which had four junctions at each end. As the junctions at each end were inserted into a small pipe to measure the water temperature they had to be constructed to be as small as possible. Four rather than one junction being used to raise the voltage produced by the thermocouple to an accurately measurable level despite the low temperature differential between the hot and cold junctions. The junctions were each sealed separately after joining the wires and then sealed together with epoxy resin.

The thermocouple was calibrated against two National Physical Laboratory certified thermometers graduated in $0.01^{o}C$ increments. The calibration covered a range of $0 - 1.0^{o}C$ with an accuracy of $0.01^{o}C$. It was carried out by warming one container of paraffin in a water bath to $2^{o}C$ above the temperature of a second identical container. Each carried a set of junctions and thermometer. As the container was warmed both were stirred, temperatures read and voltage of the thermocouple noted. The warmed container was then allowed to cool so the temperature difference fell back to zero. More measurements were made during this period to check for hysterysis effects. None were detected.

This thermocouple was then installed with one set of junctions in the inlet and one in the exit pipes of the cold water tank. Each was sealed in with epoxy resin and gave reliable service.

8.2.8 Assembly and operation

The apparatus was assembled and used as shown in figures 8.2b and d. The sample cylinder was filled whilst vertical so that the sample thermocouples could be accurately placed and the cold tank set tight against the sample for good thermal contact. This assembly was then lowered to a horizontal position in a large outer box. Fibre glass insulation was then placed around the cold tank so the only heat reaching it would come through the sample. This was verified by testing without the heater on. The check showed the increase in water temperature across the cold tank was too small to measure reliably and may be neglected.

Once filled and assembled the heater voltage and water temperature and flow rate were set and the equipment was left to settle to a steady state. Apart from checks for radial heat flow and of sample cold end temperature only the following experimental measurements needed to be taken.

(i) Water flow rate through cold tank by timed collection into a large measuring cylinder.

(ii) Sample temperature, thermocouple voltage.

(iii) Water temperature, thermocouple voltage.

The above two differential temperatures being measured using the Pye potentiometer set. The temperature difference across the sample, also between the cold tank, water at inlet and outlet can then be evaluated from the relevant calibration curves. The results were processed as shown in section 8.3.1.

Once set and left to run the apparatus required a surprisingly long time to reach a steady state, usually a week. This is shown in the table of results, figure 8.3a. Having recognised that the apparatus took so long after filling to attain a steady state the apparatus was left for one week to settle after filling before the daily readings were taken.

Tests were carried out on three samples before the equipment failed.

(i) $-\frac{1}{4}$" + $\frac{1}{8}$" sized coal, air dried

(ii) Run of mine, air dried

(iii) Run of mine, 5% moisture.

8.3 RESULTS AND DISCUSSION

8.3.1 Treatment of results

Referring to equation (1)

$$Q = -kA\left(\frac{t_2 - t_1}{x_2 - x_1}\right)$$

where

$t_2 - t_1$ = Temperature difference across sample (^oC)

$x_2 - x_1$ = Thickness of sample (m)

A = Area of sample (m^2)

Q = Heat conducted through sample (W)

The heat conducted through the sample is given by

$$Q = M_w \times C_{pw} \times \Delta t_w \qquad (2)$$

where

M_w = Mass flow rate of water (kg/s)

C_{pw} = Specific heat of water, 4187 (J/kg K)

Δt_w = Temperature change of water (^oC)

Rearranging (1)

$$k = \frac{Q.\Delta x}{A.(t_2 - t_1)} \qquad (3)$$

or

$$k = \frac{Q}{t_2 - t_1} \times \frac{0.3}{0.0177}$$

8.3.2 Sample calculation

Water thermocouple voltage 5.0×10^{-5} V

Temperature increase (t_w) 0.27°C (from calibration curve in Appendix)

Mass flow rate of water ($\dot{M}_w$) 1.49 g/s

Heat flow (Q)

$$\dot{M}_w \times C_p \times \Delta t_w = Q$$

$$1.49 \times 10^{-3} \times 4187 \times 0.27 = \underline{1.684 \text{ W}}$$

Conductivity (k)

Sample thermocouple voltage 1.252×10^{-4} V

Sample temperature difference ($t_2 - t_1$) 48.8°C

Sample thickness, fixed (x) 0.3 m

Sample area, fixed (A) 0.0177 m^3

Using equation (3)

$$k = \frac{Q}{t_2 - t_1} \times \frac{0.3}{0.0177}$$

$$= \frac{1.684}{48.8} \times \frac{0.3}{0.0177}$$

$$\underline{= 0.584 \text{ W/m}^\circ\text{C}}$$

8.3.3 Results

Figure 8.3a

Table of observed thermal conductivity of $-\frac{1}{4} \times \frac{1}{8}$" coal (dry)

Day	Water Flow g/s	Thermocouple Voltage V x 10^{-6}	Temperature Difference ^{o}C	Heat Flow W	Sample Thermo. Voltage V x 10^{-3}	Temperature Difference ^{o}C	Conductivity $W/m^{o}C$
1	1.49	50	0.27	1.684	12.52	48.8	0.584
2	0.77	60	0.33	1.064	12.05	47.0	0.384
3	1.45	39	0.20	1.214	13.85	54.0	0.381
4	0.58	50	0.27	0.656	8.85	34.5	0.322
5	0.55	32	0.28	0.645	14.18	55.3	0.197
6	0.56	39	0.20	0.469	20.88	81.4	0.098
7	0.48	47	0.25	0.502	18.98	74.0	0.115
8	0.45	42	0.22	0.414	18.88	73.6	0.095
9	0.35	52	0.28	0.410	18.83	73.4	0.095
10	0.38	50	0.27	0.430	17.95	70.0	0.104
11	0.42	47	0.25	0.440	18.31	71.4	0.104
12	0.36	55	0.30	0.452	19.26	75.0	0.102

Conductivity mean of days 6-12 = 0.102 $W/m^{o}C$

Figure 8.3b

Table of observed thermal conductivity of ROM coal (dry)

Day	Water Flow g/s	Thermocouple Voltage $V \times 10^{-6}$	Temperature Difference ^oC	Heat Flow W	Sample Thermo. Voltage $V \times 10^{-3}$	Temperature Difference ^oC	Conductivity W/m^oC
1	0.48	47	0.25	0.502	19.51	76.1	0.112
2	0.61	43	0.23	0.587	20.75	80.9	0.123
3	0.59	43	0.23	0.568	20.37	79.4	0.121
4	0.45	55	0.30	0.565	20.31	79.2	0.120
5	0.37	65	0.36	0.557	20.13	78.5	0.120
6	0.42	50	0.27	0.474	18.52	72.2	0.111

Mean Conductivity = 0.118 W/m^oC

Figure 8.3c

Table of observed thermal conductivity of ROM coal (5% moisture)

Day	Water Flow g/s	Thermocouple Voltage $V \times 10^{-6}$	Temperature Difference ^oC	Heat Flow W	Sample Thermo. Voltage $V \times 10^{-3}$	Temperature Difference ^oC	Conductivity W/m^oC
1	0.48	50	0.27	0.542	18.47	72.0	0.128
2	0.42	65	0.36	0.633	17.08	66.6	0.161
3	0.49	57	0.31	0.636	15.35	69.8	0.180
4	0.41	67	0.37	0.635	19.49	76.0	0.141
5	0.52	55	0.30	0.653	19.98	74.0	0.149
6	0.55	59	0.32	0.736	17.90	69.8	0.179

Mean Conductivity = 0.156 W/m^oC

8.3.4 Discussion

In the early stages of operating the equipment problems were encountered with the water supply mainly due to blockages in the pipework and freezing at the low flow rates used. Using a higher flow rate through a single chiller and putting only the amount required through the cold tank and wasting the rest cured the problem. A stable cold water supply at less than 5°C was obtained.

The ambient temperature in the experimental room was high 25 - 30°C so with the hot end at 75°C and the cold end near 0°C it can be seen that radial heat flow through the insulation would be minimised. None could be detected. The readings taken at all stages were repeatable including the low voltages on the water temperature thermocouples. Mean values for conductivity were:-

0.102 W/m°C for $-\frac{1}{4} + \frac{1}{8}$" sized coal

0.118 W/m°C for mixed ROM

0.156 W/m°C for mixed ROM (5% moisture)

No other work can be found to compare these values with directly but they appear to be of the right order. By comparison the thermal conductivity of solid coal is 0.2 - 0.3 W/m°C and air 0.028 W/m°C.

Many factors such as sample size and the experimental temperatures used could influence the values obtained for a given coal. These variations might arise from the effects of convection currents within the sample. Certainly conditions within the sample cylinder were less than uniform when it was opened after the test on the moist coal. The water appeared to have all migrated to the cold end. Condensation had taken place, not surprisingly, onto the cold tank and the coal at that end was certainly moist whilst that at the hot end had dried. For this reason the result for the moist coal must be treated with caution.

Little work has been carried out to measure the bulk thermal conductivity of granular materials and for the reasons described a definitive value would be difficult to obtain. Probably a standardised technique would help, the method described is suggested as a basis for this.

This method is based on steady state readings and therefore doubt must be expressed about the application of the same values to situations where rapid change is taking place. For this reason it was decided to design a method which would use a more representative transient condition. It was hoped that this would result in improved accuracy but would require measurement and processing of greater complexity. The results obtained so far would be useful for comparison.

Unfortunately before any more tests could be made on the steady state apparatus the cold tank ruptured due to icing. It was decided not to repair the tank, but to proceed on the transient evaluation as this was considered more realistic.

8.4 TRANSIENT MEASUREMENT OF DIFFUSIVITY

8.4.1 Theoretical basis

This experiment modelled part of a plane parallel slab (as described in Chapter 7.10) and derived thermal characteristics by monitoring how it cooled from a uniform temperature. It was assumed that the coal sample formed part of the infinite slab and the assumption was made that no heat flowed parallel to the surface but only perpendicular to it. In such a situation the temperature distribution with time is described by the one dimensional Fourier differential equation of conduction:-

$$\frac{\partial t}{\partial \tau} = \frac{k}{\rho C} \cdot \frac{\partial^2 t}{\partial x^2} = a \, \frac{\partial^2 t}{\partial x^2} \qquad (4)$$

where

t = Temperature (^{o}C)

τ = Time (s)

x = Distance (m)

k = Thermal conductivity ($W/m^{o}C$)

ρ = Density (kg/m^3)

C = Specific heat (kJ/kg K)

a = Thermal diffusivity (m^2/s)

The two differentials may be evaluated using a finite difference method to find the diffusivity.

The equation on which the experiment was based has the following input data.

Temperature, which varies with both position and time until eventually over a long period steady state is reached. For this experiment it was measured at several discrete positions and its variation with time measured.

Position, at which the temperature is measured. This was facilitated by placing the temperature measuring devices at specified fixed points.

Time. The two other parameters were recorded against a time base.

8.4.2 Equipment

The apparatus for this experiment consisted of an insulated sample box, temperature probes and a stop watch. The sample box was cubic with 200 mm sides and an open top. It was constructed of 10 mm polystyrene and insulated with a further 100 mm of fibre glass wool. The low specific heat of polystyrene and the good insulation would mean that heat would leave the sample hopefully, only from the top surface. Thus it was assumed that conditions for the Fourier equation were met.

A coal sample put into the open box would lose heat to the air as it was allowed to cool and temperature probes inserted during filling at known positions on the central axis gave the temperature of the coal continuously at fixed positions. The configuration of the apparatus is shown in figure 8.4a.

The temperature measuring system consisted of a set of small thermistor probes on flexible wire to measure the temperature in the coal and a thermistor probe to measure air temperature. These were led to a multiway switch box and hence to a meter displaying the temperature directly in a range 0 - 50^{o}C. The system was carefully calibrated before use and found to be within 0.05^{o}C of a standard thermometer for all probes. The instruments were manufactured by the Yellow Springs Instrument Co., Ohio, USA.

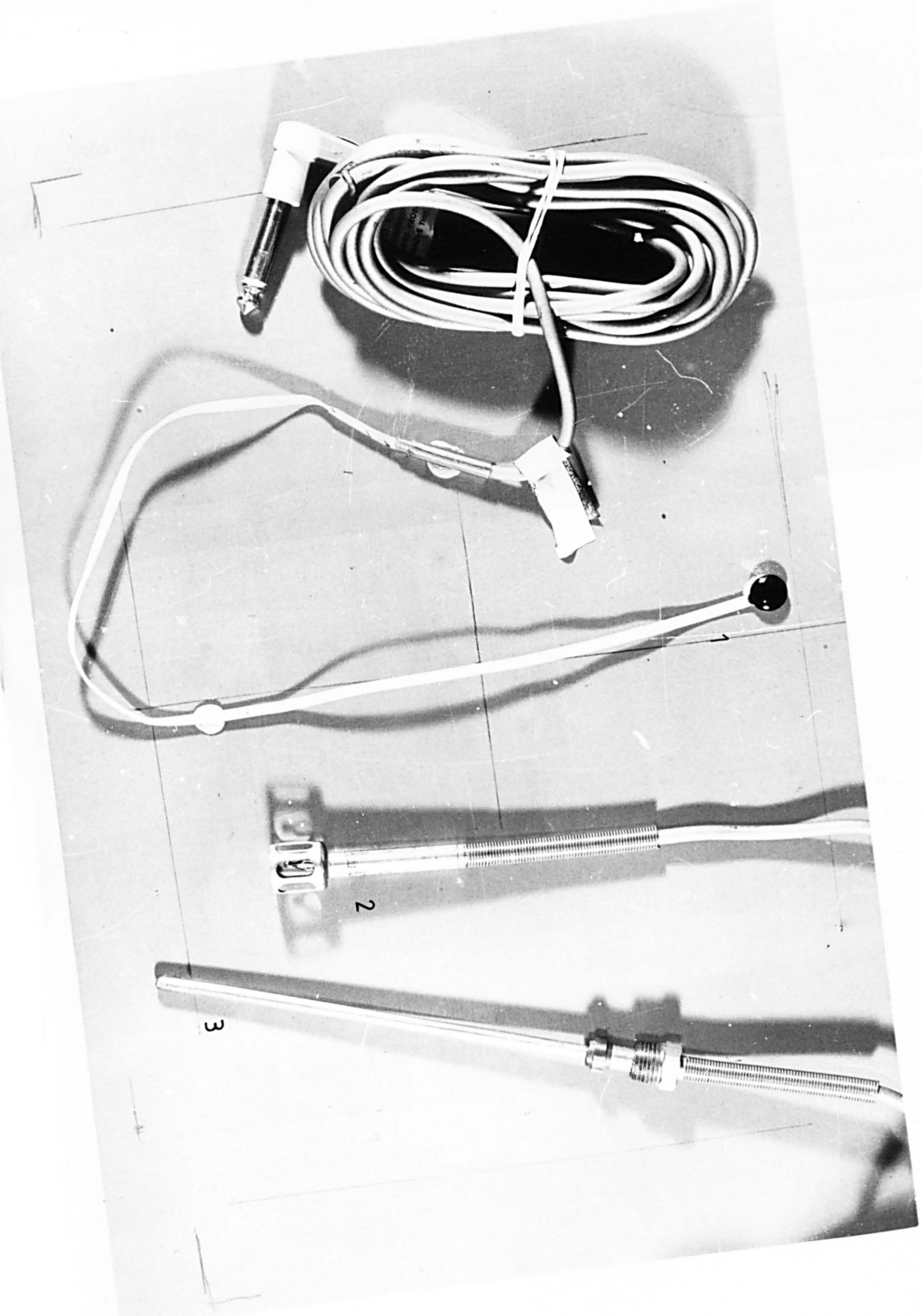
Plate 4

'YSI' thermistor probes

1 YSI model 409

2 YSI model 405

3 YSI model 403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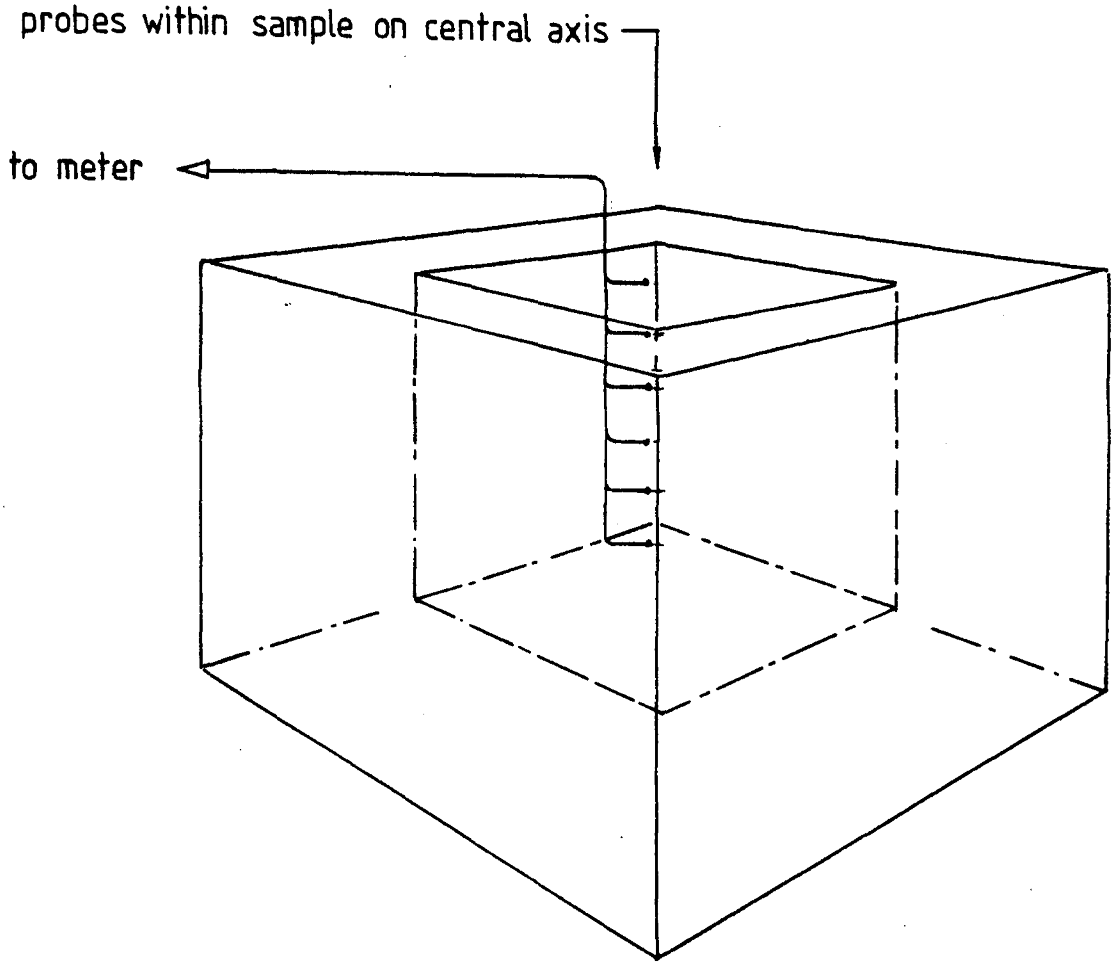


FIG 8·4a SEMI INSULATED CONTAINER FOR TRANSIENT DIFFUSIVITY MEASUREMENT

8.4.3 Operation

The coal sample spread in a tray was warmed in an oven for 24 hours at 60^{o}C as was the sample box. They were then removed and the coal was quickly transferred to the sample box. As it was filled the probes were carefully placed in their required positions specified by markings on the inside of the box. The probe wires were then securely taped to the side of the box to avoid moving them within the coal and the thermistor leads were connected to the meter.

The sample was allowed to cool naturally for a period to allow the required number of probes to fall within the range of the meter. The temperature of each probe in turn was then read at regular intervals and the results recorded numerically and graphically as shown in figures 8.4b, c and d. From the table of results the diffusivity is calculated as described in the next section. The table of results is for the first experiment with an uncovered sample of $-\frac{1}{4} + \frac{1}{8}$" dry sized coal. Four probes were used at depths of 0, 50, 100, 150 mm.

For any corresponding time, temperature and depth not on the edge of the table, the calculated diffusivity reading is shown in brackets under the temperature.

The test was repeated four more times on different samples and with 0.03 m probe spacings. For all the tests but one the coal was filled flush with the top of the box and exposed to the air. On one test the top surface of coal was covered with a cellophane sheet.

8.4.4 Treatment of results

The Fourier equation is

$$\frac{\partial t}{\partial \tau} = a \frac{\partial^2 t}{\partial x^2} \qquad (5)$$

Therefore

$$a = \frac{\partial t}{\partial \tau} \Big/ \frac{\partial^2 t}{\partial x^2} \qquad (6)$$

The differentials may be evaluated by finite difference method. At the i th line and j th column of the table:-

$$\frac{\partial t}{\partial \tau}_{(i)} = \frac{t_{i+1} - t_{i-1}}{2\Delta\tau} \qquad (7)$$

$\Delta\tau$ is the time between temperature measurements.

$$\frac{\partial^2 t}{\partial x^2}_{(j)} = \frac{t_{j+1} - 2t_j + t_{j-1}}{(\Delta x)^2} \qquad (8)$$

Δx is the distance between temperature measuring probes.

Therefore

$$a = \frac{t_{i+1} - t_{i-1}}{2\Delta\tau} \Big/ \frac{t_{j+1} - 2t_j + t_{j-1}}{\Delta x^2} \qquad (9)$$

So for example referring to table figure 8.4o the 'instantaneous' diffusivity of 15000 seconds and 0.05 m depth is

$$a = \frac{29.70 - 31.60}{6000} \Big/ \frac{34.20 - 2 \times 30.60 + 25.20}{0.05^2}$$

$$\underline{= 4.398 \times 10^{-7} \ m^2/s}$$

We can find the conductivity knowing the diffusivity, density and specific heat of the broken coal mix.

$$k = a \cdot \rho \cdot C \qquad (10)$$

where

k = Conductivity (W/m^oC)

a = Diffusivity (m^2/s)

ρ = Density (kg/m^3)

C = Specific heat ($J/kg\ K$)

Bulk density of $-\frac{1}{4} + \frac{1}{8}$" sized coal = 845 kg/m^3

Bulk specific heat = 1000 J/kg K

To calculate mean conductivity (apparent) test (table 8.4a).

Mean diffusivity = $3.27 \times 10^{-7}\ m^2/s$

$$\text{Mean conductivity} = k \cdot \rho \cdot C$$

$$= 3.27 \times 10^{-7} \times 845 \times 1000$$

$$= \underline{0.276\ W/m^oC}$$

For the ROM coal samples

Bulk density = 1070 kg/m^3

Bulk specific heat = 1000 J/kg K

8.4.5 Results

Figure 8.4b

Table of results for transient diffusivity test

Time (s)	Probe distance from surface (m)			
	0.0	0.05	0.10	0.15
	Temperature (^{o}C)			
0	27.70	38.75	42.85	43.25
3000	26.60	36.20 (3.7)	40.90 (3.4)	41.10
6000	26.10	34.35 (3.9)	39.10 (3.3)	39.20
9000	25.80	32.90 (4.1)	37.25 (3.5)	37.40
12000	25.55	31.60 (4.7)	35.60 (3.4)	35.90
15000	25.20	30.60 (4.4)	34.20 (3.4)	34.50
18000	25.00	29.70 (5.0)	32.90 (3.6)	33.30
21000	24.80	28.80 (6.3)	31.75 (3.4)	32.10
24000	24.55	28.10	30.80	31.10

Figures in brackets calculated diffusivity values (x 10^{-7} m^2/s).

Mean diffusivity = 4.01 x 10^{-7} m^2/s

Mean conductivity (apparent) = 0.339 $W/m^{o}C$

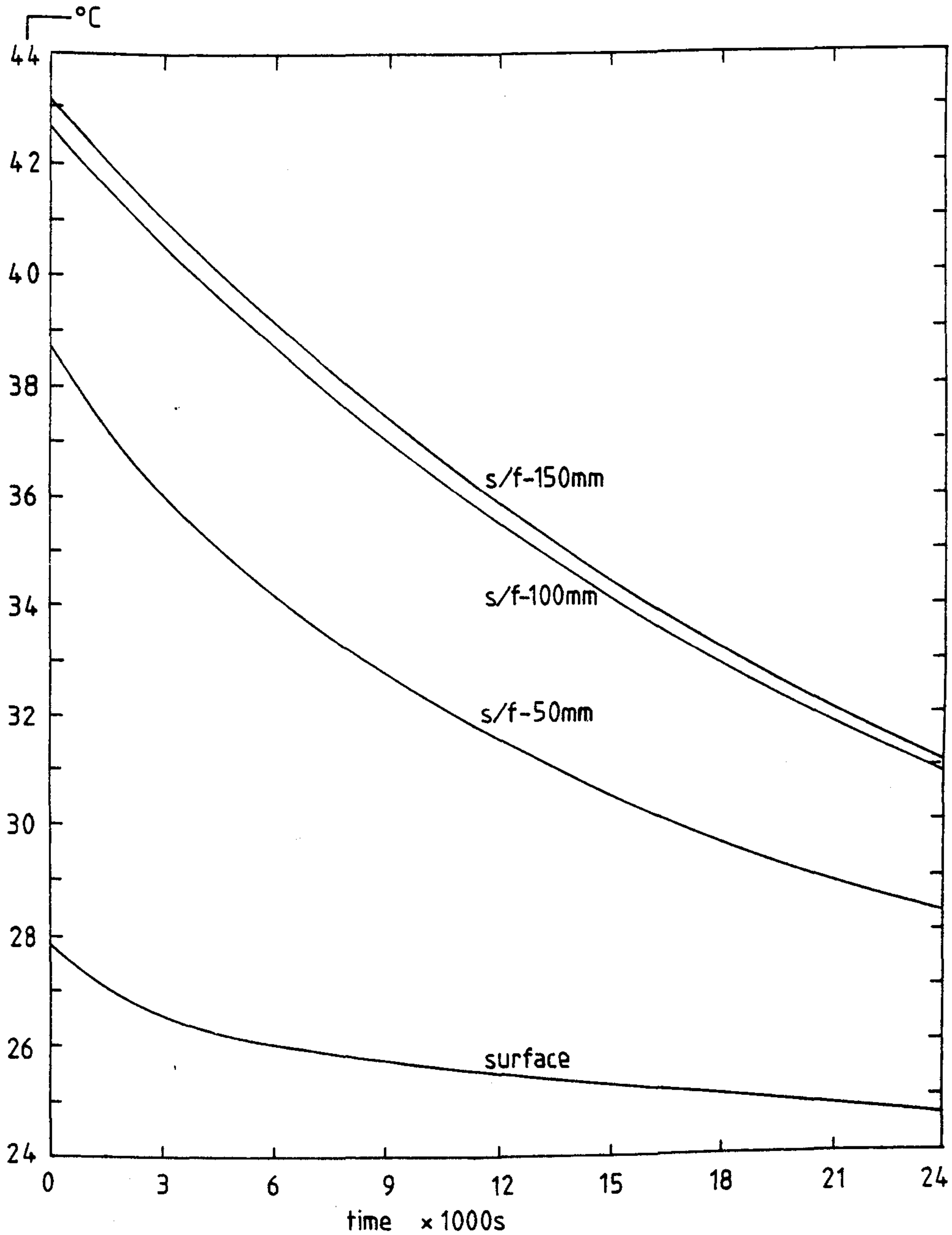


FIG 8·4c COOLING CURVES FOR COAL SAMPLE

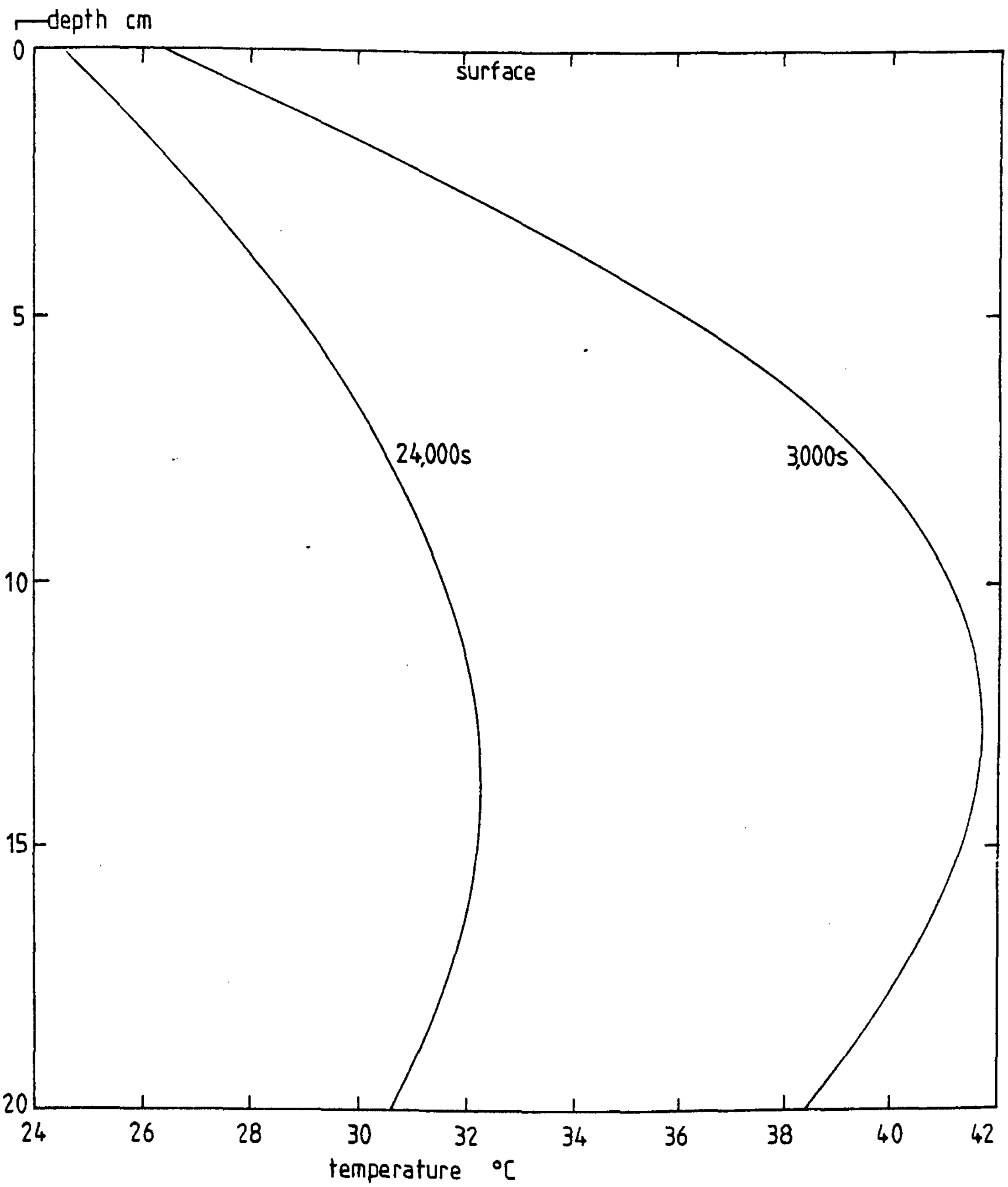


FIG 8.4d TEMPERATURE PROFILE IN COAL SAMPLE

Figure 8.4e

Table of results for transient diffusivity test

$-\frac{1}{4} + \frac{1}{8}$ dry coal, uncovered

Time (s)	Probe distance from surface (m) 0.0	0.3	0.05	0.09
	Temperature (°C)			
0	36.80	44.25	45.70	46.60
1800	34.90	42.80 (1.92)	45.60 (2.75)	45.90
3600	33.30	41.10 (1.20)	43.50 (9.10)	45.00
5400	32.40	39.80 (1.27)	42.30 (7.19)	44.00
7200	31.50	38.60 (1.33)	41.20 (6.39)	42.90
9000	30.80	36.30 (1.44)	38.90 (6.39)	40.60
10800	30.10	36.30 (1.46)	38.90 (6.11)	40.60
12600	29.50	35.30 (1.36)	39.80 (7.14)	39.60
14400	29.00	34.50 (1.45)	36.90 (4.29)	38.60
16200	28.60	33.70	36.00	37.70

Figures in brackets diffusivity values (x 10^{-7} m^2/s)

Mean diffusivity = 3.80 x 10^{-7} m^2/s

Mean conductivity (apparent) = 0.321 W/m^oC

Figure 8.4f

Table of results for transient diffusivity test

$-\frac{1}{4} + \frac{1}{8}$" dry coal, covered

Time (s)	Probe distance from surface (m) 0.0	0.03	0.06	0.09
	Temperature ($^\circ$C)			
0	30.90	42.20	45.75	46.60
1800	29.20	40.20 (1.45)	44.50 (2.14)	45.70
3600	28.20	38.30 (1.65)	43.16 (1.90)	44.50
5400	27.40	36.70 (1.69)	41.70 (1.51)	43.40
7200	26.86	35.46 (1.64)	40.50 (1.80)	42.20
9000	26.30	34.30 (1.72)	39.25 (1.93)	41.00
10800	25.90	33.30 (1.76)	38.00 (2.01)	39.90
12600	25.60	32.40 (1.82)	37.00 (2.27)	38.80
14400	25.20	31.70 (1.59)	36.00 (1.43)	36.80
16200	25.00	31.00	35.00	35.90

Figures in brackets diffusivity values (x 10^{-7} m^2/s)

Mean diffusivity = 0.18 x 10^{-7} m^2/s

Mean conductivity (apparent) = 0.152 $W/m^\circ C$

Figure 8.4g

Table of results for transient diffusivity test

ROM coal, dry, uncovered

Time (s)	Probe distance from surface (m)			
	0.0	0.03	0.06	0.09
	Temperature (^oC)			
0	35.90	41.60	43.60	44.20
1800	33.90	39.80 (2.50)	42.50 (3.60)	43.60
3600	32.30	38.40 (1.80)	41.30 (3.80)	42.70
5400	31.20	37.10 (2.32)	40.20 (3.09)	41.60
7200	30.35	35.90 (3.35)	39.20 (2.78)	40.70
9000	29.60	34.90 (2.25)	38.20 (2.79)	39.80
10800	29.00	34.10 (2.10)	37.30 (2.65)	38.80
12600	28.50	33.30 (2.06)	36.40 (2.83)	38.00
14400	28.10	32.70 (1.76)	35.60 (2.88)	37.20
16200	27.70	32.10	34.90	36.50

Figures in brackets diffusivity values (x 10^{-7} m^2s)

Mean diffusivity = 2.66 x 10^{-7} m^2/s

Mean conductivity (apparent) = 0.284 W/m^oC

Figure 8.4h

Table of results for transient diffusivity test

ROM coal, 5% moisture, uncovered

Time (s)	Probe distance from surface (m) 0.0	0.03	0.06	0.09
	Temperature (^oC)			
0	35.70	40.30	43.10	45.90
1800	34.50	38.90 (4.64)	41.90 (5.00)	43.70
3600	33.50	37.70 (4.58)	40.70 (5.00)	42.60
5400	32.70	36.70 (4.75)	39.70 (4.38)	41.50
7200	32.00	35.80 (4.50)	38.60 (5.00)	40.40
9000	31.40	34.90 (6.07)	37.70 (4.50)	39.50
10800	30.90	34.10 (8.00)	36.80 (5.00)	38.60
12600	30.40	33.30 (11.16)	35.90 (5.31)	37.70
14400	30.00	32.70 (9.17)	35.10 (5.36)	36.80
16200	29.50	32.10	34.40	36.00

Figures in brackets diffusivity values (x 10^{-7} m^2/s)

Mean diffusivity = 5.78 x 10^{-7} m^2/s

Mean conductivity (apparent) = 0.618 W/m^oC

8.4.6 Discussion

The test values for apparent thermal conductivity of broken coal are as follows

$-\frac{1}{4} + \frac{1}{8}$" dry	0.339 and 0.327 W/m^{o}C
$\frac{1}{4} + \frac{1}{8}$" dry (covered)	0.152 W/m^{o}C
ROM, dry	0.284 W/m^{o}C
ROM + 5% moisture	0.618 W/m^{o}C

Examination of the temperature profile of the first test (figure 8.4c) shows that heat did not leave the sample only through the top surface. The temperature gradient toward the bottom of the container indicates that some heat flowed downwards. It is therefore reasonable to assume that heat also flowed radially. The temperature gradient towards the base though was much less steep than toward the surface indicating less heat flow downwards than to the surface. To overcome the problems of radial heat flow there are several solutions. Improved insulation and lower thermal capacity of sample box would result in less heat transfer to and through the container. The sample container used could not be made of a significantly better material at reasonable cost so this was not a practical solution.

A much larger sample could have been used which would result in heat flow near a plane surface being almost perpendicular to it or the same size sample used and a thinner section near the surface monitored as it cooled. This is the approach which was adopted and the probe spacing was reduced so that only the top 90 mm of the sample was monitored. A single test on a large sample yielded no better results despite the inconvenience of heating and handling a larger sample.

It is noticeable that the values for the conductivity measured by this method were higher than the values obtained by the steady state method. The conductivity should not vary when the measuring method is changed so the values derived by the transient method should be qualified as apparent or equivalent conductivity as they are the ones less likely to be definitive values. The most likely reason for these higher values is due to convective heat transfer taking place within the sample and not just at the top surface resulting in air movement within the sample. It can be seen that when a sample was covered restricting convection currents its apparent conductivity fell by more than 50% from 0.321 to 0.152 W/m^{o}C. The coal sample of $-\frac{1}{4} + \frac{1}{8}$" sized coal has a void age ratio in the order of 50% so it is quite likely that the air movement within it was significant.

The ROM sample with 5% moisture had a high apparent conductivity probably due to latent heat transfer as well as the convection effects mentioned. This would result in an unknown moisture content and distribution after time zero.

Although the conductivity values are apparent and varied with time and depth within the sample the experiment was still judged worthwhile as indications were given of what happens actually in the coal bed on a conveyor. Whilst the apparent conductivity is not claimed as a definitive value it is more representative of the real situation. The next Chapter describes the evaluation of a model length of coal and conveyor in a duct which hopefully should get closer still to representative results.

8.5 THEORETICAL PREDICTION OF VALUES OF BULK THERMAL CONDUCTIVITY

8.5.1 Introduction

The problems of measurement of thermal conductivity for a multiphase substance have arisen in the past, particularly with measurements of conductivity on porous rocks. The effects of porosity or voidage in this context is of interest due to changes which occur when confining pressure in situ due to overburden is released and samples are tested under different conditions in a laboratory. Naturally cracks may open when confining pressure on a rock is lowered and the effects of varying voidage have to be evaluated.

Research by Walsh and Decker [22] in the USA was carried out on the effects of pressure and saturating fluid on the thermal conductivity of rock. It was concerned mainly with low porosity granite and the equations given here should be used with caution on a high porosity broken coal. Nevertheless they are included here due to the interesting results the equations yielded. Also the work that Walsh and Decker based their study on could yield more specialised equations for high porosity materials like broken coal. The original study was carried out by Hashim and Schtrikman [23] and applied to magnetic permeability, but they claim the methods used to predict upper and lower bounds of permeability of multiphase substances may also be used for other characteristics, thermal conductivity included.

Walsh and Decker's approach was as follows. For evaluating the effect of pure fluid (gas or liquid) the material can be approximated to an isotropic material containing pores. Thus the problem is one of finding the effective conductivity of a composite material with two isotropic phases,rock and pore fluid.

Two procedures are possible. Upper and lower bounds on the bulk thermal conductivity may be established. Alternatively a procedure exists to calculate a discrete value of bulk thermal conductivity.

8.5.2 Bounds of bulk conductivity

Assuming the rock matrix has a thermal conductivity (k) and the pore fluid with volume concentration (voidage, porosity etc) (N) has a conductivity (k'). The maximum and minimum values for the effective (bulk) conductivity can be expressed as

$$\frac{k - k_e}{k} = 3N\ (1 - E)/(2 + E + N) \qquad (11)$$

for the maximum bound, and:-

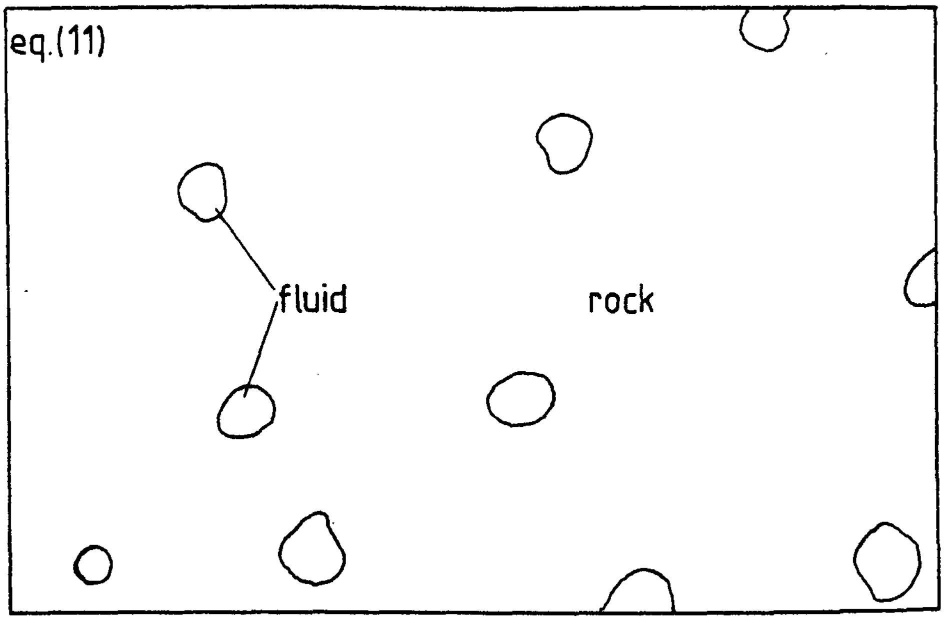
$$\frac{k - k_e}{k} = N\ (1 + 2E)(1 - E)/(3E + N - NE) \qquad (12)$$

for the minimum bound

where

$$E = \frac{k'}{k}$$

The situations represented by equations (11) and (12) are a continuous solid phase with fluid inclusions and a continuous fluid phase with solid 'inclusions' respectively the latter situation being representative of the broken coal. See figure 8.5a.



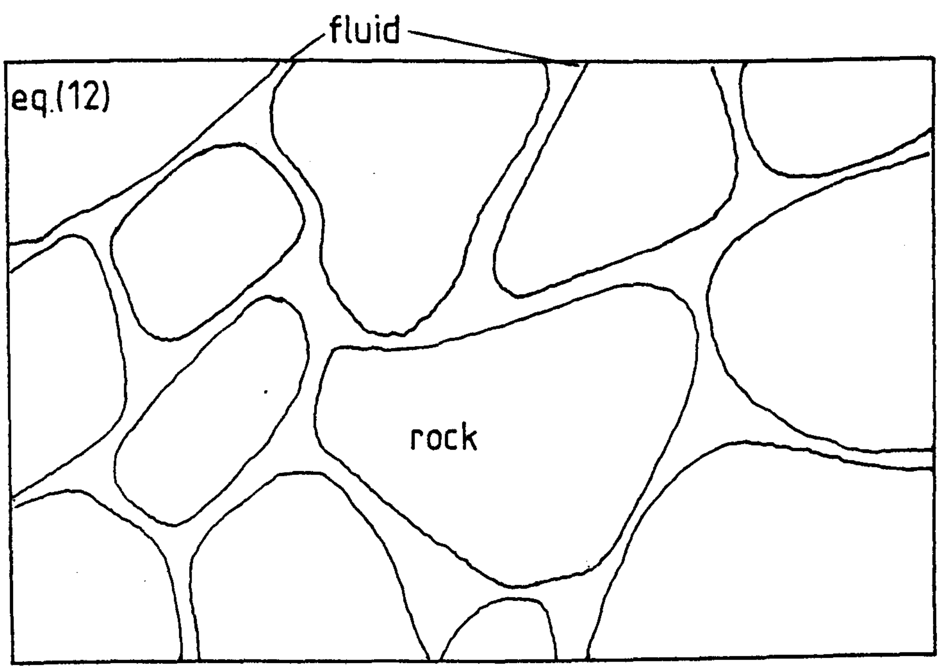


FIG 8·5a PORE CONFIGURATIONS APPLICABLE TO EQUATIONS (11) & (12)

An attempt to evaluate the bulk thermal conductivity of broken coal by inserting some known values into the equations (11) and (12).

Thermal conductivity of solid coal $k = 0.25$ W/m^{o}C

Thermal conductivity of air $k' = 0.028$ W/m^{o}C

Ratio of conductivities $k'/k = E = 0.112$

Voidage of $-\frac{1}{4} + \frac{1}{8}$ coal = 0.5

Voidage of ROM coal = 0.3

The following values for thermal conductivity are given.

$-\frac{1}{4} + \frac{1}{8}$" upper bound $k_e = 0.122$ W/m^{o}C

lower bound $k_e = 0.076$ W/m^{o}C

ROM upper bound 0.167 W/m^{o}C

lower bound 0.115 W/m^{o}C

8.5.3 Actual effect of pore fluid

As mentioned in section 8.5.1 a second method is available which predicts an actual value between the bounds. It is described by equation (13).

$$\frac{k - k_e}{k} = N/3E + N \qquad (13)$$

Insertion of the values quoted for sized and ROM coal gives values for equivalent conductivity of:-

$-\frac{1}{4} + \frac{1}{8}$" coal $k_e = 0.100$ W/m^{o}C

ROM coal $k_e = 0.132$ W/m^{o}C

Values actually measured (section 8.3)

$-\frac{1}{4} + \frac{1}{8}$" sized coal	$k = 0.102$ W/m^{o}C
ROM coal	$k = 0.118$ W/m^{o}C

8.5.4 Discussion

Comparison of the results of the steady state tests for thermal conductivity of ROM and sized coal with the values predicted by equations (11), (12) and (13) shows good agreement. Despite the equations being designed primarily for low porosity rocks the measured values fell between the upper and lower bounds and close to the actual predicted values.

The original work on prediction of characteristics of multiplane substances by Hashim and Shtrickman [23] could yield equations which are more applicable to substances with high porosity and would hopefully provide more accurate predictions. The work so far carried out only applies to a material with one fluid in its pores. It cannot include a mixture, of air and water simultaneously so to include this some conductivity value for air and water must be estimated. Whilst this analytical method has yielded good results for dry substances the problems of evaluation of some equivalent conductivity for the situation of the coal on a conveyor still remain.

CHAPTER 9

MODEL OF A CONVEYOR IN A DUCT

CHAPTER 9

MODEL OF A CONVEYOR IN A DUCT

9.1 BASIC CONCEPT

The laboratory model of a length of conveyor should be compatible with the theory so far described, and as representative as possible of the real underground situation. With due consideration to the practicalities of building, instrumenting and operating such a model at reasonable cost. The configuration shown in outline in figure 9.1a was chosen. For compatability with theory the following conditions were assumed.

The coal was regarded as a plane parallel slab and the sides of the slab ignored.

Heat and mass were assumed to pass only through the end planes of the control zone. How these two conditions were met is described later.

Rather than have an actual conveyor in a control zone it was judged sufficient to have a stationary model of a section of conveyor and provide 'movement' by variation of the airspeed. Such a stationary section of conveyor with its coal could easily be loaded and instrumented as required. The model conveyor was constructed in a duct which allowed almost a full scale model thereby overcoming the problems of similarity encountered with small scale models.

The duct which was well insulated had air forced along it by a variable speed centrifugal fan. The wet and dry bulb temperatures

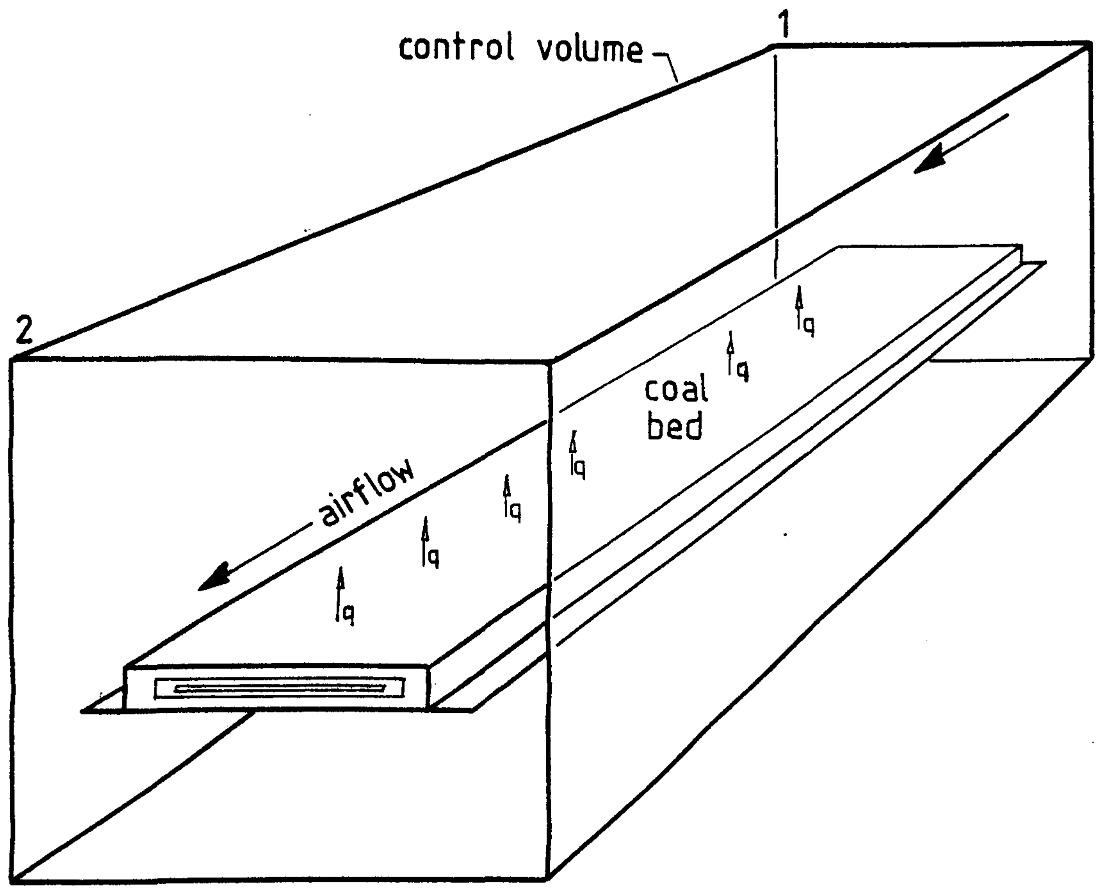


FIG 9·1 a MODEL OF COAL ON CONVEYOR IN AN AIRSTREAM were measured at the inlet and outlet of the control zone and these allowed, given a barometric pressure, the heat in the air at either end of the duct and hence the heat gain due to the coal to be evaluated. This enabled a balance of air heat gain against coal heat loss to be attempted.

The coal could not be loaded and instrumented quickly enough, to monitor its cooling, by any practical method available so it was left in the duct and heated by passing hot air over it to reach a uniform temperature. Cool air was then passed over it and the rate of cooling monitored.

This then was the basic model and mode of operation. The next section describes the choosing, designing and building of the individual components.

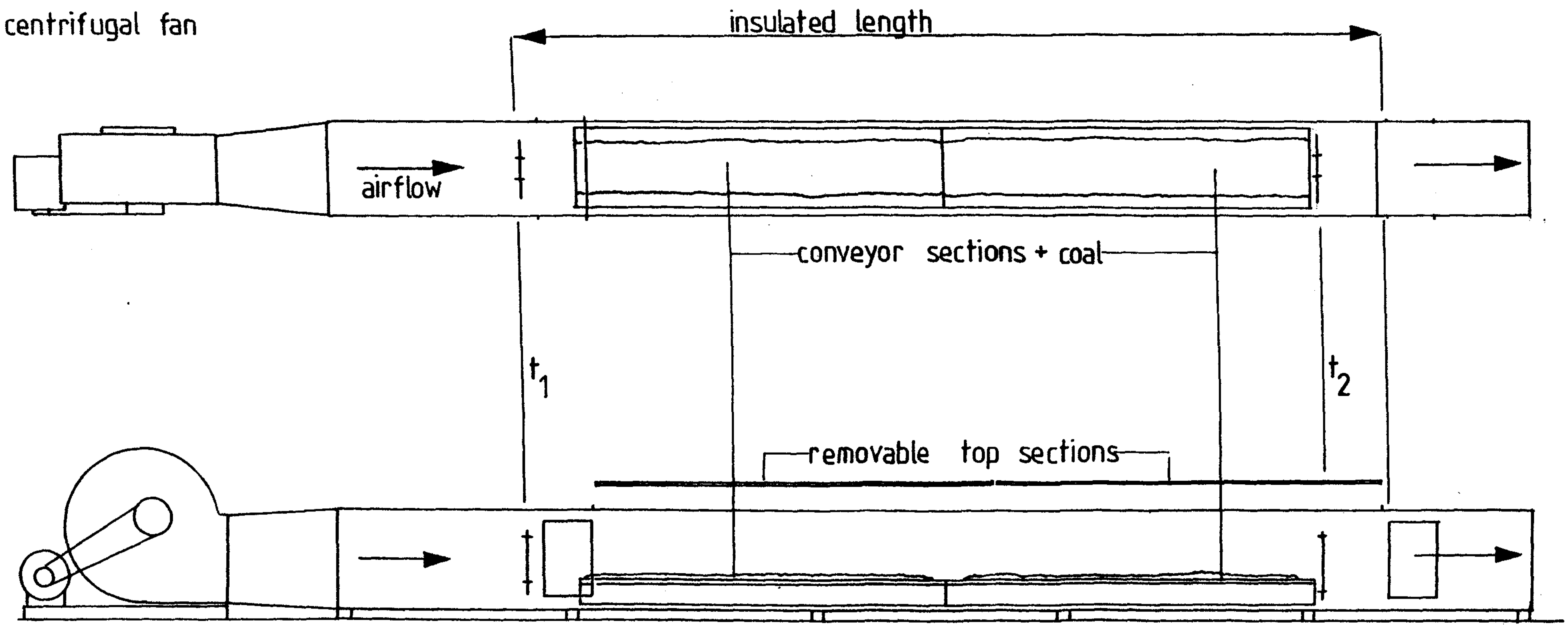
9.2 EQUIPMENT

9.2.1 Duct

The model was constructed in a 10 m long by 0.762 m square ventilation duct in the University of Nottingham, Mining Laboratory. The duct was of plywood on a 50 mm square wooden external framework. The construction was modular, the duct being bolted together in large sections, hence the top was easily removable for access. Four hatches in the sides positioned as shown in figure 9.2a allowed quick access whilst the duct top sections were bolted down. Each hatch was secured by four quick release clamps and sealed by a felt gasket to stop air leakage. The insides of the hatches were flush with the inside walls of the duct.

FIG 9·2a LAYOUT OF VENTILATION DUCT AND MAIN EQUIPMENT

SCALE 1:50



The air was forced through the duct by a 1.5 m diameter centrifugal fan. This was powered by a variable speed motor which allowed any airflow between 0 to 15 m/s to be selected. Between the fan transition piece and the duct was a honeycomb screen flow straightener.

Before any experiments the duct which was not designed for heat measurement work had to be insulated along the test length. This was to ensure that any heat change measured in the air passing through the duct would be attributable wholly to the coal rather than having some unknown and difficult to evaluate, quantity of heat passing through the walls. The inside of the duct was lined with three layers of 1.5 mm polystyrene sheet and covered with polythene. This was held in place with drawing pins which were taped over to prevent tearing. The end result was a tight smooth 'quilt' of laminated polystyrene.

The outside of the duct was insulated with 50 mm of fibre glass wool contained in the box sections formed by the external framework. This was held in place by polythene sheet stapled around the whole duct.

9.2.2 Conveyor sections

Two 3 m long sections of static model conveyor were constructed to carry the coal in the insulated duct. The section length was dictated by the 7 m long removable duct top and the requirements of light weight and ease of handling due to restricted crane access to the duct. Each section could easily be lifted into the duct by one person at each end. The framework was constructed from punched steel strip (Dexion) and the coal was carried on a piece of conveyor belting cut down to 650 mm width. The belt was at a height of 250 mm and flat in profile with the edges upswept a further 50 mm to retain the coal, whilst allowing a coal bed of near constant thickness to be constructed.

Originally it was intended to allow the air passing over and below the belt to mix and measure only one set of temperatures at the duct outlet. During construction of the experiment it was decided that keeping the airflow above and below the belt separate could prove useful. Whilst complicating temperature and velocity measurements this would allow the heat flow from the coal surface and that from the underside of the belt to be separately evaluated. To keep the airflows separate a polythene sheet seal was fitted between the belt edges and the duct wall. It was taped to the duct walls permanently and at the free side tucked tightly between the framework and conveyor belt. This formed an adequate seal to separate the airflow but still allowed access to tubes and wires running under the belt to the instruments at the inlet to the control zone.

The coal itself was ordinary unsized ROM from a local colliery (Pye Hill). It was placed on the conveyor as required from pre-weighed polythene sacks containing 30 kg each. The coal was then spread on the belt taking care to form the plane parallel slab of coal whilst still keeping a naturally rough surface.

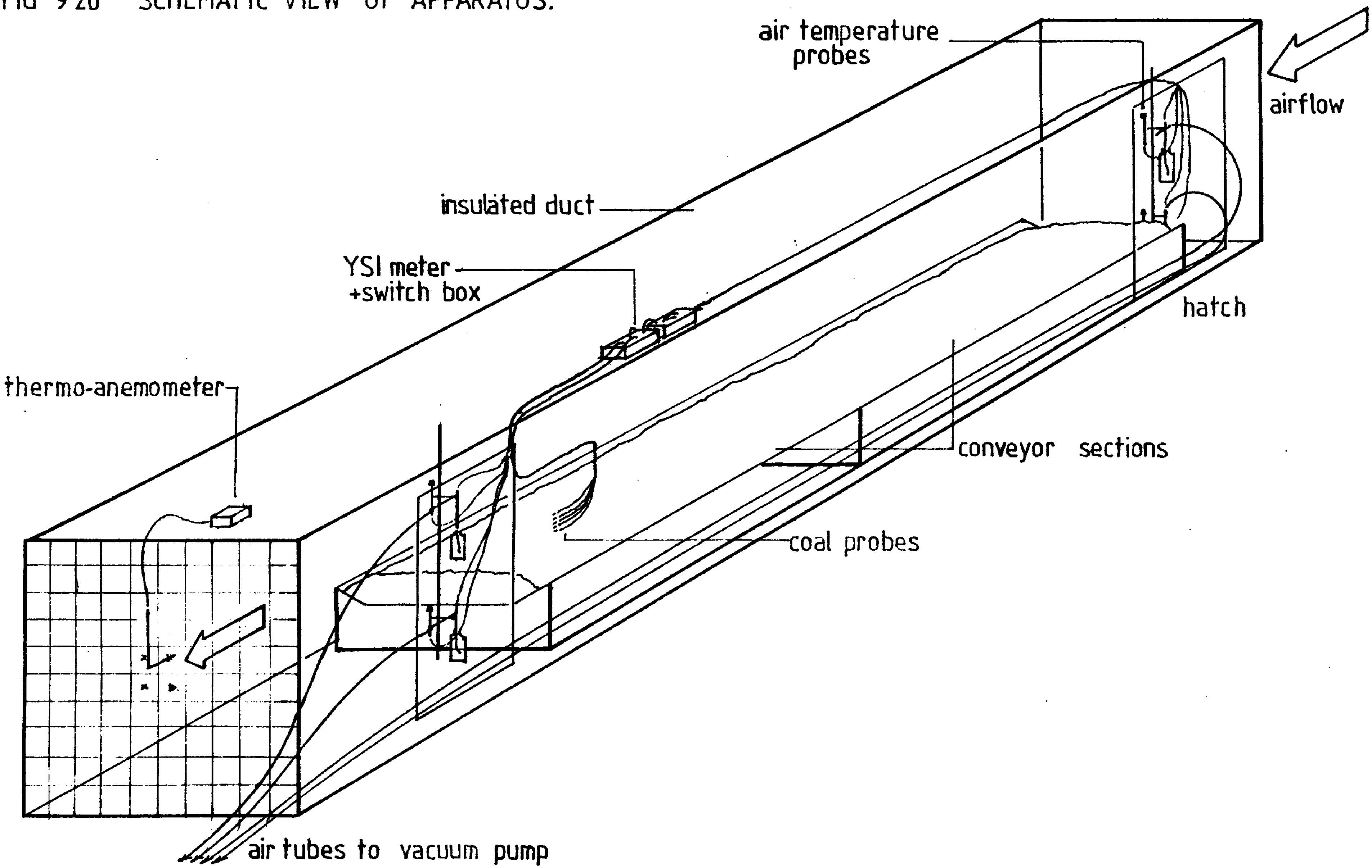
9.2.3 Inlet heater

A temperature difference between the air and coal was a prerequisite of this experiment. A system was needed which would allow air at a lower temperature to be passed over the coal bed. A refrigeration system could have been used but would have been difficult to procure and set up. It was decided at an early stage to use a heater system on the fan inlet to warm incoming air which would be passed along the duct over the coal for some period until the coal was uniformly warmed. When such a state was reached the inlet heaters were switched off and ambient air passed through the duct. This resulted in almost a step change in the air temperature and the experiment proper could begin.

For reasons described later the fan inlet was chosen as the best location for the heater. A visit to a specialist suppliers of heating elements resulted in the purchase of four 2.5 kW elements, the maximum safe temporary power supply being 10 kW. The elements were contained in asbestos insulation in a flexible 7 mm diameter, 2 m long steel tube with a terminal at each end. These elements were bent into a spiral pattern and mounted on the inside of the fan inlet grid using steel and asbestos brackets. The power supply for the central element was the 240 V main. The outer three elements were connected in 'star' configuration supplied from the 3 phase main.

The elements were mounted in the fan inlet to provide easy access and connection. The elements could be easily watched for overheating and the fire risk in a steel fan casing was much lower than in the wooden duct. Air heated in the fan inlet was also well mixed in the fan resulting in an even temperature distribution in the duct.

FIG 9 2b SCHEMATIC VIEW OF APPARATUS.



9.3 INSTRUMENTATION

9.3.1 Air dry bulb temperature

This was measured at the inlet and outlet of the control zone above and below the belt level, four places in all. Measurement presented no special problems as equipment designed specifically for that purpose was available. The transducers used were Yellow Springs Instrument Co. thermistor probes. The model YSI 405 (air temperature). The thermistors were connnected via the switch box to the meter allowing direct reading of the temperature. This system has already been described in Chapter 8.4.2.

The thermistors were mounted in position in the duct on clamp stands. Both the dry and wet bulb probes were positioned such that the air temperature was measured in the top and bottom halves of the duct at the beginning and end of the conveyor length. At the inlet end this allowed a constant check that the air was well mixed and no temperature stratification was taking place.

At the downstream end, the probes at upper and lower levels, which were close to the end of the conveyor, measured the temperatures of two separate airstreams to allow evaluation of the heat pick up from above and below the conveyor.

9.3.2 Air wet bulb temperature

The remote measurement of wet bulb temperature posed many problems and several systems were tried before a suitable method was found. In Chapter 3.7 it was mentioned that a wet bulb temperature measuring device has two important requirements. A wet surface and a steady airflow over the probe or thermometer.

With regard to the wetting of the wick the water supply must be perfectly regulated and at the right temperature. If water is supplied too quickly the water temperature will influence the measurement making some unknown contribution. If water is not supplied quickly enough, not enough evaporation can take place at the transducer or thermometer surface and a temperature somewhere between wet and dry bulb is measured.

The airflow over the wetted wick is necessary to keep evaporation at a steady maximum level. Smaller transducers need smaller air velocities but generally if the air velocity is not sufficient the required rate of evaporation cannot take place and once again a temperature between dry and wet bulb is measured. At too high an air velocity kinetic effects cause a slight heating effect which rises as the velocity increases. The ideal air velocity over the well wetted wick is recognised as 2 - 3 m/s.

The range of air velocities expected in the duct was in the region of 0 - 8 m/s. At the higher velocities the air would be moving sufficiently fast for full evaporation to take place, but at the low velocities, less than 3 m/s, the transducers would need to be artifically aspirated. In order to satisfy the above, the following system was designed.

The temperature transducer used was the YSI 403 model. This was in the form of a thermistor mounted in a stainless steel tube. It was mounted into a 10 mm diameter plastic tube using quick fit hydraulic pipe 'T' piece. The plastic tube extended 20 mm beyond the end of the probe as shown in figure 9.3a. Air was drawn up the tube, over the probe and out through the T piece which was connected to a pipe from a vacuum pump. Four such devices were connected to

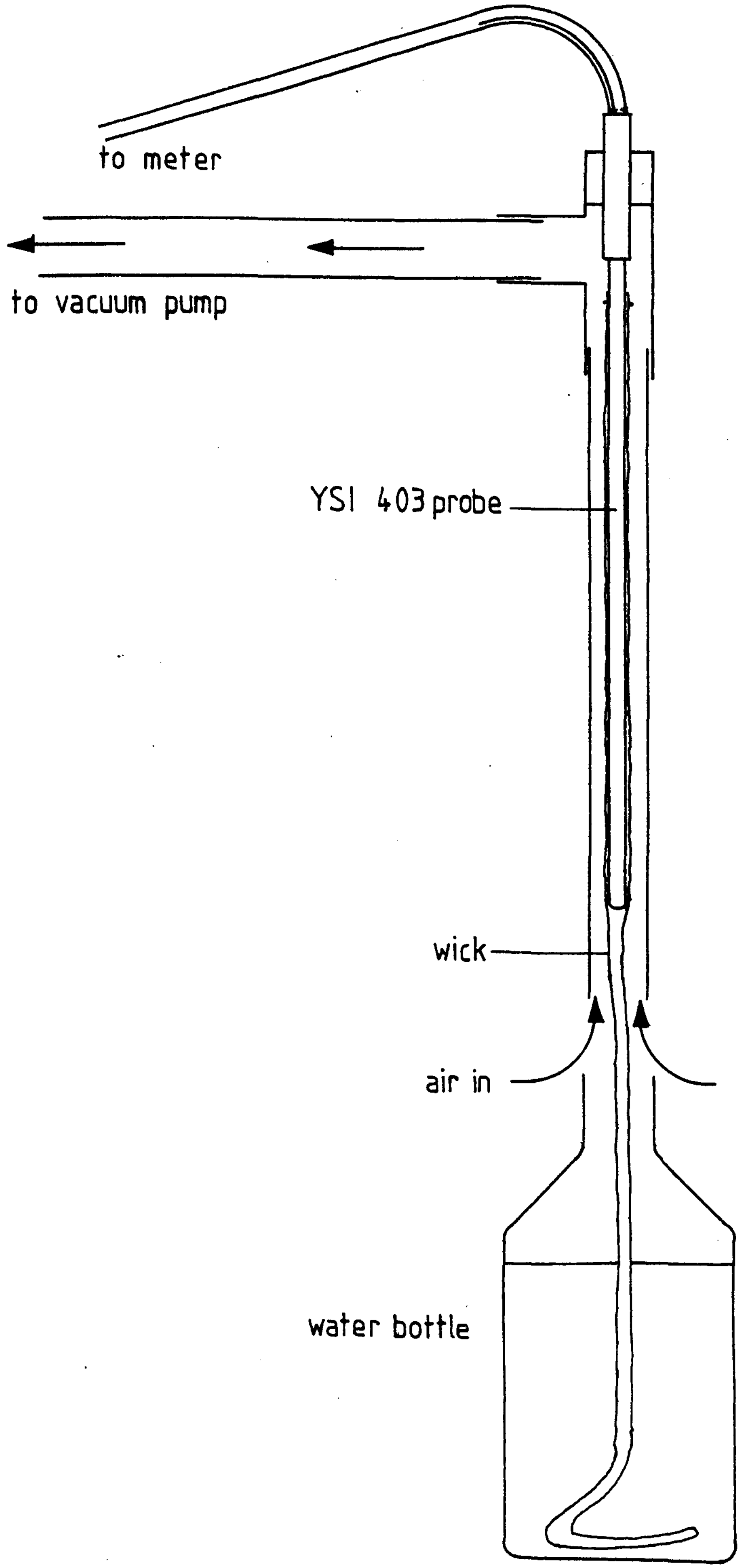


FIG 9·3a WET BULB TEMPERATURE PROBE

the pump, two at the inlet and two at the outlet of the test length in the duct. The connecting tubes were all of different lengths and so to ensure the same airflow in each a regulating device was necessary. This took the form of two pieces of steel bolted round the pipes to flatten them to regulate the resistance. Tightening the bolts flattened the pipe and reduced the airflow. Each pipe was adjusted in turn to give a 3 m/s velocity airflow over the probe. This was measured using a Wallac thermoanemometer held to the intake pipe whilst the adjustment was taking place. Once set up and checked the airflow needed no further adjustment.

The thermistor probe was converted to read air wet bulb temperature by threading a wick over it right to the T pieces. Each wick was about 250 mm long. To keep the wick properly wetted several designs were tried where water was fed down tubes from reservoirs outside the duct. It was hoped to site the water reservoirs outside the duct for convenience of filling and monitoring the levels. However the system proved difficult to regulate and unreliable. The water supply to the wicks eventually used was a 100 ml plastic bottle of water mounted just below the air intake to the measuring device. The wick was dipped into the bottle which was narrow necked to minimise evaporation losses. Although being inside the duct made the water bottles less accessible, once filled they provided water for a whole day in even the hottest conditions. The thermistor probe was connected via the switch box to the YSI meter and with the water bottle filled and vacuum pump running reliable wet bulb temperature reading could be taken regardless of air velocity in the duct.

9.3.3 Air velocity and quantity

The air velocity distribution in the duct had to be carefully checked before any temperature measurements could be taken. A complete set of velocity contours and profiles were measured so that the flow characteristics of the duct were fully understood and a convenient representative measuring point chosen.

All velocity measurements were made using thermo-anemometers to give instant numerical readouts. Two models were used. For the compilation of the velocity contours and profiles an expensive and accurate 'Alnor' was used and for the experimental work, where only measurements on the duct centre line were required, the 'Wallac' model. Before and after this work both were calibrated against a pitot tube and Betz micromanometer set in a low turbulence duct designed expressly for anemometer calibration.

Reference to figure 9.2a shows that air passing along the duct was divided by the conveyor and followed two separate flow paths before mixing again downstream of the conveyor. The air was deliberately kept separate at the top and underside so that the two heat gains could be separately evaluated. Naturally the air quantity and velocity above the belt would be greater than that travelling through the path below the belt with many supports forming obstructions.

To avoid making a complete set of velocity measurements each time the fan was run at a different speed, a series of velocity traverses above and below the belt and at the duct end were carried out and a single reading on the duct centre line assigned to each. Thus a graph was compiled which for a given centre line reading indicated the flow in various parts of the duct. This was facilitated as follows.

The air velocity traverses were carried out in three places, at the outlet end of the duct and above and below the belt at the first access hatch. The outlet end of the duct was divided into 100 (10 x 10) equal sized squares by a set of fine wires to aid accurate traversing. To make a traverse, the fan speed was set and allowed to settle to a steady air flow. Checks of stable fan speed were made at intervals using a revolution counter on the shaft. In each of the 100 squares of the duct end a velocity reading was made and recorded. The squares around the edge having a mean reading recorded by subdividing the square into 9 (3 x 3) smaller squares to give an accurate value where the velocity gradient was greatest. The mean of the readings assigned to each of the 100 squares then gave the mean value for the whole duct. The anemometer probe was held in a clamp stand which was moved about and adjusted to place the measuring tip accurately where needed.

The traversing procedure was repeated inside the duct above and below the conveyor at the first access hatch. The probe tip was carefully positioned by measurement in imaginary squares as no positioning wires existed here. A mean air velocity above and below the belt was derived from the mean reading similar to the duct end previously described.

Thus at a single fan speed a mean velocity was found for the whole duct where the flow had mixed, and for the paths above and below the conveyor. A note was also made of the mean velocity of the four squares on the duct centre line. This procedure was repeated for a range of fan speeds and a graph of the duct velocity distribution compiled (figure 9.3b).

The graph is based on a coal bed of 0.5 m width and 0.075 m depth on the conveyor giving the following cross sectional area

CSA of duct	0.5805 m^2
CSA of conveyor and coal	0.0440 m^2
CSA of top air path	0.3515 m^2
CSA of bottom air path	0.1851 m^2

For the rest of the experiments when only the velocity or air quantity in the duct was required measurements were taken on the four grid squares on the duct centre line and the mean velocity calculated. The airflow distribution graph was then used to arrive at the velocities present in the various sections of the duct. For example

Centre line mean velocity 5 m/s from graph

Duct mean velocity	3.87 m/s
Top path mean velocity	5.05 m/s
Bottom mean velocity	2.55 m/s

Velocity x CSA = Volumetric flow rate

Top path	$5.05 \times 0.3515 = 1.775$ m^3/s
Bottom path	$2.55 \times 0.1851 = 0.472$ m^3/s
Whole duct	$3.87 \times 0.5806 = 2.246$ m^3/s

Ideally the volume and velocity should have been corrected for the slightly different coal bed depths but the effect was too small to measure in a turbulent airstream and a theoretically calculated

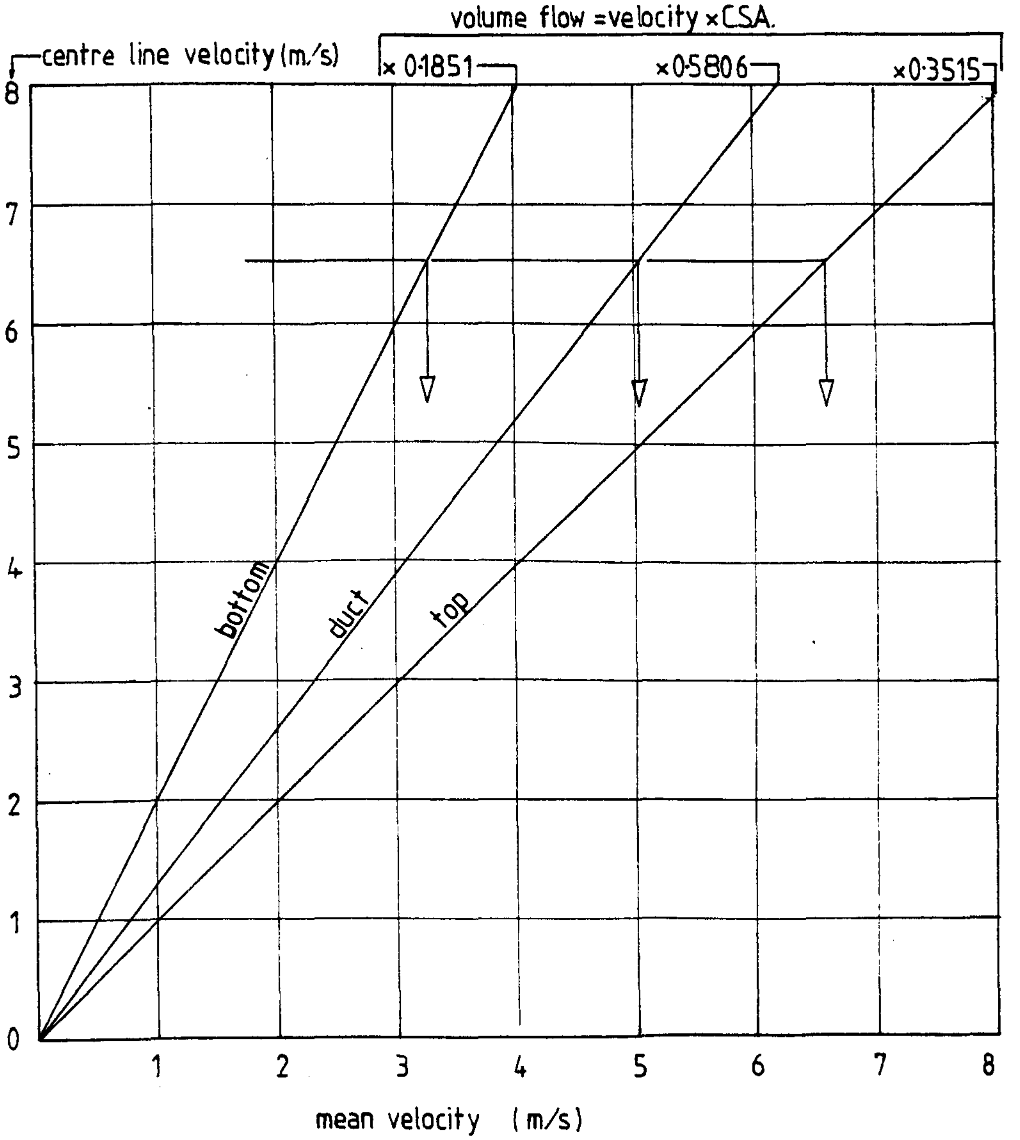


FIG 9·3b AIRFLOW DISTRIBUTION IN DUCT

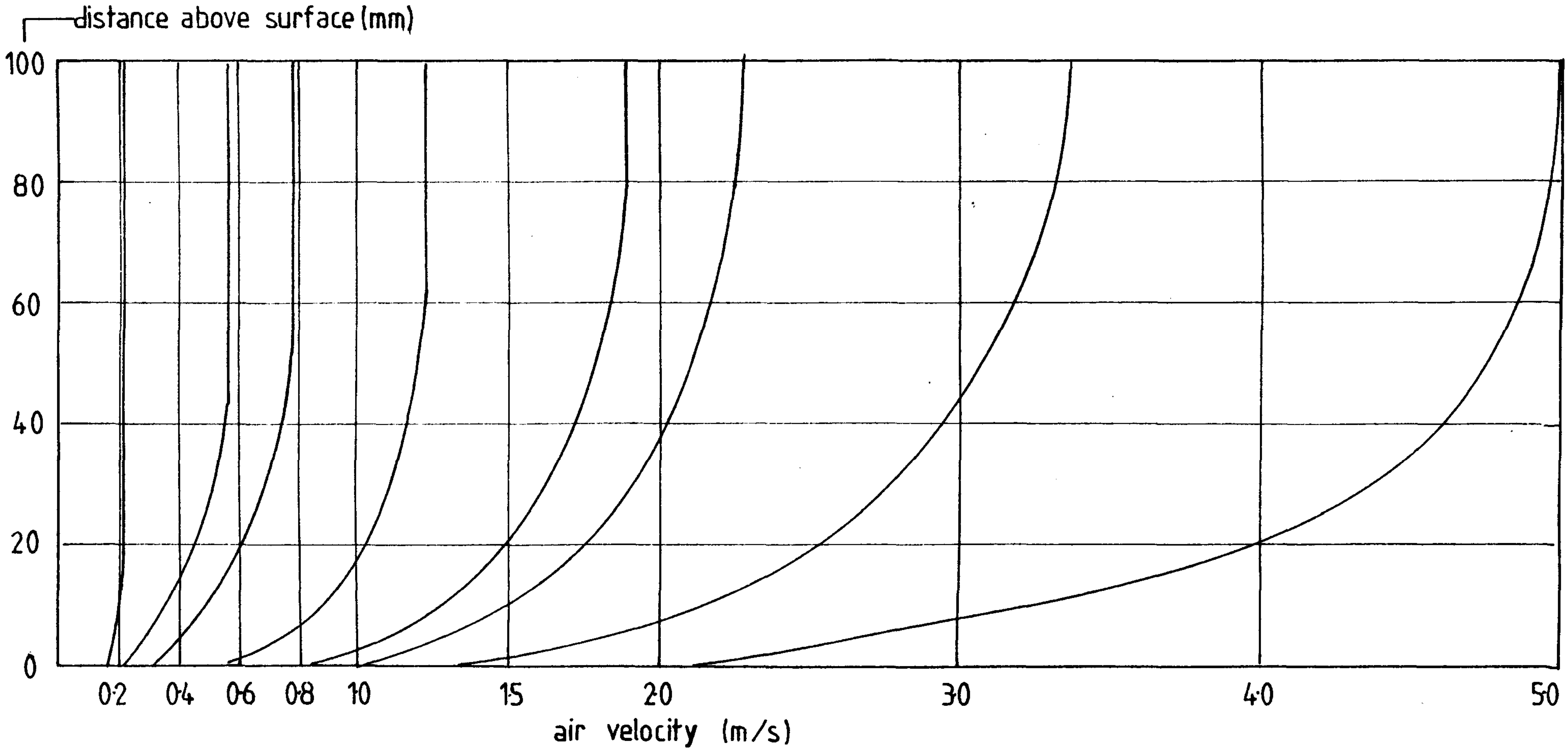


FIG 9·3c VELOCITY PROFILE ABOVE COAL SURFACE percentage correction less than the airflow measuring accuracy.

Whilst the 'Alnor' anemometer was available a detailed study of the airflow over the rough coal surface was made. The probe on the 'Alnor' was 500 mm long and 3 mm in diameter and so allowed readings close to a surface to be made. The air velocity profiles are shown in figure 9.3c.

9.3.4 Coal temperature

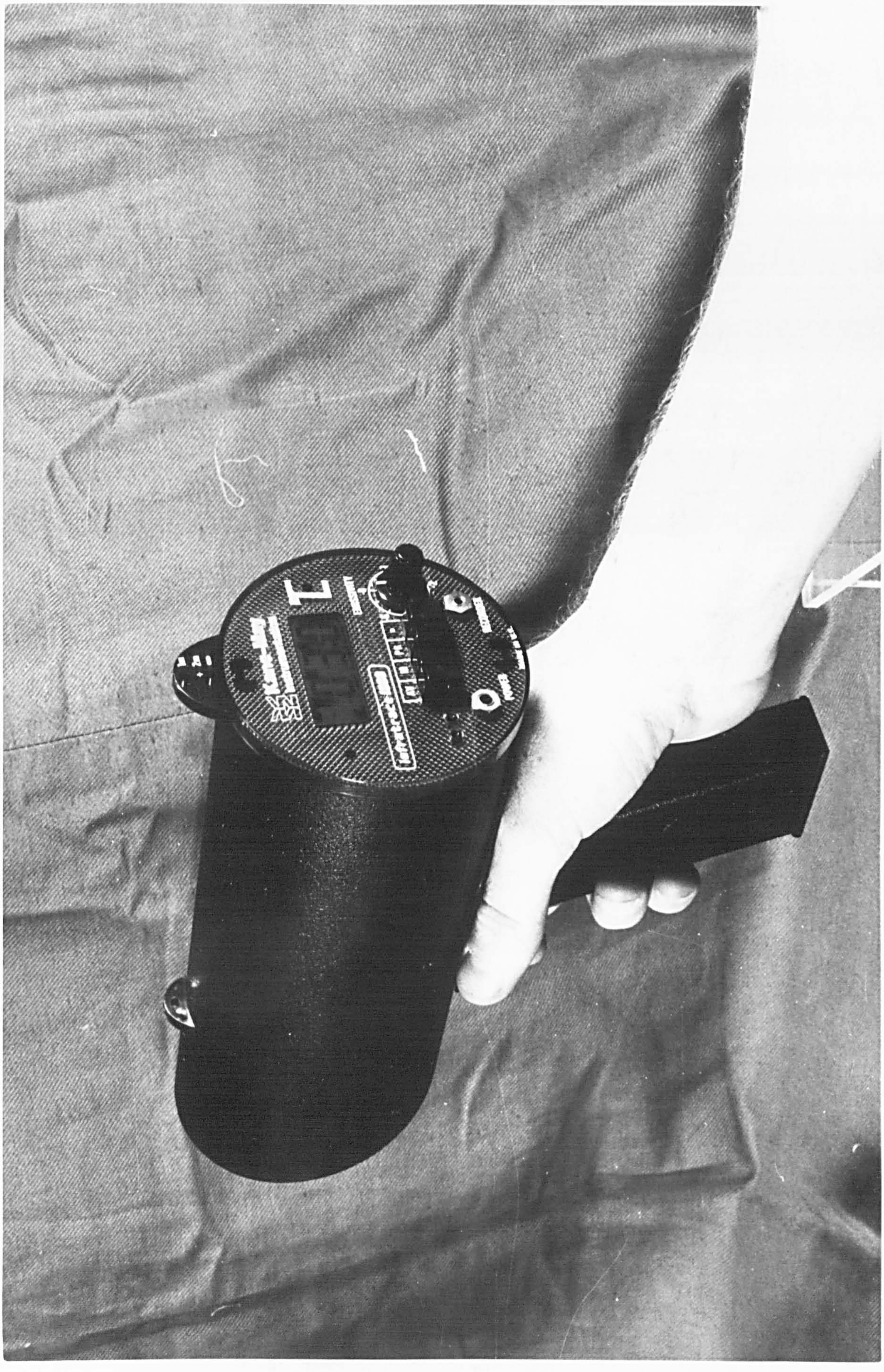
The temperature of the coal was measured using YSI 409 thermistor probes connected, via the switch box, to the direct reading meter. The probes were buried in the coal where required. Great care was taken to accurately position and bury the probes in the coal bed. The leads were well taped down to avoid the airstream moving the probes near the surface. The probes were not placed directly above each other but staggered within the coal so that they could be placed accurately without disturbing each other and any small contribution such a device would have on the heat flow within the coal bed would be well distributed.

9.3.5 Surface temperature

Surface temperature measurements on the coal were made using an infrared, non-contact thermometer. The specifications and instructions for the instrument, the 'Infratrace', are reproduced in full in the appendix (A4). Briefly the instrument, resembling a pistol in shape measured the temperature of the surface it was pointed at when the trigger button was pressed. The temperature of the surface was shown on a digital display on the instrument.

Plate 5

Infrared thermometer 'Infratrace'



Before use the emissivity* of the target surface was set using the dial on the instrument. This necessitated prior knowledge of the emissivity of the surface. Previous calibration and checking of the instrument before experimental use revealed that the emissivity control could be set at 1.0 for use on a rough coal surface. Although the instrument displayed in increments of $1^{o}C$, a scale expansion facility could be used to allow measurement of differences of $0.2^{o}C$ by setting the emissivity control at 0.2 for a surface with a true emissivity close to 1.0. In such a state the instrument would display temperature changes of $0.2^{o}C$ in $1.0^{o}C$ increments on the display. This meant that the temperature reading was not absolute but this facility was useful to follow closely the temperature of a cooling surface. Such readings could be referred to on occasional absolute reading with the emissivity control at 1.0.

In use the 'Infratrace' was mounted on a camera stand and aimed down the duct mouth to a particular area of coal surface. The target surface being a large elipse, 100 x 600 mm. It was connected to the 240 V mains via a transformer and was operated continuously for stability of readings whilst experiments were in progress.

9.4 OPERATION

Before the main experiment could be started the coal was warmed until it reached the required temperature. This was carried out by using the inlet heaters to blow hot air along the duct. The heaters were always run on full power of 10 kW and the temperature regulated by changing the fan speed and hence the air

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* See Chapter 5.4 Radiative heat transfer.

quantity absorbing the fixed amount of heat. As the coal always started the tests dry, the air dry bulb temperature dictated the coal temperature.

The coal was heated until the probes in the coal all showed the same temperature within 0.5°C. Ideally they should eventually all have been the same, but this did not always occur, probably due to slight air temperature variations in the laboratory and hence a slightly fluctuating heating temperature. Where possible this system was operated during holiday periods and weekends whilst the laboratory was subject to less temperature variations due to opening doors and the operation of other equipment, notably fans. The coal temperature was generally within the limits stated after 24 hours of heating.

Before a cooling run was started all the equipment was switched on and checked and wet bulb thermometer reservoirs topped up. If all equipment was in order the heaters were switched off and the fan quickly reset to give the required airflow. No time was available to set the fan exactly to the required air velocity. The fan speed control unit gave a rough idea what could be expected and a precise check was made later. If cold inlet air was needed the large sliding laboratory doors were opened and as the tests were conducted in winter temperatures in the order of 10°C could be reached. This procedure took 3-5 minutes and partly due to air in the laboratories taking time to mix with incoming air and also because some heat was stored in the fan causing the step change in inlet air temperature was not attained. However a large temperature difference was realised quite quickly as shown in the results.

The temperatures of the coal and air were read and recorded as quickly as possible as the coal cooled. Initially the interval between readings was five minutes, but as the cooling slowed down the time intervals between readings was increased. A set of readings took about two minutes and the order of reading was reversed each time so that when the results were graphed they would straddle a quickly changing curve. Continuous recording equipment would have been desirable here but was not available.

Two sets of experiments were undertaken. One set with a bed of coal of 160 kg which was 6 m long x 500 mm wide x 50 mm deep. The coal probes were in two sets of three near each end of the conveyor at depths of 1, 25 and 50 mm. The second set of experiments used 320 kg of coal and had a 100 mm deep bed with the length and width being unchanged. For this the probes were arranged in a single set at depths of 1, 20, 40, 60, 80 and 99 mm.

Twelve cooling runs were undertaken with each coal load (160 and 320 kg) using a range of air velocities. The number of instruments read and frequency of reading generated large amounts of data. Rather than present all this data and the cooling curves for each experiment, which vary only slightly, one typical run is treated in detail complete with its sample calculations. The other results are summarised and the various trends and effects illustrated with reference to the worked example.

9.5 PROCESSING OF RESULTS

9.5.1 Classification of results

The results may be divided into three main groups as follows:

1. Basic information which is fixed for a single experiment. Barometric pressure, air velocity, probe placement, specific heat of coal etc.

2. Measured values which varied and were measured through the experiment. Air probe temperatures, coal probe temperatures etc.

3. Derived values which are derived from the previous two groups. Conductivity, heat flows, heat transfer coefficients etc.

The worked example which follows is subdivided as above.

9.5.2 Basic information

Air conditions

Barometric pressure	101.1 kPa
Air velocity (centre)	0.951 m/s

Coal

Mass			320 kg
Bed dimensions			6 x 0.5 x 0.1 m
Probe number	1	depth	1 mm
	2	"	20 mm
	3	"	40 mm
	4	"	60 mm

	5	depth	80 mm
	6	"	99 mm
Bulk density			1070 kg/m^3
Specific heat			1.13 kJ/kg K

(The specific heat was derived before this experiment in a calorimeter using method of mixtures.)

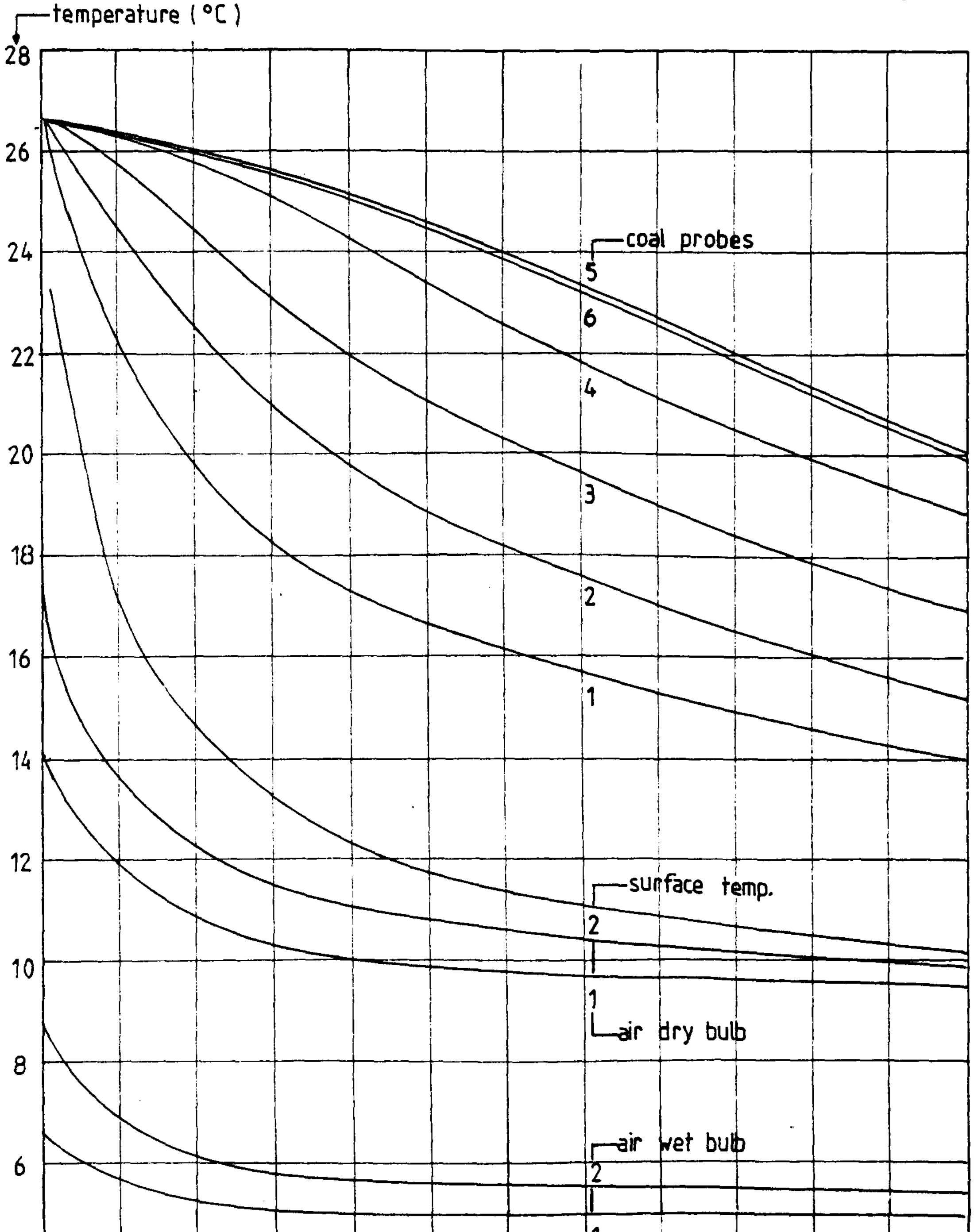
9.5.3 Measured values

Figure 9.5a

Table of results for coal cooling in 1 m/s airstream

Time (min)	0	5	10	15	20	30	40	60	80	100	120
t_{wb1} (°C)	6.4	6.0	5.6	5.3	5.2	5.0	5.0	5.1	5.2	5.2	5.2
t_{db1} (°C)	14.2	12.5	11.8	11.2	10.8	10.2	10.0	10.2	9.8	9.6	9.6
t_{wb2} (°C)	8.9	7.3	6.9	6.3	6.0	5.8	5.7	5.7	5.6	5.4	5.4
t_{db2} (°C)	17.3	15.0	13.8	12.9	12.3	11.4	11.0	10.5	10.4	9.9	9.7
Probes											
1 (°C)	26.6	24.6	22.3	20.9	19.8	18.4	17.3	16.2	15.3	14.5	14.0
2 (°C)	26.6	25.7	24.5	23.4	22.6	21.0	19.8	18.1	17.0	16.1	15.2
3 (°C)	26.6	26.3	25.6	25.0	24.5	23.0	22.0	20.3	19.0	17.9	17.0
4 (°C)	26.6	26.5	26.3	26.0	25.7	25.1	24.3	22.6	21.2	20.7	18.9
5 (°C)	26.6	26.5	26.4	26.2	26.0	25.7	25.2	24.0	22.7	21.4	20.2
6 (°C)	26.6	26.5	26.4	26.1	25.9	25.5	25.0	23.8	22.5	21.2	19.9
Surface (°C)	24.0	20.0	17.2	15.6	14.6	13.4	12.6	11.4	11.0	10.4	10.2

FIG 9·5b COAL COOLING CURVES AND AIR TEMPERATURES



9.6 DERIVED INFORMATION (WORKED EXAMPLE)

9.6.1 Heat pick up by air

The heat gain of the air travelling over the top of the coal and conveyor is dealt with here. The heat gain of air travelling underneath the conveyor proved too small to measure reliably (see discussion 9.8.3). The heat gain of the air passing over the coal is the product of the sigma heat change of the air entering and leaving the control zone and the air mass flow rate. The sigma heat is specified at a given pressure by the wet bulb temperature and was evaluated using the psychrometric computer program "PS". The air mass flow is derived from the centre line air velocity measured and the air density once again taken from "PS". An example follows.

Worked example at time = 0 minutes

Air mass flow

Centre line velocity		0.951 m/s
Mean velocity above conveyor (from duct velocity distribution graph)		0.963 m/s
Volume flow rate		0.338 m^3/s
Air density at duct exit		
wet bulb temperature	8.9°C	
dry bulb temperature	17.3°C	
pressure	101.1 kPa	
from "PS"		1.209 kg/m^3
Air mass flow		
volume flow x density		
0.338 x 1.209		0.409 kg/s

Heat gain by air

Inlet conditions (1)	
Wet bulb temperature	6.4°C
Dry bulb temperature	14.2°C
Sigma heat (from "PS")	21.26 kJ/kg
Outlet conditions (2)	
Wet bulb temperature	8.9°C
Dry bulb temperature	17.3°C
Sigma heat (from "PS")	26.52 kJ/kg

Heat gain by air =

(Sigma heat out - Sigma heat in) x Air mass flowrate

$$(26.52 - 21.26) \times 0.409 = 2.15 \text{ kW}$$

The heat gain by the air was so derived for all the readings of the experiment to yield the results shown in table 9.6a.

9.6.2 Heat lost by coal

Several methods of deriving this information were considered. Although the answer should be independent of the method used to process the raw data consideration had to be given to the amount of subsidiary information and its possible use later to derive other values. For example taking the mean of the six probe readings at each time increment and using this to evaluate the rate of cooling could give the heat loss rate. However it would give no information on what was happening within the coal bed. The method chosen divided the coal bed into 5 layers of equal thickness (20 mm) and found the rate of cooling of each layer in each time interval. From this the net heat loss rate in each layer could be found. Totalling the heat loss for each layer gave the heat loss for the whole coal bed. The method is best illustrated by example. From table 9.5a the temperatures of the coal probes at 5 and 10 minutes are so shown.

Probe	5 minutes	10 minutes	Heat loss (kW)
1 layer 1	24.6 (24.85)	22.3 (23.4)	0.349
2 layer 2	25.7 (26.0)	24.5 (25.05)	0.229
3 layer 3	26.3 (26.4)	25.6 (25.95)	0.108
4 layer 4	26.5 (26.5)	26.3 (26.35)	0.036
5 layer 5	26.5 (26.5)	26.4 (26.4)	0.024
6	26.5	26.4	
		TOTAL	0.746

The mean temperature in each layer is found by interpolating between the upper and lower probe temperature giving the temperature shown in brackets. Thus the mean temperature of each layer at a given time is known.

The net heat loss in each layer is equal to the temperature fall multiplied by the specific heat and mass. Dividing by the time taken gives the rate of heat loss.

For example

layer 1	Temperature at 5 minutes	$24.85^{\circ}C$
	Temperature at 10 minutes	$\underline{23.40^{\circ}C}$
	Temperature fall (Δt)	$1.45^{\circ}C$

Mass of coal layer (m) 320/5 = 64 kg

Specific heat of coal (C) 1.13 kJ/kg K

Time interval (τ) 5 min = 300 s

$$\text{Heat loss rate } Q = \frac{m.C.\Delta t}{\tau}$$

$$= \frac{64 \times 1.13 \times 1.45}{300}$$

$$= \underline{\underline{0.349 \text{ kW}}}$$

This procedure is followed for each layer in the coal bed hence the heat loss for the whole coal bed may be evaluated by adding the net heat loss for all five layers.

This procedure is repeated at each time interval of the cooling run to give a cooling rate for the coal bed for the whole 120 minute period. The results shown in figure 9.5a were processed similarly to give the table, figure 9.6a, and the graph, figure 9.6b.

The mean heat gain of the air was found from the 11 values from 0 to 120 minutes and the mean heat gain of the coal found from the 10 relevant values. The purpose of this was to assign a single numerical value to the heat transferred to the air and from the coal for each experiment. This allowed a comparison of the results to be made more easily.

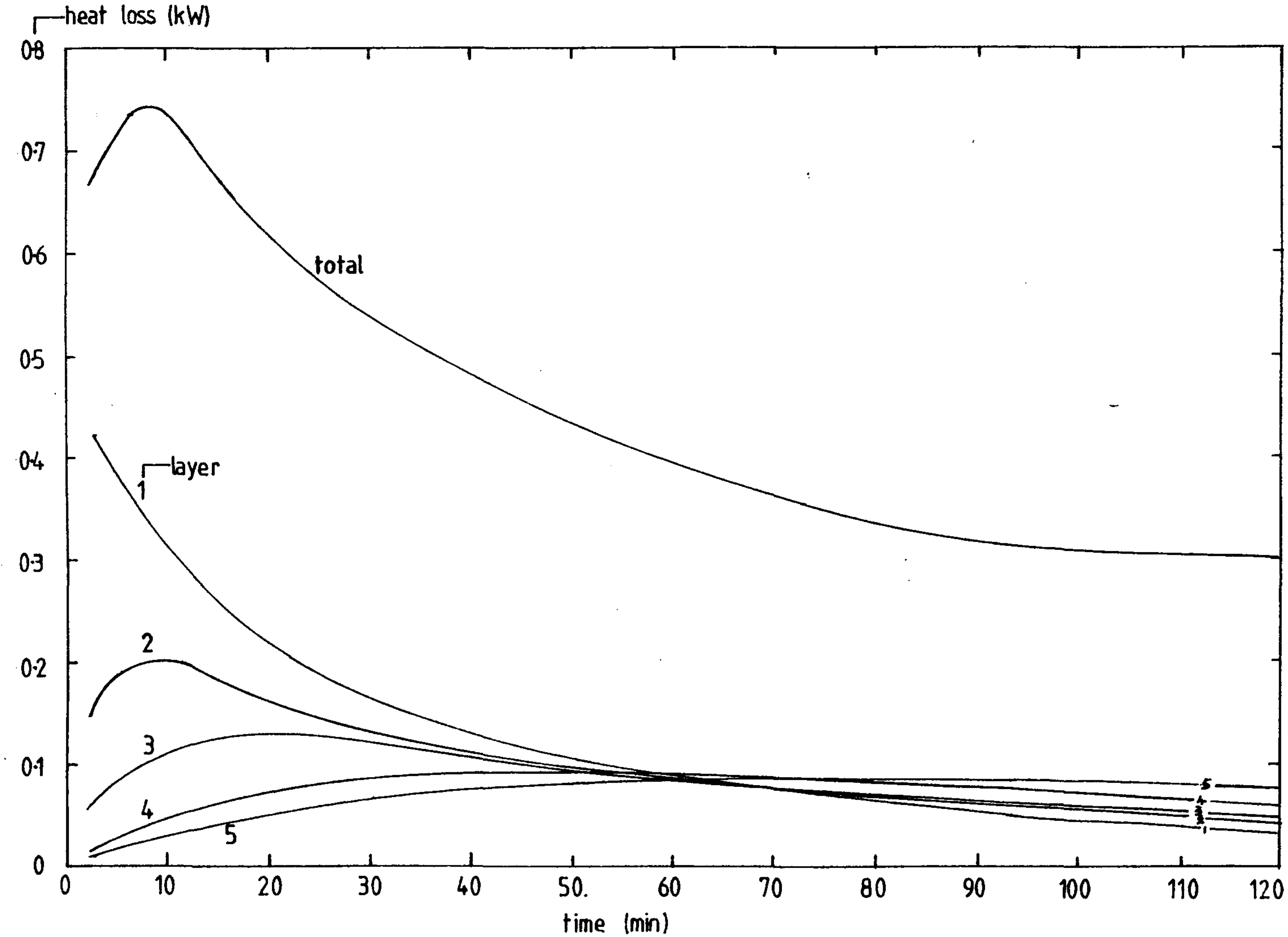
Figure 9.6a

Table showing heat pick up by air and heat lost by coal in 1 m/s airstream

Time (min)		0	5	10	15	20	30	40	60	80	100	120
Sigma heat 1 (kJ/kg)		21.26	20.45	19.66	19.07	18.88	18.49	18.49	18.68	18.89	18.88	18.88
Sigma heat 2 (kJ/kg)		26.52	23.11	22.28	21.06	20.46	20.06	19.86	19.86	19.67	19.27	19.27
Air heat gain (kW)		2.151	1.088	1.072	0.814	0.646	0.642	0.560	0.483	0.319	0.159	0.159
Coal heat loss (kW)												
Probe	Layer											
1												
	1	0.422	0.349	0.277	0.253	0.181	0.139	0.084	0.078	0.033	0.042	
2												
	2	0.145	0.229	0.205	0.156	0.187	0.133	0.102	0.073	0.060	0.054	
3												
	3	0.048	0.108	0.108	0.096	0.126	0.108	0.102	0.027	0.069	0.060	
4												
	4	0.024	0.036	0.060	0.060	0.042	0.090	0.087	0.082	0.075	0.069	
5												
	5	0.024	0.024	0.060	0.048	0.042	0.060	0.072	0.078	0.078	0.075	
6												
	TOTAL	0.663	0.746	0.710	0.613	0.578	0.530	0.447	0.338	0.315	0.300	

Mean air heat gain = 0.735 kW   Mean coal heat loss = 0.524 kW   Balance 71%

FIG 9 6b HEAT LOSS RATE FROM COAL BED



A percentage heat balance was also produced by dividing the mean heat loss from the coal by the heat gained by the air.

9.6.3 Convective heat transfer coefficient

A convective heat transfer coefficient was evaluated using the information given in tables 9.5a and 9.6a and b together with the air dry bulb temperature and cooling rate of coal respectively. As described in more detail in the discussion the search for temperatures which revealed a pattern on which to base a coefficient indicated the air dry bulb and the number 1 probe. The method of deriving the coefficient is shown once again by an example.

Referring to Chapter 5.3 the heat transfer by convection is given as

$$Q_c = h_c . A\ (t_s - t_a) \qquad (1)$$

where

Q_c = Heat transferred by convection (W)

h_c = Convective heat transfer coefficient ($W/m^{2\,o}C$)

A = Area of surface involved (m^2)

t_a = Air temperature (^oC)

t_s = Surface temperature (^oC)

Rearranging (1) we have

$$h_c = \frac{Q_c}{A\ (t_s - t_a)} \qquad (2)$$

hence the convective heat transfer coefficient may be calculated.

For example @ time of 10 minutes.

Air, mean dry bulb temperature in duct (t_a)

$$\frac{t_{db1} + t_{db2}}{2} = \frac{11.8 + 13.8}{2} = 12.8^oC$$

Probe temperature (t_s)	22.3^oC
Temperature difference ($t_s - t_a$)	9.5^oC
Heat transfer from coal at 10 minutes (Q_c) (from table 9.6a or graph 9.6b)	0.740 kW
Area of surface (A)	3 m^2

Using equation (2)

$$h_c = \frac{Q}{A\ (t_s - t_a)}$$

$$h_c = \frac{0.740}{3 \times 9.5}$$

$$= 0.026\ kW/m^{2o}C$$

Repeating at intervals enables series of values to be calculated.

Figure 9.6c

Heat transfer coefficients at airflow 0.95 m/s

Time (min)	Air temperature (°C)	Surface temperature (°C)	Difference (°C)	Heat Transfer (kW)	Heat transfer coefficient ($kW/m^2 °C$)
10	12.8	22.3	9.5	0.740	0.026
20	11.55	19.8	8.25	0.635	0.026
30	10.8	18.4	7.60	0.540	0.024
40	10.5	17.3	6.80	0.480	0.023
60	10.25	16.2	5.95	0.380	0.021
80	10.1	15.3	5.20	0.330	0.021
100	9.75	14.5	4.75	0.310	0.022
120	9.65	14.0	4.35	0.300	0.022

kW/m²°C

Mean convective heat transfer coefficient = 0.023 ($W/m^2 °C$)

9.6.4 Thermal diffusivity and conductivity

As stated in Chapter 8 the thermal diffusivity of coal may be evaluated in a dynamic cooling situation and the conductivity derived from it. The procedure used to evaluate the diffusivity, the finite difference method, was described in Chapter 8. This is demonstrated in the following worked example.

From table 9.5a the temperature readings of probes 2, 3 and 4 at 10, 15 and 20 minutes are

		Time (min)		
		10	15	20
Probes (°C)	2	24.5	23.4	22.6
	3	25.6	25.0	24.5
	4	26.3	26.6	25.7

The thermal diffusivity a is given as

$$a_{ij} = \frac{t_{i+1} - t_{i-1}}{2\Delta\tau} \Bigg/ \frac{t_{j+1} - 2t_j + t_{j-1}}{(\Delta x)^2}$$

where

$\Delta\tau$ = Time interval (300 s)

Δx = Probe spacing (0.02 m)

So for probe 3 at 15 minutes the diffusivity is

$$a = \frac{24.5 - 25.6}{2 \times 300} \Bigg/ \frac{26.0 - 2 \times 25.0 + 23.4}{0.02^2}$$

$$\underline{a = 1.22 \times 10^{-6}\ m^2/s}$$

The conductivity (k) is found using

$$k = a.\rho.C$$

where

ρ = Bulk density (1070 kg/m^3)

C = Specific heat (1.13 kJ/kg K)

$$k = 1.22 \times 10^{-6} \times 1070 \times 1.13$$

$$= 1.475 \times 10^{-3}\ kW/m^{o}C$$

$$\text{or}\quad \underline{1.475\ W/m^{o}C}$$

This procedure was repeated with the remaining readings in the column to give the values tabulated below.

		15 minute	
		Diffusivity (m^2/s)	Conductivity (W/m^oC)
Probe	2	1.41×10^{-6}	1.70
	3	1.22×10^{-6}	1.47
	4	5.0×10^{-7}	0.60
	5	8.8×10^{-7}	1.06

When the experiments using the 160 kg coal bed were carried out the probes were arranged in two groups of three. This meant that only a single diffusivity reading for each group could be evaluated. In this case the mean of the two groups was recorded at time 15 minutes.

9.7 RESULTS

9.7.1 Presentation

The raw data produced by 24 experiments with about 200 instrument readings and a similar amount of derived data would be too voluminous to comprehend easily. For this reason the most important aspects have been grouped and evaluated into a small number of representative coefficients and mean readings as described in sections 9.5 and 9.6.

The only parameter which was deliberately changed between one experiment and the next in each group was the air velocity. Thus all data is presented with respect to the mean air velocity over the conveyor.

9.7.2 Results

Figure 9.7a

Table of results for cooling experiments (160 kg coal bed)

Air Velocity m/s	Air mean heat gain kW	Coal mean heat loss kW	Heat balance %	Coal Peak heat loss kW	Convective heat transfer coeff. $W/m^2{}^oC$
0.505	0.460	0.270	59	0.455	15.5
0.750	0.400	0.295	74	0.665	15.5
0.875	0.435	0.300	69	0.830	25.0
1.120	0.450	0.315	70	0.710	34.0
1.570	0.420	0.305	62	0.910	26.0
1.695	0.430	0.305	71	1.100	44.5
1.980	0.510	0.310	61	0.735	34.5
2.215	0.515	0.325	63	1.085	31.5
2.400	0.585	0.315	54	0.925	39.5
2.500	0.605	0.345	57	1.080	50.0
3.450	0.645	0.315	49	1.215	60.5
4.965	0.635	0.350	55	1.210	64.0

Figure 9.7b

Table of results for cooling experiments (320 kg coal bed)

Air velocity m/s	Air mean heat gain kW	Coal mean heat loss kW	Heat balance %	Coal peak heat loss kW	Convective heat transfer coeff. $W/m^2 {}^oC$
0.305	0.645	0.400	62	0.410	14.5
0.495	0.525	0.415	79	0.555	19.0
0.885	0.845	0.500	59	0.695	20.5
0.965	0.735	0.525	71	0.745	23.0
1.325	0.720	0.520	72	0.960	30.5
1.650	0.855	0.505	59	0.950	35.5
1.975	0.870	0.530	61	1.265	40.5
2.485	0.845	0.515	61	1.160	45.0
2.~~2~~925	0.805	0.505	63	1.130	49.0
3.205	0.855	0.495	58	1.150	52.0
3.650	0.955	0.505	63	1.185	52.0
5.005	0.835	0.510	61	1.305	65.0

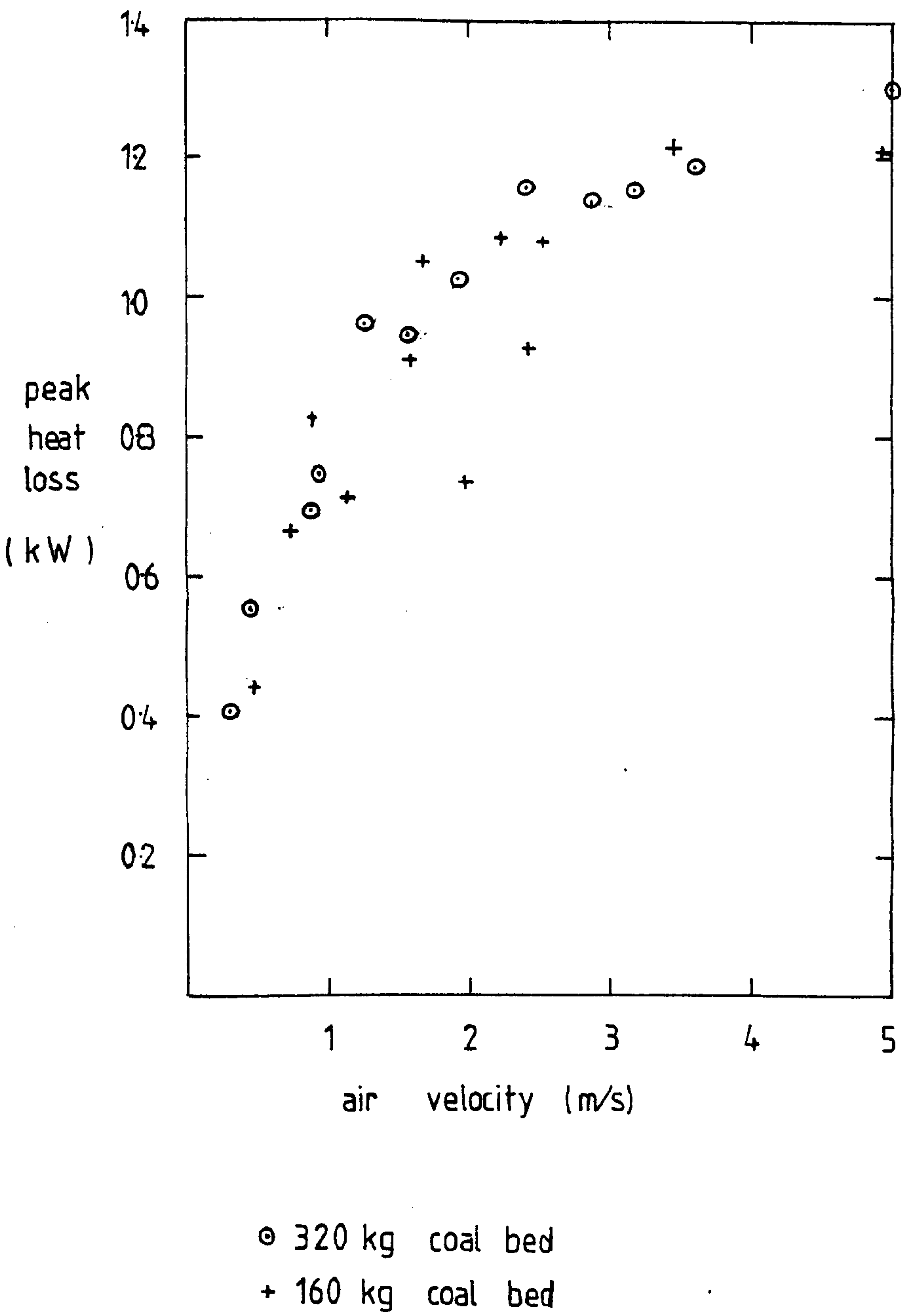


FIG 9·7c PEAK HEAT LOSS RATES AGAINST AIR VELOCITY

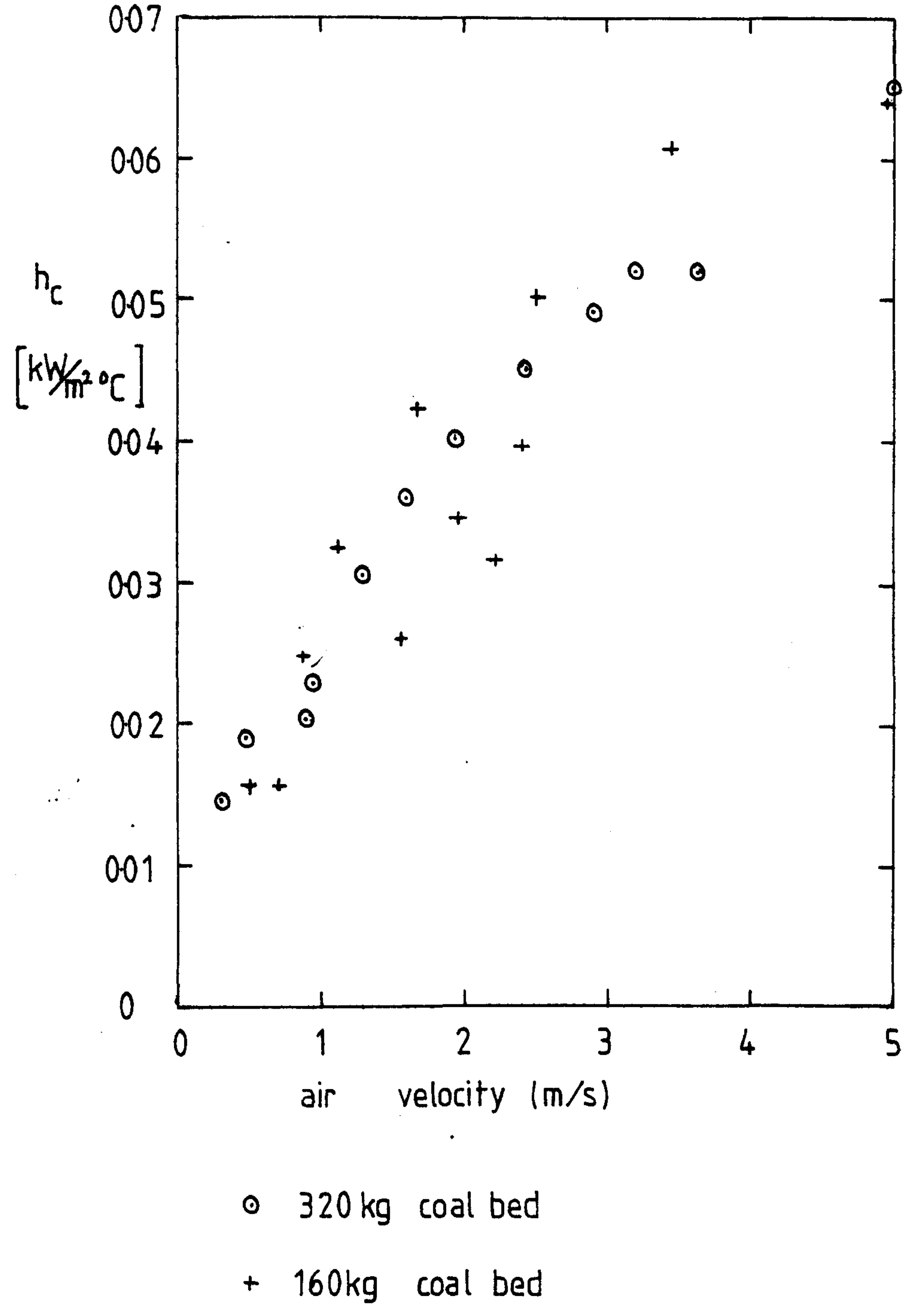


FIG 9·7d CONVECTIVE HEAT TRANSFER COEFFICIENTS AGAINST AIR VELOCITY

Figure 9.7e

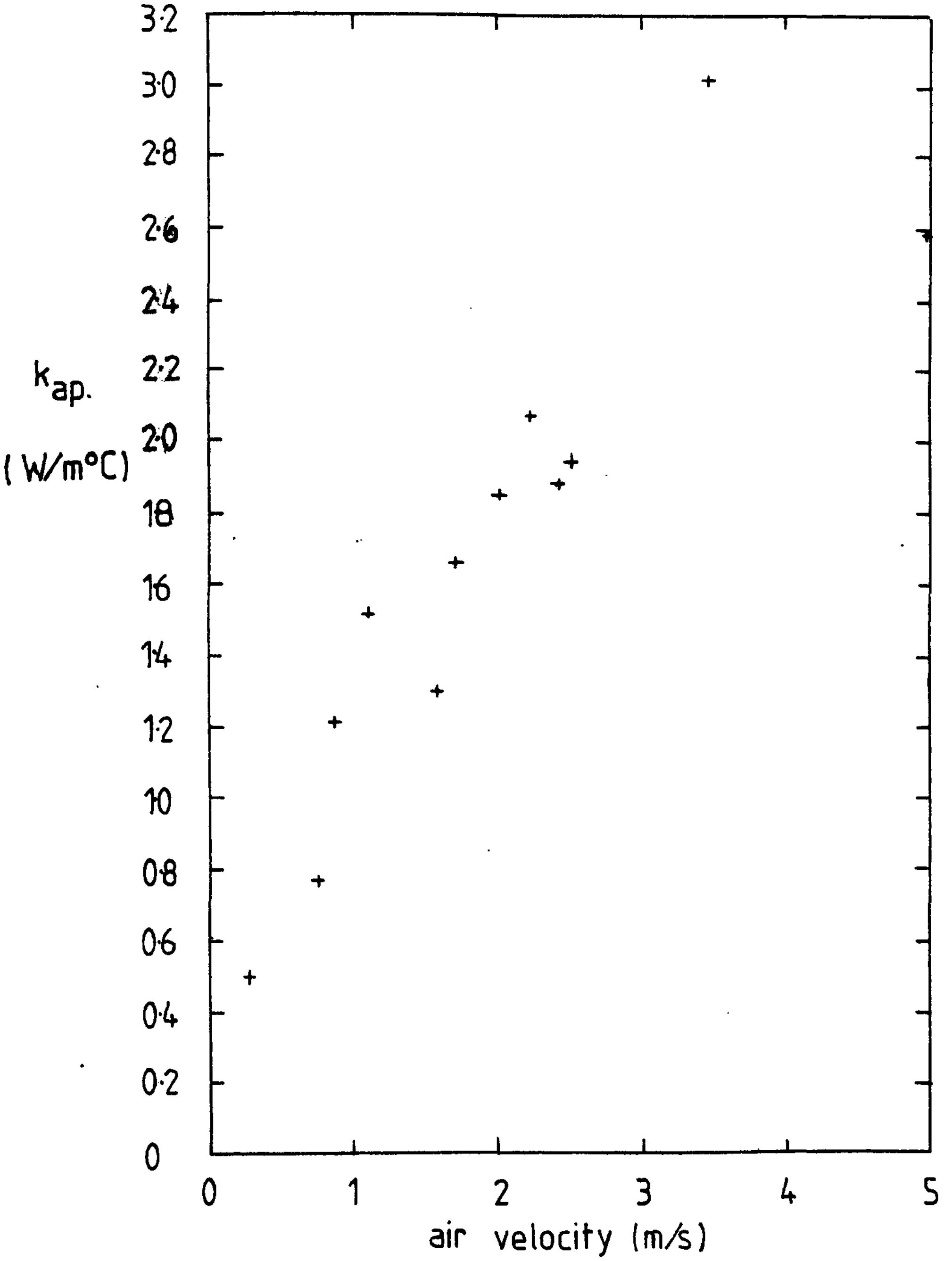
Table of apparent conductivity values (160 kg coal bed)

Air velocity m/s	Conductivity (apparent) W/m°C
0.505	0.49
0.750	0.77
0.875	1.21
1.120	1.51
1.570	1.30
1.695	1.66
1.980	1.83
2.215	2.07
2.400	1.88
2.500	1.94
3.450	3.07
4.965	2.59

Figure 9.7f

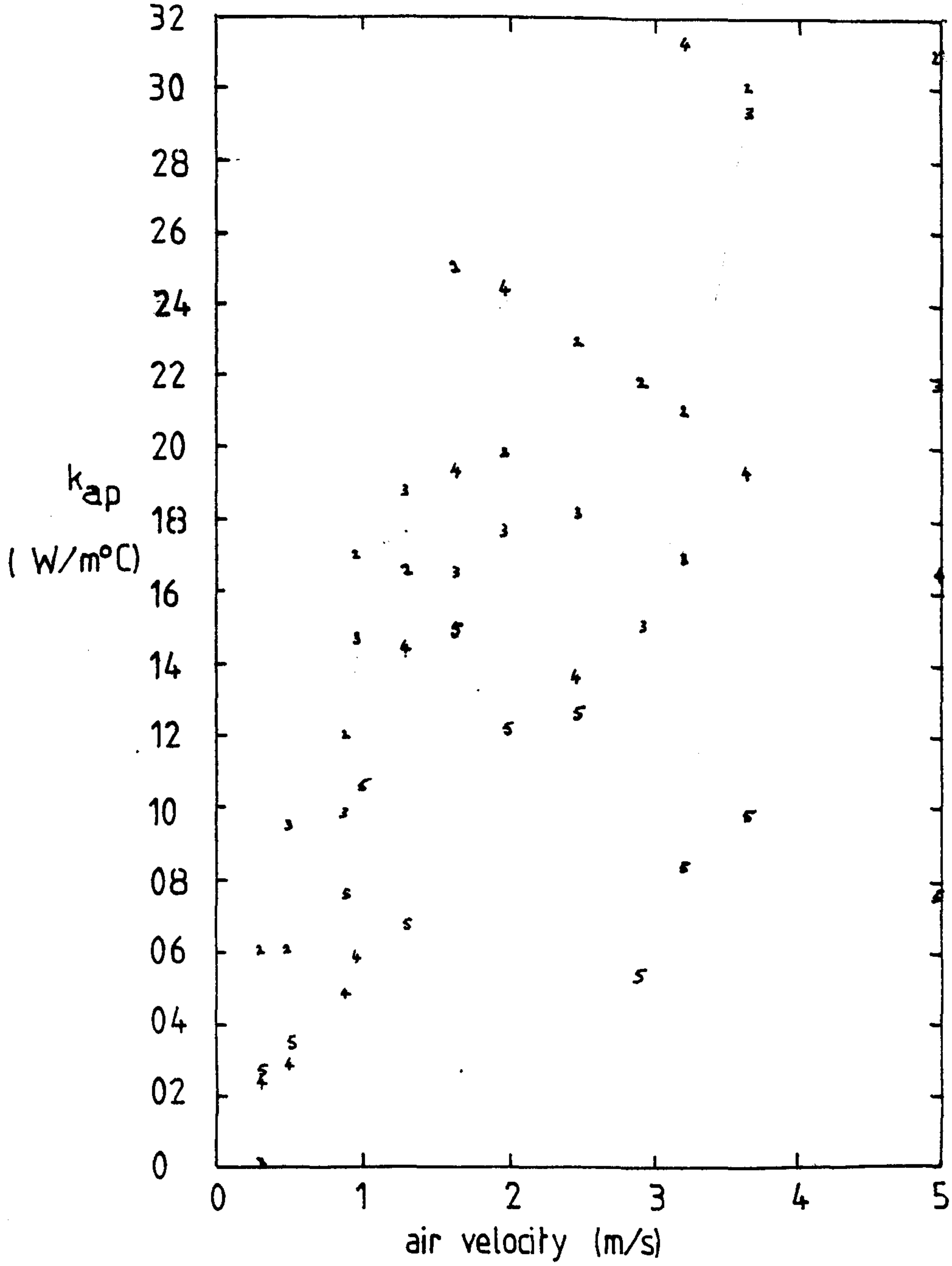
Table of apparent conductivity values (320 kg coal bed)

Air velocity m/s	Conductivity (apparent) W/m°C p2	p3	p4	p5
0.305	0.60	0.0	0.24	0.27
0.495	0.60	0.95	0.29	0.35
0.885	1.20	0.99	0.49	0.77
0.965	1.70	1.47	0.60	1.06
1.325	1.66	1.89	1.44	0.68
1.650	2.51	1.72	1.95	1.50
1.975	1.97	1.77	2.65	1.21
2.485	2.29	1.81	1.37	1.27
2.925	2.18	1.49	3.14	0.53
3.205	2.10	1.69	1.59	0.83
3.650	3.01	2.95	1.93	0.98
5.005	3.10	2.18	1.66	0.77



160kg coal bed

FIG 9·7g APPARENT THERMAL CONDUCTIVITY AGAINST AIR VELOCITY



320 kg coal bed

2, 3, 4, 5 = probes at 20, 40, 60 + 80mm

FIG 9·7h APPARENT THERMAL CONDUCTIVITY AGAINST AIR VELOCITY

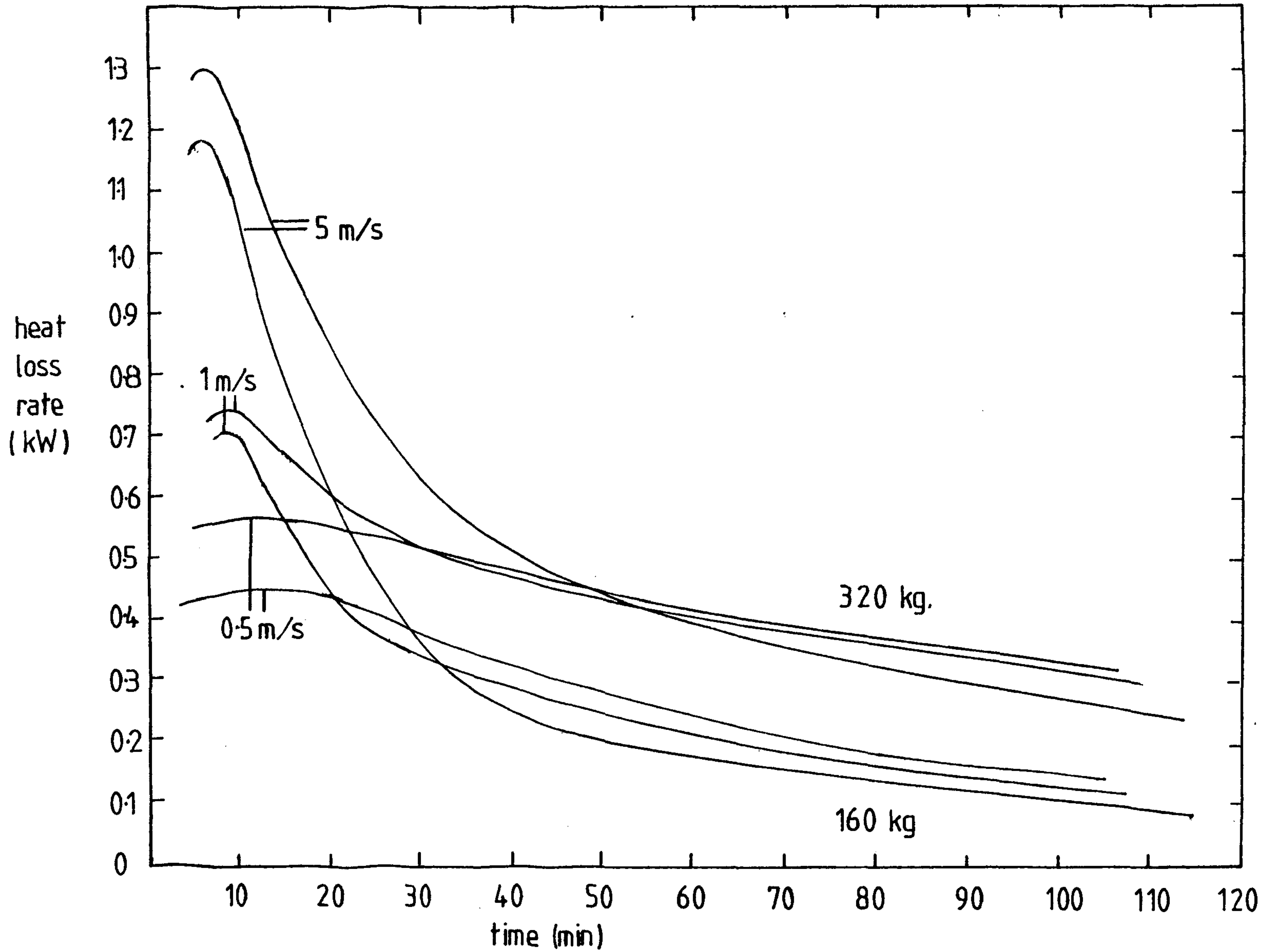


FIG 97i EXAMPLES OF COAL HEAT LOSS RATES AT DIFFERENT AIR VELOCITIES

9.8 DISCUSSION

9.8.1 General comments

Whilst this series of experiments did not provide all the information to the expected accuracy it did provide some valuable quantitative information. The basic concept of the insulated duct and the pre-heated coal is still considered sound. The shortfalls in performance and experimental accuracy were due mainly to the limited resolution of the instruments, particularly those measuring air wet bulb and coal surface temperatures. Difficulties also arose when attempting to produce the required step temperature change to the inlet temperature at time zero. Underground tests showed a large proportion of the heat from conveyed coal was due to latent heat transfer and this is recognised as a major failing of this work. This aspect is discussed later in the suggestions for further work.

9.8.2 Reynolds number and air velocity

The results for the tests are all expressed against the average air velocity in the path through the duct over the coal. In some cases, for instance the convective heat transfer coefficients, the results should possibly have been displayed against the dimensionless characteristic Reynolds number, but to allow comparison of results from one set of results to another results were expressed against the velocity.

Reynolds number may be derived in different ways representing various situations and giving different answers. In the case of these experiments it could have been based on conditions of the rough coal surface with the characteristic length some function of surface roughness and air velocity at the actual surface. A Reynolds number could also be based on the whole air path through the duct. In this case it would vary linearly with air velocity. The Reynolds number derived in this way based on the air path over the coal taking the mean diameter (d) as 0.675 m, air viscosity (μ) as 1.7×10^{-5} Ns/m^2 and air density (ρ) as 1.2 kg/m^3 would be at velocities 0.5 and 5.0 m/s be 2.25×10^4 and 2.25×10^5.

Turbulent flow is characterised by a Reynolds number of above 2×10^3, therefore it may be taken that the air in the duct was fully turbulent throughout the range of velocities tested even the first run in the 320 kg experiment at 0.3 m/s. This has to be verified since if the air flow was laminar, layering could take place and a single temperature reading (say) would not be representative of the whole cross section.

9.8.3 Heat transfer through the conveyor belt

The heat transferred by conduction through the conveyor belt material and by convection to the air proved to be too small to measure reliably. It can be seen from the distribution of probe temperatures within the coal that most of the heat transfer within the coal was upward. A probe taped to the bottom surface of the conveyor and checks with the infrared thermometer showed that the lower surface of the belt was close to the air dry bulb temperature confirming the low heat transfer rate.

It may be shown that the main barrier to heat transfer is the coal air mixture itself rather than the belt.

The thermal conductivity of conveyor belting is about 0.15 W/m^{o}C, Nylon and PVC the main constituents both having ranges of 0.12 - 0.17 W/m^{o}C. The temperature difference between the coal and air (dry bulb) was about 15^{o}C average. The belt was 8 mm thick and the area involved 6 x 0.5 = 3 m^2. Inserting these values into the Fourier equation of conduction we have an estimate of the potential heat transfer rate

$$Q = -k\ A\,\frac{dt}{dx}$$

$$= -0.15 \times 3 \times \frac{15}{0.008}$$

$$= 0.843\ \text{kW}$$

The actual figure was much less than this due to the thin layer of coal immediately above the belt having its own self insulating effect due to the very low thermal conductivity and specific heat of granular coal. The heat transfer rate predicted would only exist for a short period until the thin layer cooled slightly, reducing the temperature difference across the belt material and hence the 'driving potential' for heat conduction.

Having recognised that the proportion of heat transferred through the belt was small it was decided not to attempt further measurement of this, but to concentrate on the evaluation of heat flow from the exposed coal surface.

9.8.4 Air heat pick up and wet bulb temperature

The heat pick up was evaluated by measuring the air quantity and the wet bulb temperature, which specifies the heat in the air, at the inlet and exit to the control volume. The resolution of the wet bulb temperature measuring system proved to be insufficient. For example, at the higher air velocities, near 5 m/s, the mass flow of air was 2.1 kg/s. The sigma heat content of air at the temperatures reached near the end of each run ($\sim 5^{\circ}$C) varied by about 2 kJ/kg per $^{\circ}$C wet bulb. Therefore a 0.1°C change in wet bulb temperature between inlet and outlet represents a heat gain of 0.42 kW. Thus at the higher air velocities and near the end of the cooling run the heat pick up and hence wet bulb temperature change was smaller than could be detected accurately by the measuring system (see figure 9.7i). Although the temperature probes were considered to be of acceptable accuracy the meter on which the temperature was displayed could only be read to 0.1°C. The heat pick up and temperature increases in the early stages of the experiments were large enough to measure with some confidence of a lower percentage error. For this reason the overall mean air heat pick up was used to compare with the more accurately assessed coal heat loss.

A temperature measuring system accurate to 0.01°C would be needed for acceptable accuracy. The author has already constructed and used such a system based on an eight junction thermocouple but this was used in a steady state situation where the time taken to use the Pye potentiometer (see Chapter 8.2) was no problem. The use of a sensitive multi-channel chart recorder and temperature instrumentation by thermocouples is suggested if such experiments are carried out in the future.

The heat balance figures of between 49 and 79% are thought to be due mainly to errors in the air heat pick up measurement rather than the heat loss rate of the coal. Heat balances of 100% rarely occur and in most industrial tests on heat exchangers 50% balances are considered acceptable*.

9.8.5 Heat lost by coal

The heat lost by the coal could be measured more accurately than the heat given to the air. Despite using equipment of the same measuring resolution the treatment of the observed values was different. An error in temperature measurement of a probe at a given time would, due to the heat loss being evaluated by temperature difference, result in two compensating values of heat loss over two time intervals. Errors could be caused by placement of probes inaccurately at the fixed depth in the coal bed but the probes were installed before the experiments and time could be taken to ensure accurate positioning. A slight error would be caused by the top probe being 1 mm below the surface and the bottom probe being 1 mm above the belt resulting in a 19 mm rather than 20 mm distance to the next probe. This would result in a smaller error though than putting the probe on the surface where a large air temperature contribution would be made. It is likely that with the probe on the surface the temperature value would be very close to the air dry bulb temperature.

Some typical cooling curves for different coal loads and air velocities are shown in figure 9.7i. Noticeable features are the peaks which occur in every case at between 5 to 10 minutes. If a

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* Dr. M.J. McPherson, Private Communication

step fall in temperature could have been arranged this peak would not have occurred but the temperature would have fallen sharply at time zero. The peaks in the heat gain occurring at 1 and 5 m/s are not much smaller for the 160 kg coal bed than the 320 kg coal bed. This is to be expected since the surface areas and the starting temperatures of the coal were the same. Therefore air travelling over the surface would initially gain the same amount of heat. The difference between the thermal capacities of the two coal beds becomes more apparent with increasing time.

As expected the higher air velocity results in an increased cooling rate earlier in the experiment and a reduced cooling rate later, the thermal capacity of the coal being the same regardless of air velocity. Eventually coal in a cool airstream will reach the temperature of the airstream regardless of the velocity.

The peak heat loss rate and the heat loss rate for the first 10 minutes or so are important parameters. These experiments were continued over a long period to derive certain information, but coal on a conveyor underground is rarely on a single conveyor for more than ten minutes before it is transferred to another conveyor and the cooling process recommences. The amount of coal on a conveyor will affect the peak heat loss rate in the underground situation since a heavier coal load will result in a higher average coal temperature after mixing at a transfer point.

The peak heat transfer rate is displayed graphically in figure 9.7c and a trend is apparent. Several efforts to fit a line through the points were made and the following relationship was derived between peak heat loss rate and air velocity (U)

$$Q_{max} = 0.7_3\sqrt{U}$$

9.8.6 Convective heat transfer coefficient

The derivation of a convective heat transfer coefficient was particularly successful. Latent heat transfer did not take place due to the coal being dry and radiative heat exchange with the duct wall would be negligible due to the coal surface and duct walls both being close to dry bulb temperature. Thus heat transferred may be attributed wholly to convection. The heat transfer coefficients so derived would be as accurate as the heat loss from the coal on which they were derived and the spread of the results, as shown graphically in figure 9.7d, seems to confirm this. Referring back to section 9.6 shows how the coefficients were derived and the table 9.6c represents a typical example of the spread of results for heat transfer coefficient for a single experiment. A trend is apparent on figure 9.7d which allows the following relationship between convective heat transfer coefficient and air velocity (U) to be expressed

$$h_c = 0.25\ U^{0.7}$$

9.8.7 Thermal diffusivity and conductivity

The thermal diffusivity was derived using a similar process to that used in Chapter 8.4. These experiments took place in slightly less suitable conditions with time intervals between readings being subject to slight variation to which the transient method of assessing diffusivity is sensitive. Consequently results were not so accurate. The representative results from each run display two trends. Firstly, as the velocity increases the apparent conductivity increases. Secondly, as shown on figure 9.7b the apparent conductivity falls as depth into the coal bed increases. These two effects are undoubtedly due to the limited penetration of air into the coal bed which transfers heat as it moves. As the velocity increases so will the amount of heat transferred resulting in a higher apparent conductivity. The values obtained are mostly much higher than those obtained in the experiments designed specifically for the purpose (Chapter 8). Only the probes well below the coal surface have a low conductivity comparable to the previously derived values. This shows that results derived in specific situations must be used extremely cautiously outside their original context.

9.8.8 Surface temperature measurement

The measurement of the actual surface temperature of the coal using the 'Infratrace' showed some interesting results. It was expected that the convective heat transfer coefficient would need to be based on this temperature rather than that measured slightly below the surface. Figure 9.5b shows the variation of surface temperature with time. It can be seen that the surface temperature quickly fell toward the air dry bulb temperature. At higher air velocities the fall was steeper, at lower velocities slightly more shallow. The infra-red measuring technique gives the temperature of the target surface regardless of how thin it may be. A method using a probe or thermometer measures some contribution by the air itself in the coal. These two implications would suggest that the infrared measured temperature would, subject to instrumental accuracy, be more reliable. Nevertheless no relationships between the surface temperature and heat transfer could be quantified.

A likely explanation for the closeness of the surface temperature curve to the air dry bulb temperature in that the very low conductivity of the coal allowed the formation of a thin sub skin of coal at air temperature rather than that of the coal below. The heat transfer coefficients derived from the probe temperature seem to bear this out.

The problems encountered here are typical of the problems of assigning representative values to the properties of two phase media.

9.8.9 Tests using wet coal

Attempts to evaluate latent heat transfer from wet coal in this system were totally unsuccessful. The basic problem being to provide a bed of coal with known and uniform moisture content. If the air could be humidified to a well controlled degree in the inlet a similar concept to that used to provide coal at uniform temperature could be used. On the scale of this experimental duct with the air volumes involved such humidification would be impractical at reasonable expense.

Methods which were tried were quickly removing the top sections of the duct and spraying the coal before replacing the sections and starting to monitor the cooling but the time involved resulted in a large amount of heat being lost before monitoring could begin. Spraying water into the coal with access through the hatches was tried but in the confined space it was difficult to distribute the water evenly. To evaluate latent heat transfer a completely different and much smaller system was needed.

CHAPTER 10

TESTS ON A CONVEYOR AT PYE HILL COLLIERY

CHAPTER 10

TESTS ON A CONVEYOR AT PYE HILL COLLIERY

10.1 SELECTION OF SITE AND INVESTIGATIONS

10.1.1 Choice of site

Due to the difficulties encountered in evaluating the heat contribution of conveyed coal in the laboratory a search was made for a suitable underground site for an in situ investigation. It is accepted that underground climatic conditions are subject to large variation and interactions, as described in Chapter 6, and not ideal for controlled evaluation of isolated heat sources. However careful choice of an underground site of manageable size and complexity with the minimum of unwanted interacting heat sources would enable meaningful detailed measurements. Until a fairly detailed investigation is actually carried out the researcher does not know how suitable an apparently promising site is, but certain characteristics may give some prior indication.

For the purposes of this study the ideal site would be a conveyor road with no junctions or nearby roadways along its length to cause air losses or mixing. The air inlet conditions should be as stable as possible. The best site should have no machinery installed apart from the conveyor which would hopefully run at a steady rate with a uniform load. This important condition means that generally such sites would be fed by a bunker system meaning it would be well outbye.

Discussions with the Area Ventilation Engineer NCB South Nottinghamshire Area produced a possible site at Pye Hill No. 2 Colliery which was then visited and found suitable.

10.1.2 Description of site

The site chosen at Pye Hill No. 2 Colliery was in a length of conveyor road known as '5's'. The intake roadway ran from the bottom of the main conveyor drift to the surface,a distance of 1040 m making a right angle junction with another airway containing the main bunkers. These bunkers handled most of the coal produced from the Blackshale seam by both Pye Hill Nos. 1 and 2 Collieries and consequently a fairly steady stream of coal from the bunkers along '5's' and up the surface drift could be relied upon. The fluctuations in production by the faces supplying the coal being well damped by the bunker. Fortunately the discharge rate from the bunkers was monitored and phoned to the surface control room by the bunker operator so coal flow rates were readily available.

An airway known as the 'old loader' ran from the pit bottom to a junction with '5's' conveyor road at a distance of 25 m from the drift bottom. The air quantity supplied along this road was of the same order as that supplied down the surface drift to '5's' and mixing of the two airstreams took place at the junction. Consequently the test length in '5's' was arranged to start a suitable distance downstream (20 m) for full mixing of the two airstreams to take place before entering the test length of roadway. No other airways were nearby to affect the airflow by leakage. Being well outbye and not subject to fluctuations caused by opening and closing of air doors the airflow was volumetrically stable. Naturally the temperature was expected to vary in accordance with surface conditions.

The roadway depths of the test length were 187 m at the conveyor discharge end and 259 m at the bottom of '5s' where the coal was loaded. The mean VRT taken from a local borehole profile was 16°C. The roadway was in good condition and appeared to be dry and this was confirmed later. The coal was conveyed along '5s' by two consecutive conveyors of equal power (3 x 90 kW motors each) and were of approximately equal length and speed. The motor-drive units and supply transformers for both conveyors fell within the test length. The belt was 0.9 m wide and rollers were spaced at 1.4 m intervals along the top run of belt and 4.2 m intervals on the return run. Figure 10.4a shows a schematic plan of the district.

10.1.3 Choice of survey techniques

The survey techniques used to investigate a particular site should be selected carefully to gather the maximum of useful information and minimise errors. Preliminary visits to the test site resulted in a strategy being devised to evaluate the various heat sources. The heat sources in '5s' conveyor road could be divided into three main categories.

(i) Strata heat which was expected to be comparatively low due to the low VRT at the shallow depth.

(ii) Machine heat from the 2 conveyors and their associated electrical supplies.

(iii) Heat from the conveyed coal.

It was decided to evaluate each aspect using subsidiary specialised surveys. Then combine the knowledge gained to attempt one fast concentrated survey to assess any interactions.

Consequently the following surveys and investigations were called for.

(i) A one week continuous temperature record at some point on the test length.

(ii) An airflow survey to evaluate the air flow rates in the test length and adjoining airways.

(iii) Several temperature traverses at various times to include peak production and base load conditions to evaluate heat and moisture gains by the air.

(iv) A survey of power supplies, including cable routes connections, transformers, and meters etc which would allow power measurement to the motors. Also any motors and machines using electrical power would be included.

(v) A coal temperature survey using both sampling and infrared measurements. Samples would be taken to measure moisture content changes along the conveyor.

10.2 CONTINUOUS TEMPERATURE RECORD

10.2.1 Purpose or record

A continuous temperature record for a typical working week was needed to find the expected temperatures at certain times of the week and also gain some prior knowledge of the scale and rate of change of temperature fluctuations. This would allow the times of surveys to be chosen to investigate certain aspects and also avoid making a survey at an unsuitable time for instance whilst the temperature was changing rapidly. Also if any unexpected results were obtained it was hoped a continuous record might provide some helpful clues to the likely cause.

10.2.2 Equipment and installation points

The continuous temperature record was made using two thermo-hygrographs. A detailed description of the mode of operation of the instruments and the temperature record produced appears in Chapter 6.3. The instruments were sited at the bottom of the surface drift and at the inbye end of the test length at the bottom of '5's'. The instruments were installed on a Tuesday and ran for one week. Figure 10.7a shows the records produced.

10.3 AIRFLOW MEASUREMENTS

The air volume flow was measured at the bottom of the surface drift, the 'old loader' roadway and '5's' conveyor road near stations 1, 2 and 3 in figure 10.4a at designated measuring points of known cross sectional area. The standard measuring procedure described in reference [19] was used. The instrument used was a vane anemometer. Airflow measurements were usually made before and after each temperature survey and the volume flow was always found to be stable during a survey. The volume flow rate was converted to a mass flow rate by multiplying by the density of the air calculated from the temperatures and pressure at the nearby measuring station. The results of the airflow surveys are combined with those of the temperature survey to which they relate.

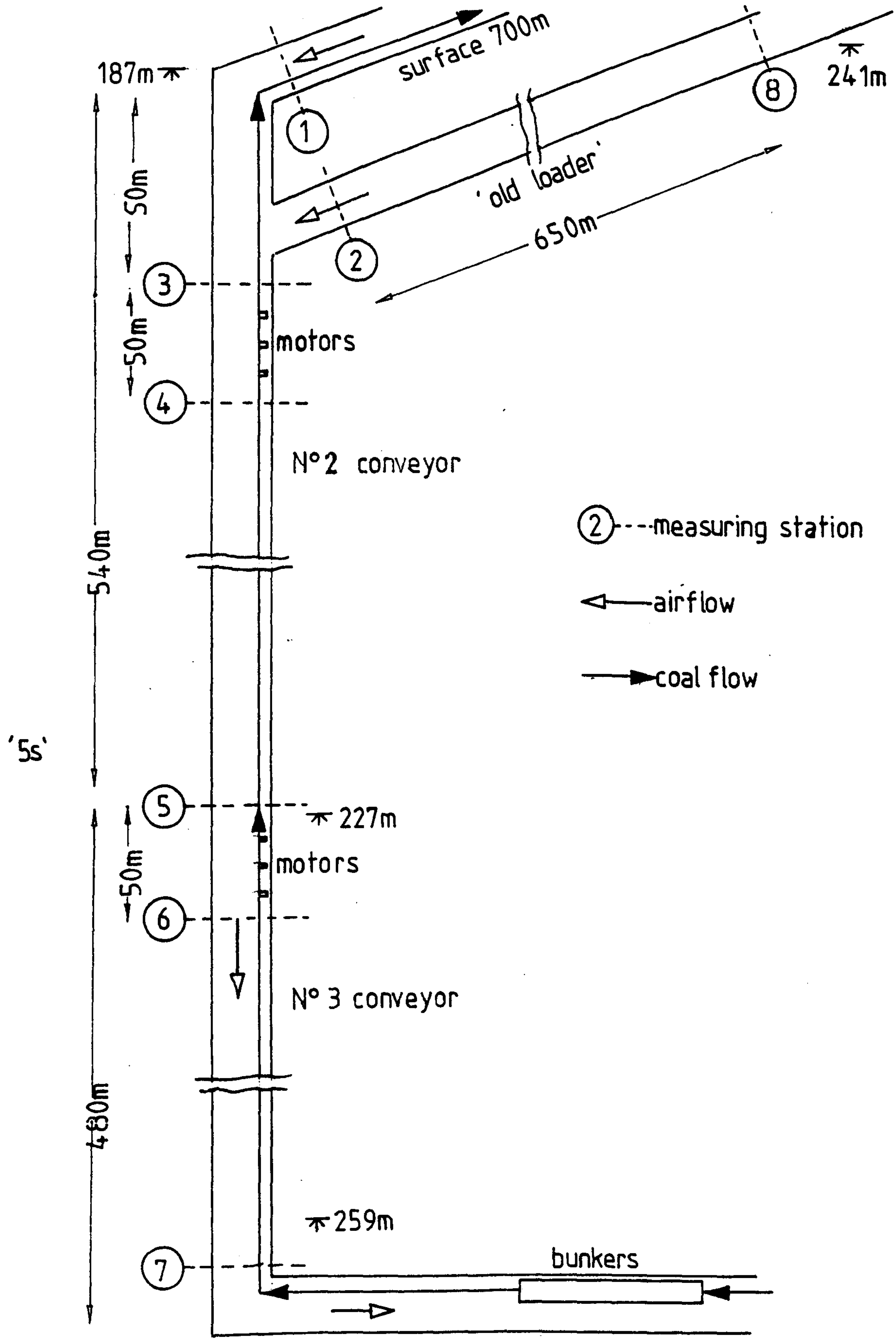


FIG 10.4a 5's CONVEYOR ROAD TEST SITE
PYE HILL N°2 COLLIERY

10.4 TEMPERATURE SURVEYS

10.4.1 Position of measuring stations

The conduction of a temperature traverse has already been described in detail in Chapter 6. Once again in these surveys the measuring stations were chosen beforehand to include a particular length of roadway or concentration of machines. The position of the stations is shown in figure 10.4a.

The last length, was divided into 4 sections by stations 3 to 7. Two short (50 m) sections 3-4 and 5-6 contained the conveyor motor-drive units and the associated electrical sub-stations. These sections were deliberately chosen to be as short as possible so that the strata heat contribution within the length would be minimal. Thus all the heat in the section could be attributed with little error to the machinery or coal on the conveyor.

Two long sections 4-5 and 6-7 were lengths of airway containing only conveyor. No electrical energy was present in these sections and so any heat gains would be due to the strata, the conveyor itself, or its coal load. It should be stressed that the heat produced by the conveyor as friction is supplied originally as electrical power to motors in the other sections. Therefore careful accounting is needed when carrying out heat balances.

The 'old loader' roadway from station 2 to 8 was included in some of the surveys as it was of similar depth and gradient as the test length in '5's' conveyor road. Station 8 being 650 m from station 2 at a depth of 241 m. The conditions in the 'old loader' being similar to '5's' apart from the installed conveyor and airflow would allow an estimate of strata heat flow into the test length if needed.

10.4.2 Temperature survey in static conditions

A temperature traverse along the test length was made in the static non-production period of a Sunday. The survey was carried out between 7 and 9 a.m. to coincide with the deputy's inspection of the district and cause minimal disruption of colliery routine. The previous night was cold and surface temperature was $1^{o}C$ whilst the survey was in progress. Temperatures were measured using an 'Assman' hygrometer at stations 3 to 7. The purpose of this survey was to evaluate the heat and moisture pick-up by the air from the strata. The heat gain would be at a maximum whilst the machinery was not working and the inlet air to the test length was at a low temperature. The moisture pick up by the air from the strata is affected less by interactions with other sources whilst the air is well below saturation as was the case at this site. Consequently the water flow from the strata to the air would be more independent of the climatic variations in the airway. This point is treated in more detail in the discussion.

10.4.3 Temperature survey in production conditions

Several temperature traverses were made in production periods. Results obtained were of the same order but varied slightly with varying production and inlet conditions. The survey recorded here was the last one made. All temperature measuring stations were used and an airflow and power consumption survey was combined with it. The stations in the test length, 3 to 7, were traversed as quickly as possible to obtain a full set of readings over as near as possible the same inlet and production conditions.

The temperature survey was carried out using an 'Assman' hygrometer and the techniques used were those described in Chapter 6.2. The pressure at each station was measured using a precision aneroid barometer.

10.5 ELECTRICAL POWER MEASUREMENT

Before any temperature surveys were made a survey of the electrical machinery and power circuits was conducted. The only electrical machinery in the roadway designated as the test length was associated with the conveyor. The electrical supply to the two conveyors (number 2 trunk and number 3 trunk) was taken from the main 3.3 kW ring circuit at a sub-section adjacent to the motors as shown in figure 10.5a. The 3.3 kW ring circuit ran down the drift from the surface sub station round '5's' conveyor road and the bunker site and back to surface. The current could flow either way through the circuit but the direction of current flow could be ascertained from the meters in the sub stations. All sections of the supply had ammeters and the motor supplies had both ammeters and voltmeters. This provided a good estimate of the power consumption of the numbers. It could not be measured exactly due to no power factor meters being installed. The power drawn by a 3 phase motor is given by

$$Q = \sqrt{3} \times V \times I \times \cos \phi \qquad (1)$$

where

Q = Power drawn (W)

V = Voltage (V)

I = Current (A)

$\cos \phi$ = Power factor

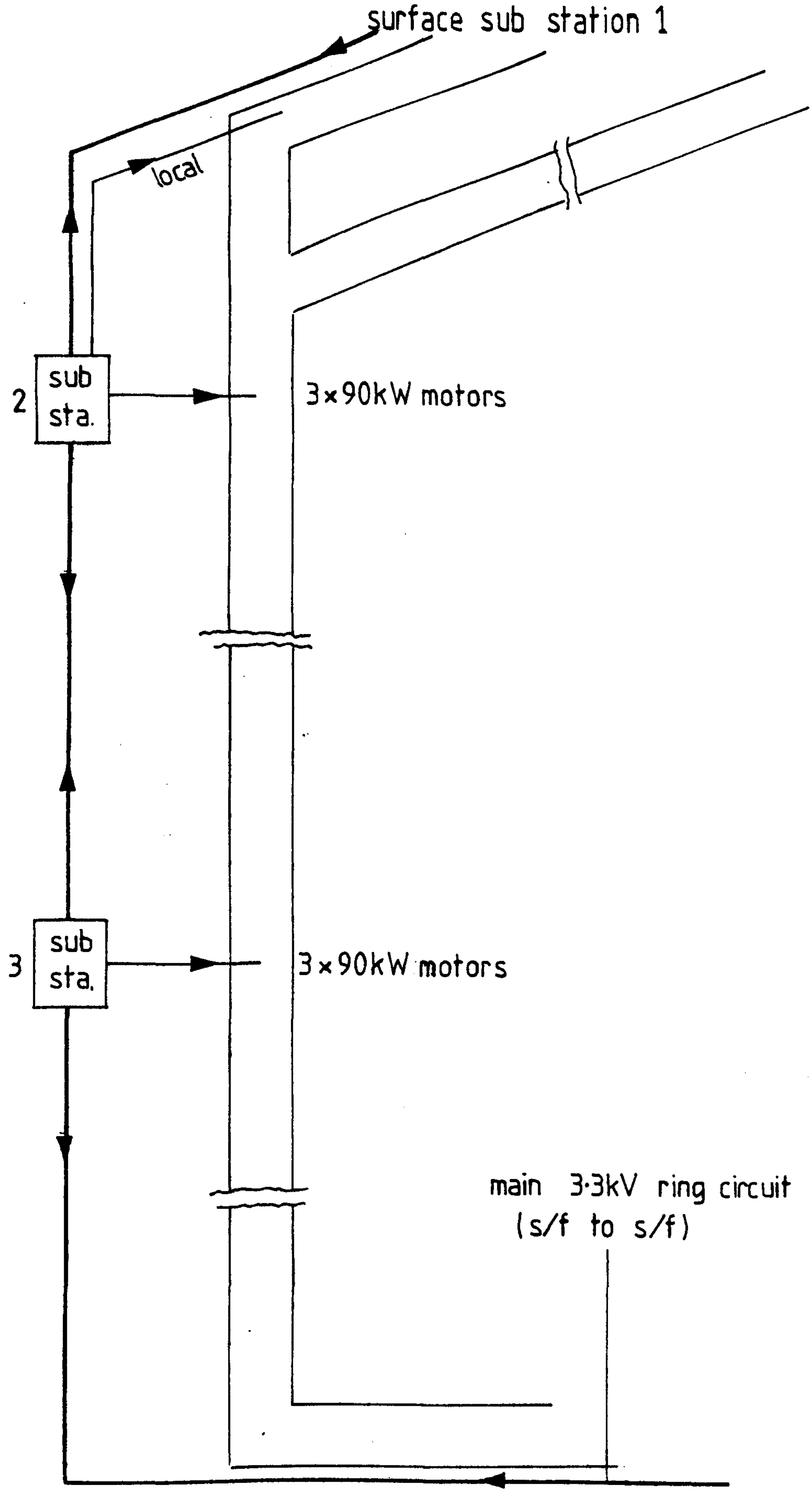


FIG 10·5a ELECTRICAL SUPPLY TO 5's CONVEYORS

The power factor varies according to the load on the motor. Discussions with the Colliery Electrical Engineer who had tested the motors suggested the following estimates be made:-

$$\cos\phi = 0.6 \text{ running light}$$
$$= 0.7 \text{ average load}$$
$$= 0.8 \text{ full rated load}$$

The circuit section marked 'local' on figure 10.5a was the supply for the lighting and pump circuits at the drift bottom. None of this power, apart from the negligible amount used for a few fluorescent lights, was expended in the test length.

The results of the power usage measurements taken simultaneously with the main temperature survey are shown diagramatically in figure 10.7d. The distribution of the motor power drawn may be apportioned as follows:

(1) Motor drive losses

This is usually about 20% of power drawn and the heat appears in the vicinity of the motors [9].

(2) Work done against gravity

This is useful work used to raise the potential energy of the coal moved. Calculated by the product of mass flow rate, change in elevation and gravity. No heat is produced.

(3) Friction losses

This is the power used along the conveyor to overcome roller friction. This appears as heat spread over the working length of the conveyor the power being transmitted from the motors through the belt. This is found by subtracting the power used in the motor drive unit and against gravity from the total power consumed.

10.6 COAL TEMPERATURE AND MOISTURE CONTENT MEASUREMENTS

10.6.1 Bulk temperature measurement

Measurements of coal temperature by the method described in Chapter 6.4 using the insulated container were made at the three transfer points, where No. 3 conveyor was loaded, No.3 to No. 2, and No. 2 to the drift belt. These points corresponded closely to stations 7, 5 and 3 respectively. The measurements were made during a production shift with five samples taken at each point going inbye and five samples at each point going outbye. This was to cancel the effect of any temperature drift during the production period. The measurements at each station varied by less than 2^{o}C. The mean value was recorded so once again the heat lost by the coal could be calculated from the product of the temperature fall and specific heat.

10.6.2 Surface temperature measurement

Surface temperature measurements were taken using the infra-red non contact thermometer described in Chapter 9 and Appendix 4. For use underground the instrument was modified to make it intrinsically safe and a letter of no objection allowing its use on this specific study obtained. The 'Infratrace', having the ability to give an instant temperature reading even of a moving target,was used to take the temperature of the coal on the conveyor or any other surface of interest. Although only able to give an absolute reading to $1^{o}C$ accuracy it allowed the results of the bulk coal temperature survey (previous section) taken on one day to be used in conjunction with the results of the main air temperature survey taken on a different date. This was facilitated by verifying that the coal was at approximately the same temperature at either end of the test length on both days.

10.6.3 Moisture content measurements

In conjunction with the bulk coal temperature survey (10.6.3) samples of coal were taken to evaluate the moisture content of the coal at either end of the two conveyors under observation with a view to attempting a moisture balance between the air and the coal. Six samples of coal of approximately 2 kg each were taken in sealed plastic bags, one with each batch of temperature measurements. The samples were taken back to the laboratory and were weighed, air dried at $35^{o}C$ and weighed again to establish the moisture content by difference. These samples were also used to find the specific heat of the coal by calorimetric method of mixtures. This was used in conjunction with the bulk coal temperature measurements to find the heat lost by the coal on the conveyor.

10.7 RESULTS

10.7.1 Presentation

As described in the previous section several surveys were made to evaluate different aspects of the airway and conveyor system. The object of these studies was to quantify the heat from all sources in the roadway in order to include them in an overall energy balance and relate this to the contribution from conveyed coal. As a result the surveys described were all components designed to fit together to form an integrated picture. The main framework of this integrated picture was the final air temperature and power survey conducted in production conditions. The previous air temperature, coal temperature and power surveys were used to gain an understanding of the variations and interactions taking place in the test area. The results given here are for each of the surveys described in the previous sections 10.3 to 10.6. They are all referred to the final definitive air temperature survey and are presented in an order which allows logical arrangement and construction of the overall picture rather than strictly in the order described.

10.7.2 Processing of results

The results of the temperature survey were calculated from the observed values using the methods described in Chapter 6.2. The electrical power distribution calculations are shown in full from observed data to calculated values. The same applies to the coal bulk temperature measurements and heat gain.

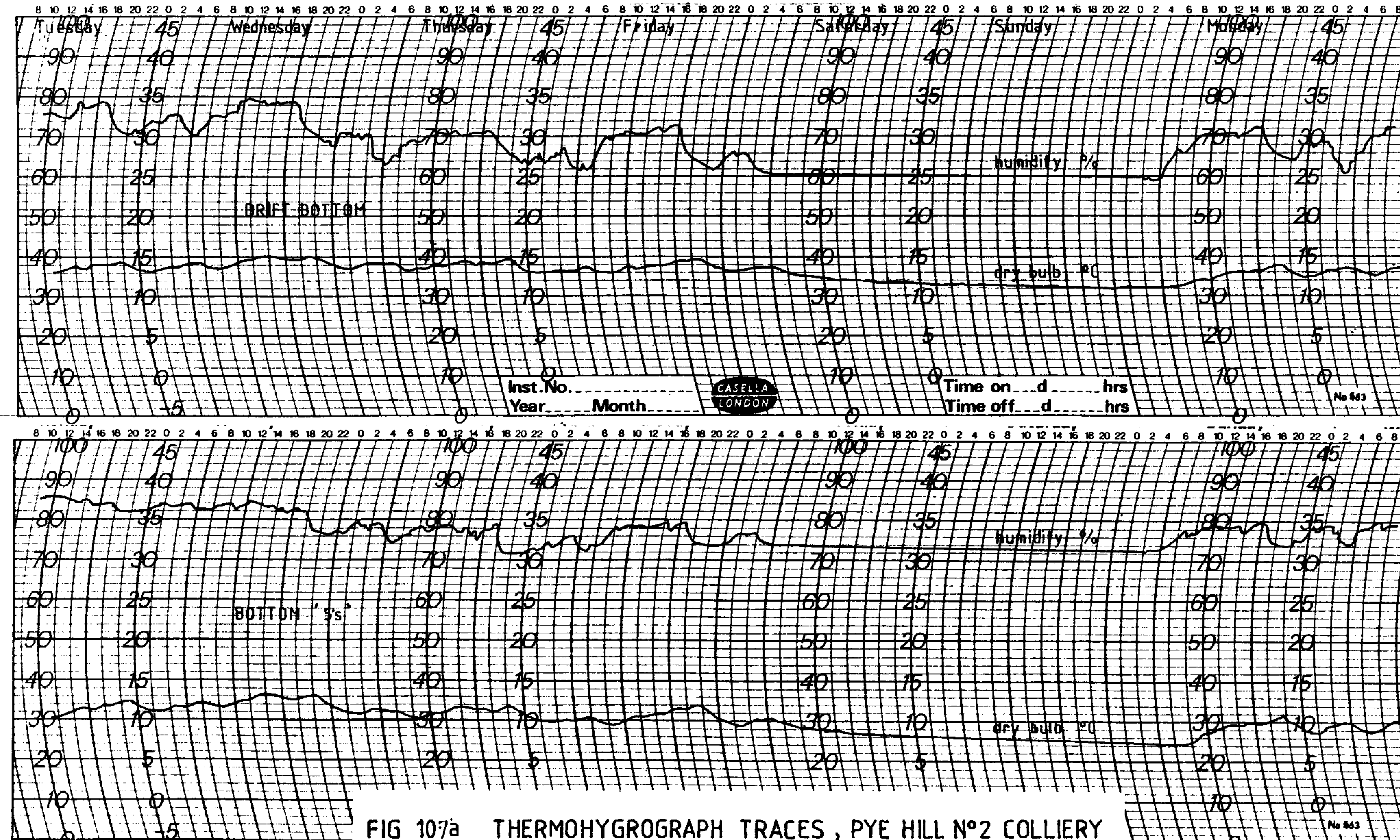


FIG 107a THERMOHYGROGRAPH TRACES, PYE HILL N°2 COLLIERY

Figure 10.7b

Table of results for airflow - temperature survey in non production conditions

Station	Wet bulb temperature °C	Dry bulb temperature °C	Sigma heat kJ/kg	Heat gain kW	Moisture content g/kg	Moisture gain g/s
3	4.6	6.4	17.48		4.43	
				15.20		1.20
4	4.8	6.7	17.86		4.46	
				109.20		6.80
5	6.2	9.0	20.59	(-15.70)	4.63	
				-		-
6	6.2	9.0	20.59		4.63	
				58.8		2.00
7	7.0	10.5	22.06	(-12.56)	4.68	
			TOTAL	183.2	TOTAL	10.00

Air mass flow rate 40.0 kg/s

The figures shown in brackets are the amounts of heat gained by the air due to autocompression as it moves to greater depth. They are calculated from the mass flow rate, change in level and gravity,

eg $$40.0 \times 32 \times 9.81 = 12556$$

$$kg/s \times m \times m/s^2 = \frac{Nm}{s} = W$$

$$(1\ kg = 1\ Ns^2/m)$$

The autocompression (adiabatic compression) is evaluated since it is a heat rise which occurs without an external heat source.

Figure 10.7c

Table of results for airflow temperature survey

in production conditions

Station	Wet bulb temperature ^{o}C	Dry bulb temperature ^{o}C	Sigma heat kJ/kg	Heat gain kW	Moisture content g/kg	Moisture gain g/s
3	12.0	13.7	34.46		8.35	
				41.50		3.99
4	12.4	14.5	35.50		8.45	
				86.59		25.54
5	13.2	13.1	35.67	(-15.66)	9.09	
				82.19		19.96
6	14.0	16.3	39.73		9.44	
				88.58		28.33
7	14.8	16.8	41.95	(-12.52)	10.15	
			TOTAL	299.52	TOTAL	71.98

Air mass flow rate 39.9 kg/s

'Old Loader'

8	10.8	13.9	31.73		7.15	
				0.15		7.50
2	10.9	12.6	31.74	(+8.16)	7.65	

Air mass flow rate 15.4 kg/s

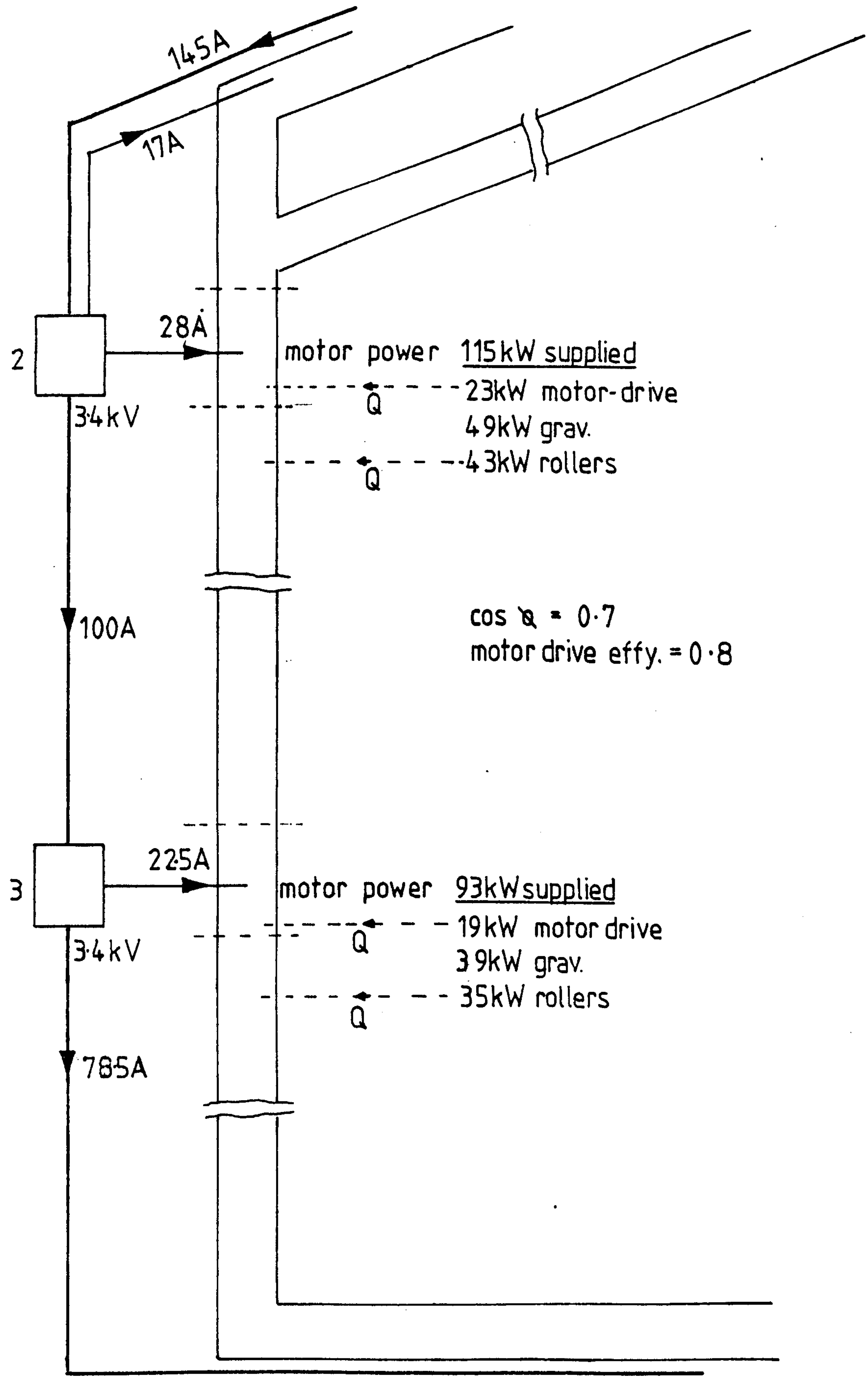


FIG 10·7d POWER USAGE IN 5s CONVEYOR ROAD

10.7.6 Power measurements

No. 3 Conveyor (Stations 7-5)

$$\text{Power drawn} = \sqrt{3} \times V \times I \times \cos\phi$$

$$= \sqrt{3} \times 3{,}400 \times 22.5 \times 0.7$$

$$= 92750 \text{ W} = \underline{92.75 \text{ kW}} \qquad (1)$$

$$\text{Motor drive losses} = \text{Power used } (1 - \text{Effy})$$

$$= 92.75\ (1 - 0.8)$$

$$= \underline{18.55 \text{ kW}} \text{ (heat to section 5.6)} \qquad (2)$$

$$\text{Power to lift coal} = \text{Mass flow rate} \times \text{gravity} \times \text{lift height}$$

$$= 125 \times 9.81 \times 32$$

$$= 39240 \text{ W} = \underline{39.24 \text{ kW}} \qquad (3)$$

$$\text{Power to overcome roller friction} = 1 - (2 + 3)$$

$$= 92.75 - (18.55 + 39.24)$$

$$= \underline{34.96 \text{ kW}}$$

This is dissipated evenly along the conveyor in sections 7-6, 430 m and 6-5, 50 m in length. Its distribution is proportional to the conveyor length in a given section.

$$\text{Section 7-6,} \quad \frac{430}{480} \times 34.96 = \underline{31.32 \text{ kW}}$$

$$\text{6-5,} \quad \frac{50}{480} \times 34.96 = \underline{3.64 \text{ kW}}$$

No. 2 Conveyor

Power drawn $= \sqrt{3} \times V \times I \times \cos\phi$

$= \sqrt{3} \times 3400 \times 28 \times 0.7$

$= 115423 \text{ W} = \underline{115.42 \text{ kW}}$ (1)

Motor drive losses = Power used (1 - Effy)

$= 115.42\ (1 - 0.8)$

$= \underline{23.08 \text{ kW}}$ (heat to section 3.4) (2)

Power to lift coal = Mass flow rate x gravity x lift height

$= 125 \times 9.81 \times 40$

$= 49050 \text{ W} = \underline{49.05 \text{ kW}}$ (3)

Power to overcome roller friction $= 1 - (2 + 3)$

$= 115.42 - (23.08 + 49.05)$

$= \underline{43.29 \text{ kW}}$

This is distributed evenly in 3 sections.

Section 5-4, $\frac{440}{540} \times 43.29 = \underline{35.27 \text{ kW}}$

4-3, $\frac{50}{540} \times 43.29 = \underline{4.01 \text{ kW}}$

3-1, $\frac{50}{540} \times 43.29 = \underline{4.01 \text{ kW}}$

10.7.7 Coal bulk temperatures

(i) Start at No. 3 conveyor, near station 7

Inbye	18.8	18.9	18.9	18.4	20.1	
Outbye	18.9	18.7	18.3	19.3	19.0	mean = 18.93°C

(ii) Transfer point No. 3 conveyor to No. 2 conveyor near station 5

Inbye	18.1	18.9	18.7	18.4	18.7	
Outbye	17.9	18.7	18.2	18.2	18.9	mean = 18.47°C

(iii) End of No. 2 conveyor between near station 1

Inbye	18.7	17.9	16.8	17.8	18.4	
Outbye	17.8	18.3	17.6	17.6	18.2	mean = 17.86°C

<u>Heat lost by coal on conveyor No. 3</u> (stations 7 to 5)

Temperature fall x Specific heat x Mass flow rate of coal

(18.93 - 18.47) x 1.23 x 125 = 70.72 kW

Estimated distribution <u>63.35 kW</u> in section 7-6

<u>7.37 kW</u> in section 6-5

<u>Heat lost by coal on conveyor No. 2</u> (stations 5 to 1)

(18.47 - 17.86) x 1.23 x 125 = 93.79 kW

Estimated distribution <u>76.42 kW</u> in section 5-4

<u>8.68 kW</u> in section 4-3

<u>8.68 kW</u> in section 3-1

The heat from the coal is assumed to be distributed uniformly along the conveyor run and is distributed according to length.

10.7.8 Surface temperature measurements

Location No. 2 Conveyor Motors

Air wet/dry bulb temperatures 12.5/14.0°C

	Temperature °C
Coal surface (light load)	13
(medium load)	14
(full load)	14
Belt underside	14
Floor of roadway	13
Roof of roadway	12
Motor case	39
Motor chassis	17

Location 30 m below No. 2 motors

Air wet/dry bulb temperatures 13.0/15.1°C

Coal (light load)	14
(medium load)	14
Coal large lump	17
Belt, top and bottom both surfaces	14
Rollers	15
Roof of roadway	13
Side of roadway	13
Floor of roadway	14

Location No. 2 belt 30 m above No. 3 motors

Air wet/dry bulb temperatures 13.5/15.2°C

Coal (light load)	14
Coal internal temperature (belt stopped)	18

Location No. 3 belt 30 m below motors

Air wet/dry bulb temperatures 14.0/16.2°C

Coal (medium load)	15-16

Location Loading point No. 3 belt (Station 7)

Air wet/dry bulb temperatures 14.5/15.3°C

Coal	15-16
Roadway walls	16

Coal temperatures on No. 3 belt

Coal onto belt, surface	17
Coal onto belt, internal	17
At 240 m, surface	17-16
At transfer point, surface	16
At transfer point, 25 mm below surface	17
At transfer point, 50 mm below surface	17

Coal flow rate

115 kg/s at time of survey.

10.7.9 Moisture content of coal

Sample	Location	Moisture content %	Mean %
1	No. 2 conveyor discharge	4.051	
			4.024
6	No. 2 conveyor discharge	3.998	
2	Transfer point No. 3 to No. 2	4.212	
			4.281
5	Transfer point No. 3 to No. 2	4.350	
3	No. 3 conveyor load point	4.521	
			4.567
4	No. 3 conveyor load point	4.613	

The moisture content percentages of the samples taken are shown above. The figures were not used in the heat balance as the heat required to evaporate off the amount of moisture measured could not have been present. The heat required is calculated from the product of the mass flow of coal, change in moisture and latent heat of evaporation of water. Inserting the figures obtained for either end of the test length we have

Mass flow of coal x Change in moisture content x Latent heat

kg/s (x - x) kJ/kg

125 x (0.04567 - 0.04024) x 2430

= 1649 kW

Clearly this figure is outside the bounds of acceptable results and an order of magnitude too great. These results were not used in the final heat balance.

Figure 10.8a

Table of heat balances in 5's conveyor road

Section	1 Heat gain measured kW	2 Auto compression kW	3 Corrected heat gain (2 - 1) kW	4 Motor heat kW	5 Roller heat kW	6 Coal heat kW	7 Total conveyor heat 4 + 5 + 6 kW	8 Balance (7/3) %
3-4	41.50	-	41.50	23.08	4.01	8.68	35.77	86.2
4-5	86.59	15.66	70.93	-	35.27	76.42	111.69	157.5
5-6	82.19	-	82.19	19.55	3.64	7.37	30.56	37.18
6-7	88.58	12.52	76.06	-	31.32	63.35	94.67	124.5
TOTAL 3-7	298.86	28.12	270.74	42.63	74.24	155.82	272.69	100.7

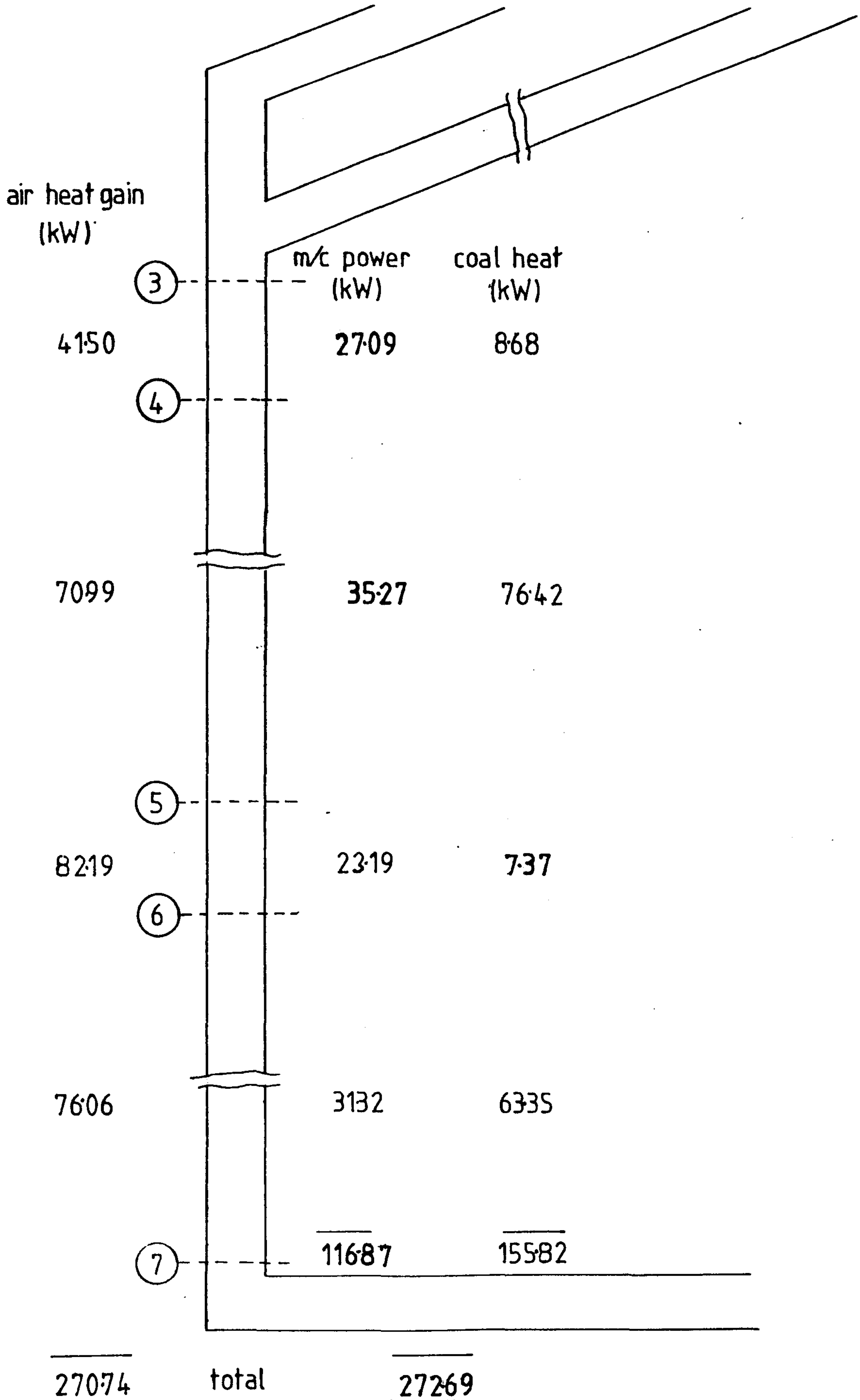


FIG 10·8b HEAT BALANCE FOR 5's CONVEYOR ROAD

10.8 HEAT DISTRIBUTION AND BALANCE

10.8.1 Heat balance

The results of the surveys reported in section 10.7 may be grouped together to form a heat balance. This allows comparison of the various sets of results obtained and increases understanding by presenting a composite integrated result. The balance which is attempted is for the test length of conveyor road between stations 3 and 7 in production conditions.

Using the information presented in table 10.7c it may be assumed that strata heat flow into the test length was negligible. This assumption is based on the observations that the dry bulb temperature in the test length was very close to VRT and that in the 'old loader', a very similar roadway to the conveyor road the strata heat was also negligible.

We may now attempt to balance the heat gain actually measured in the air along each section against that estimated from the power usage and coal cooling calculations. The results are best presented in a single table which includes the sectional values for each survey over the entire test length. The table, figure 10.8a shows the heat gain actually measured in the air in each section with a correction for auto compression if applicable. The heat dissipated by the electric motors, conveyor rollers and conveyed coal are also given for each section and the sum of heat estimates for these sources is compared with the heat measured in the air. The overall air heat gain in the test length is also balanced against the total machine and coal heat dissipated. This situation is also shown graphically in figure 10.8b.

10.8.2 Moisture evaporation

It may be seen from table 10.7b that in non production conditions water evaporation into the airway at 10.00 g/s and from table 10.7c in production conditions when the conveyor is carrying moist coal, 71.98 g/s of water evaporates.

It may be assumed that during production the moisture evaporation from the strata stays at almost the same level as moisture sources do not interact so much as heat sources. Due to the VRT and hence the strata water temperature being very close to the air temperature the strata water evaporates adiabatically as explained in Chapter 3.8. The water temperature is close to the air wet bulb temperature and therefore does not change the sigma heat content of the air.

The remaining 61.98 g/s of water evaporated is attributable to the conveyed coal. The heat required to evaporate 61.98 g/s is 150.61 kW. (0.06198 kg/s x 2430 kJ/kg). The heat dissipated by the coal was 155.82 kW. This indicates that almost all the heat transferred from the coal was used to evaporate its own moisture. The heat transferred from the coal as it cools is not being transferred sensibly to the air but to its own water which evaporates.

10.9 DISCUSSION AND ANALYSIS OF RESULTS

10.9.1 Errors and estimates

In such a complicated survey with many different aspects studied in a changing situation estimates must be made. Errors will be present in estimations and assumptions as well as in actual measurements. All estimates made are justified in the authors or other workers experience and are usually based on previous detailed measurement. Where possible results were processed to cause errors of measurement or estimation to counteract rather than compound. The errors due to each estimate or measuring method are discussed individually in the following sections. The overall heat and moisture balances which provide a check support the estimates made and methods used.

10.9.2 Continuous temperature record

The continuous temperature records for either end of the test site display the patterns of production and non production a surprisingly long way outbye. Once again a daily and weekly pattern is discernible on both traces albeit with slightly less definition than such records made closer to the workings have shown.

Surface conditions naturally have more influence this far outbye and the gradual trend of a working district to heat up through the week is not seen here.

The stabilisation of temperatures at the weekend during non production is apparent as is the sudden rise and fluctuation as production starts on a Monday.

10.9.3 Heat survey in non production conditions

The reason for this survey was to find the moisture pick up of air travelling along the test length in conditions when no machinery was working and also to measure the strata heat load when the air temperatures were well below that of the strata. It can be seen in table 10.7b that the strata heat pick up of the air is greatly increased at the weekend when no other heat sources are present to raise the air temperature. The moisture pick up by the air of 10 g/s is extremely low for this length of airway (1 km) and airflow, and confirmed the visual estimate of a dry roadway.

10.9.4 Heat survey in production conditions

The results shown in table 10.7c are for the last survey carried out which also used the measurements and calculated values from earlier surveys to attempt a heat balance. Consequently this survey was carried out as quickly and accurately as possible with two personnel observing all readings and repeating the readings if any pair did not agree.

The heat and moisture content of air is specified by pressure and air wet and dry bulb temperatures. An air volume flow rate is needed to find heat and moisture pick up rates. The air pressure measurements would contribute negligible errors to the results since the measurements were made using a 0.01"Hg,(0.03 kPa) barometer and the heat is mainly dependent on temperature. Both wet and dry bulb temperatures were measured using 0.1^{o}C thermometers resulting in a maximum error of 0.25 kJ/kg in sigma heat and

4 kW heat gain per section and for the whole test length. An error of heat gain in one section would have been counteracted by a similar error of opposite sign in the next section.

The air flow was calculated from a mean of three anemometer readings at designated measuring stations of known cross sectional area. The estimated error here was up to 2.5%. This could affect the total heat pick up by up to 7.5 kW.

The air dry bulb temperatures measured along the test length, 13.7^{o} to 16.8^{o}, were fortunately very close to the VRT of 16^{o}C. Indeed the dry bulb temperature of 16^{o}C occurred between stations 5 and 6 half way along the roadway. Thus a slight heat gain in the top sections of the airway would be offset by a slight heat loss in the bottom section. The net effect being a negligible strata heat gain. The low strata heat gain in the old loader of only 8.16 kW in 650 m at a lower dry bulb temperature confirms that this assumption is valid.

10.9.5 Electrical power measurement and distribution

The electrical power supply to the motors naturally fluctuated slightly with the changing coal load. The voltage remained stable at 3.4 kV, but the power fluctuated by up to 2 A. It may be assumed that there was no long term drift between the readings at the two sub stations since the meters on the cable connecting No. 2 to No. 3 sub stations read the same at both ends. Past experience showed this indicated a stable load. The meter accuracy was claimed at 5%. The largest error in assessment of power drawn was probably in the estimate of the power factor, cos ϕ. This could be up to 15%.

The distribution of the power drawn was subject to errors also, but these would not compound the errors made in measurement of total power drawn but merely in the apportioning of it. The estimate of motor drive losses of 20% of power drawn appears in the case of both conveyors to be too low probably due to the system being run at only one third of its rated load. The effect of under estimating the power used to overcome inefficiencies in the motor-drive unit would be an over-estimation of the power used to overcome roller friction along the belt length. The table of heat balances seems to indicate this in what might have happened. The amount of power required to raise the coal load against gravity would be sensitive mainly to errors in estimation of coal flow which would fluctuate slightly whilst the height of lift was known exactly. The coal flow rate was calculated from the tonnage run per hour monitored at the bunkers accurate to 1%. Therefore whilst the flow rate at a particular instant was not known exactly, the average flow rate for a particular hour was available since the survey took place over about 1 hour this was not such a problem for reasons discussed later.

10.9.6 Coal temperature and heat loss

The errors on coal temperature measurement are mainly in the sample taken. Ideally the same batch of coal should be sampled on its trip along the conveyors. This is impossible without a large group of operators. Taking a number of samples over a period and at a transfer point where mixing takes place, would reduce the errors which to a certain extent are unknown. The variation of temperatures measured at a particular station was more than the total temperature change along the conveyor. The accuracy of the heat balance is the only reliable indicator of the error and this appears good, but the possibility of compensating errors cannot be ruled out. The specific heat of the coal was measured in the laboratory and is such subject to very low errors (< 3%) compared with that of sample collection.

The errors in flow rate used to calculate the rate of heat loss have been discussed in the previous section.

The coal temperature and cooling survey was carried out on a different day to the main air temperature survey. However the flow rates and surface temperature measurements were similar so because of the long time needed to make a coal temperature survey and the need for a fast air temperature survey this appeared to be the most practical solution.

10.9.7 Surface temperature survey

The results of this survey were not used in the heat balance but the coal temperature was checked for the reasons described in the last section. Other surfaces of interest were measured mainly to confirm the observation often quoted that in a dry roadway the wall temperature is often very close to the air dry bulb temperature.

The coal temperatures on No. 3 belt supported the proposal given previously of a layer near the surface of the coal bed cooling and the temperature of coal within the bed not falling so quickly.

10.9.8 Moisture content of coal

The results of these measurements were very poor and not used in the heat and moisture balance. There are three reasons for this. The samples were too small in size for the required accuracy. The moisture content of the coal conveyed probably varied at the start of the conveyor road. Also the change in moisture content needed would be difficult to measure accurately by weighing on a balance robust enough to carry the coal sample. The moisture content change in the coal along the conveyor appears to have been in the region of 0.5 g/kg of coal. To measure this with any accuracy would require extremely good drying and a scale measuring to 0.005 g whilst carrying a 1 kg weight.

10.9.9 Heat balance

The heat balance between the heat measured in the air and that measured independently from the contributing heat source in the roadway gives an indication of the overall accuracy and reliability of measurement and estimation. The assumption of negligible strata heat seems to be justified by the heat balance.

The balances are low in sections 3-4 and 5-6 and correspondingly high in sections 4-5 and 6-7. This is likely to be the result of an overestimation of the efficiency of the conveyor motor-drive units causing more power drawn to be apportioned to the working lengths of the conveyors as described in section 10.9.5. The results could have been improved by changing the motor-drive efficiency estimated, but the overall balance would remain unchanged and such estimates made before processing results should not be changed for the sake of minor improvements in presentation.

This heat balance represents in the region of one hours production and is a pseudo-steady state interpretation since it was impossible to measure everything instantaneously. The error in overall heat balance of 0.7 % is considered a good result and gives reason for some confidence in the individual measurements of heat from the machinery and the conveyed coal. The heat from the conveyed coal is by far the largest contributor to the air heat pick up. In this case 156 kW of a total heat load of 264 kW. Naturally it is not possible on the strength of a single investigation to predict a contribution of such a scale in all cases, but this result and those obtained at other collieries suggest a larger contribution than is at present generally accepted.

The nature of the heat transfer appeared to have been in this case totally by evaporation of moisture. Once again there is a good correlation between moisture gain in the air, heat required to evaporate that moisture and heat lost by the coal. The situation was probably such that the coal transferred its sensible heat to the surface moisture which then evaporated. Heat transfer by evaporation was expected, but not to this extent.

CHAPTER 11

CONCLUSIONS

CHAPTER 11

CONCLUSIONS

11.1 SUMMARY

11.1.1 Laboratory investigations

The laboratory investigations made a valuable contribution to this study despite not providing values for characteristics and coefficients which could be used immediately to predict the actual underground effects of conveyed coal. The quantitative information gained, particularly the convective heat transfer coefficients, will be useful when more investigations have been carried out to link the latent and convective heat transfer. The model in the duct also allowed a quantitative study which should provide information which will allow more accurate choice of future investigations. Of particular interest are the peak heat flow rates from the coal and air penetration into the coal bed.

11.1.2 Theoretical analysis

Whilst it is difficult to comment on untried theoretical analyses the original approach is still considered sound. The treatment of a continuum rather than separate pieces of coal and air spaces appears the most practical approach to analysis and has precedents in other areas of fluid mechanics. The complexity of the analysis might be reduced by neglecting terms shown to be small by the underground survey. Radiative heat exchange for example. Possibly a much simplified analysis based on the one described in Chapter 7 could be of use. This analysis would require less coefficients to be evaluated by laboratory and underground work.

11.1.3 Underground investigations

The underground investigations were the most rewarding aspects of this study. The dynamic nature and interaction in the underground climate provided complications but allowed a fuller investigation by choice of site and timing to pick out or highlight specific heat sources. A large team of investigators would have allowed more accurate measurements in some cases. Notably on the conveyed coal investigations and temperature surveys but difficulties in matching instruments and techniques and control would have introduced some extra problems.

The adoption of measurement of a survey and analysis in a quasi-steady state of approximately one hour duration worked in the case study at Pye Hill Colliery, but careful examination of continuous temperature records will be necessary before justifying its use in other situations. The small error in overall heat balance gave confidence in the results and analysis.

11.2 FURTHER WORK

11.2.1 Laboratory investigations

Since underground investigations have highlighted the importance of the latent heat transfer it is important that any future laboratory work takes account of this. Experiments similar to those conducted in the insulated duct should be continued as such work allows conditions to be reproduced and carefully controlled.

Also examination in detail and safety compared with underground is facilitated. To provide firstly the close humidity control over a wide range and secondly the step change in temperature at the start of an experiment which were the main weaknesses in an otherwise suitable approach, the scale of the model should be vastly reduced. This would put the amount of heating, refrigerating and humidity control required within the range of that provided by a commercially available air conditioning unit. In the case of a much smaller model scale effect and similarity will need to be analysed. Also the packing and 'loading' of the coal bed would need careful consideration.

11.2.2 Theoretical analysis

Theoretical analysis should be matched to the accuracy of measurement and availability of input parameters. The equations used should still be based on well proven heat transfer equations or derived from first principles. The empirical approach whilst leading to apparent advances in the short term could lead to an analysis that breaks down when used outside the range of values tested.

The steady state analysis should be retained for its simplicity and suitability for inclusion into computer prediction programs which already exist.

11.2.3 Underground investigations

Much more underground measurement should be carried out initially for primary investigation and later for correlation of predictions in a wide range of conditions. The underground investigations should be of a similar type to that conducted at Pye Hill Colliery. This does not necessarily mean a length of airway with only a conveyor and no other machinery with negligible strata heat, but all heat sources should be accounted for and evaluated as accurately as possible. The results should form a heat balance with the conveyor and coal included as components. Such an approach, sacrificing quantity for quality will allow results which may be used with confidence not only for this work, but in the evaluation of other heat sources, heat storage and interactions. With regard to conveyors the power distribution between the motor drive and rollers would benefit from more detailed measurement and analysis. Information presently available is concerned mainly with the selection of drive motion rather than heat dissipation.

Heat storage phenomena particularly that of the strata surrounding an airway should be investigated. Strata heat storage could explain many anomalies and a quantitative evaluation would facilitate more detailed examination of other heat sources.

11.3 ESTIMATE OF CONVEYED COAL HEAT TRANSFER

At this stage a detailed prediction of the contribution of conveyed coal to the heat problem underground is not possible. However the project has produced observations which show the magnitude of the contribution at different locations underground and that conveyed coal is a significant contributor to the overall heat load. The figures obtained have been found to agree with other workers estimates.

11.3.1 Contribution to total mine heat load

The contribution of conveyed coal to the total mine heat load is of the order of 10% during production periods. The overall mean figure for a full week is about 5% (Ref. [9] compare with Bentinck Colliery, Chapter 6).

11.3.2 Contribution to heat load in intakes

The contribution of conveyed coal to the heat load in intake conveyor roads is of the order of 50%.

This figure was measured at Hickleton and Pye Hill Collieries and compares with estimates made by Voss [21] in the Ruhr Coalfield. The properties of heat from conveyed coal and from the strata remain approximately the same despite the higher VRT in the Ruhr. This is probably due to higher temperatures in the strata resulting in proportionally higher temperatures of mined coal on the conveyors.

The Germans found an estimate of coal cooling rate expressed in temperature fall per metre of conveyor which was not in agreement with results taken in this study. It was much lower.

11.3.3 Heat loss rate from coal on conveyors

The cooling rate of coal on a conveyor may be estimated by a simple empirical equation as follows

$$Q = 50\ (t_c - t_a)\ x \qquad (1)$$

where

Q = Heat loss rate from coal W

t_c = Mean temperature of coal ^{o}C

t_a = Mean air dry bulb temperature ^{o}C

x = Length of conveyor m

This equation takes account of more parameters than is immediately apparent since many variables, such as belt speed, do not vary greatly (2.5-3.0 m/s) and air velocities are limited by dust pick up considerations. Moisture content of the coal is always kept high because of dust suppression regulations and this allows maximum evaporation to take place.

11.4 CONCLUSION

The theoretical, laboratory and underground studies carried out in this project have demonstrated the nature and magnitude of the contribution of conveyed coal to the total underground heat load. In particular it is recognised that much of the heat is transferred in the form of latent heat of evaporation from the hot, wet coal to the cooler and drier ventilating air.

A satisfactory heat balance has been achieved using measurements from a carefully selected underground site with due account being taken of heat flows under operating and non operating conditions.

It has been demonstrated that conveyed coal can make a very significant contribution to the overall heat load and should be included in a computer program or other model used for the prediction of underground climatic conditions. In order to model the conveyed coal heat contribution accurately further work is required, but it is hoped that this study will provide a useful foundation.

APPENDIX 1

VAPOUR PRESSURE TABLE

Figure A.1a

Saturated vapour pressure (e) in kilopascals (kPa)

°C	0	1	2	3	4	5	6	7	8	9
0	0.610 8	0.656 6	0.705 5	0.757 5	0.813 0	0.871 9	0.934 7	1.001 3	1.072 2	1.147 4
10	1.227 2	1.311 9	1.401 7	1.496 9	1.597 7	1.704 4	1.817 3	1.936 8	2.063 0	2.196 4
20	2.337 3	2.486 1	2.643 1	2.808 6	2.983 2	3.167 2	3.360 9	3.564 9	3.779 7	4.005 5
30	4.243 1	4.492 8	4.755 2	5.030 8	5.320 1	5.623 8	5.942 3	6.276 4	6.626 5	6.993 4
40	7.377 8	7.780 3	8.201 6	8.642 4	9.103 4	9.585 6	10.090	10.616	11.166	11.740
50	12.340	12.965	13.618	14.298	15.007	15.746	16.516	17.318	18.153	19.022
60	19.926	20.867	21.845	22.862	23.918	25.016	26.156	27.341	28.570	29.846

APPENDIX 2

HICKLETON COLLIERY VENTILATION SURVEY RESULTS

Figure A.2a

Ventilation surveys at Hickleton Colliery

Station	Location	Pressure kPa	Quantity m^3/s	Static conditions Temperature		Production conditions Temperature	
				WB°C	DB°C	WB°C	DB°C
A	U/C shaft top (3)	97.95	206.0*	12.2	14.7		
B	U/C shaft bottom (3)	105.50		16.4	23.3		
C	W Return 24s slit	106.50	34.75*	21.7	29.6	24.5	30.2
D	W Return O/B 46s Return	107.75		22.2	30.2	26.0	30.7
E	4bs Return	107.75	12.55*	21.0	28.7	24.5	29.5
F	W Return I/B 46s Return	107.75		22.7	30.9	26.8	31.3
6	TO1s supply level	108.85		23.8	31.9	28.6	32.2
5	TO1s T/G 10 m O/B	108.10	11.44	21.8	30.0	29.3	32.3
4	TO1s M/G stage loader	108.50	11.34	19.5	28.9	24.7	29.1
3	W Plane at transformers	108.45		17.7	25.9	21.5	26.0
2	W Plane 1/B 46s intake	108.75	18.90	16.3	22.3	20.5	25.3
1	W Plane 1/B 24s slit	108.15	32.38	21.8	29.8	16.5	22.5
G	D/C shaft bottom (2)	108.25		16.3	21.7		
H	D/C shaft top (2)	106.60		13.4	15.4		

* Measured by colliery ventilation staff in previous survey

APPENDIX 3

INSTRUMENT CALIBRATION CURVES

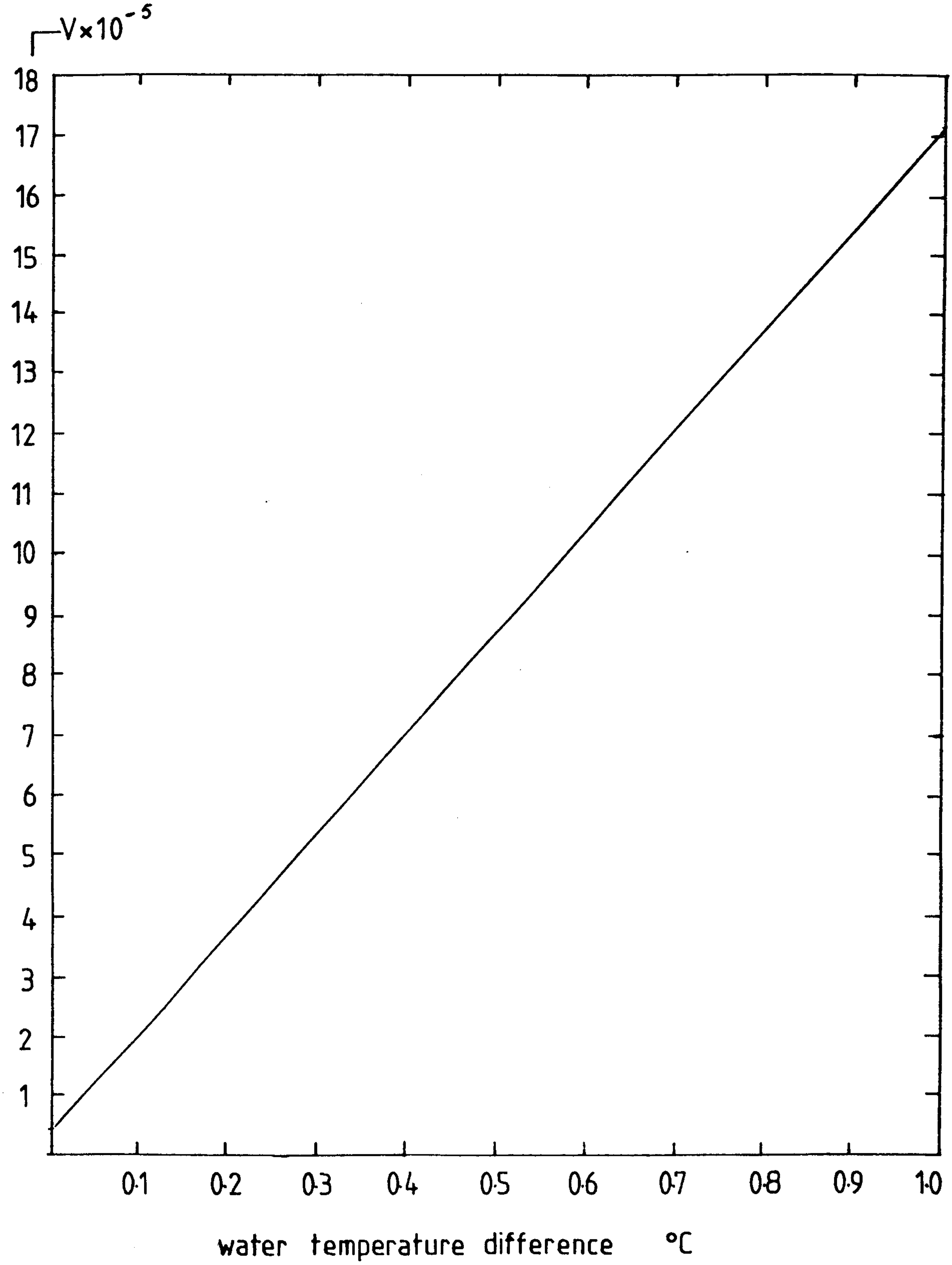


FIG A3a. THERMOCOUPLE CALIBRATION CURVE

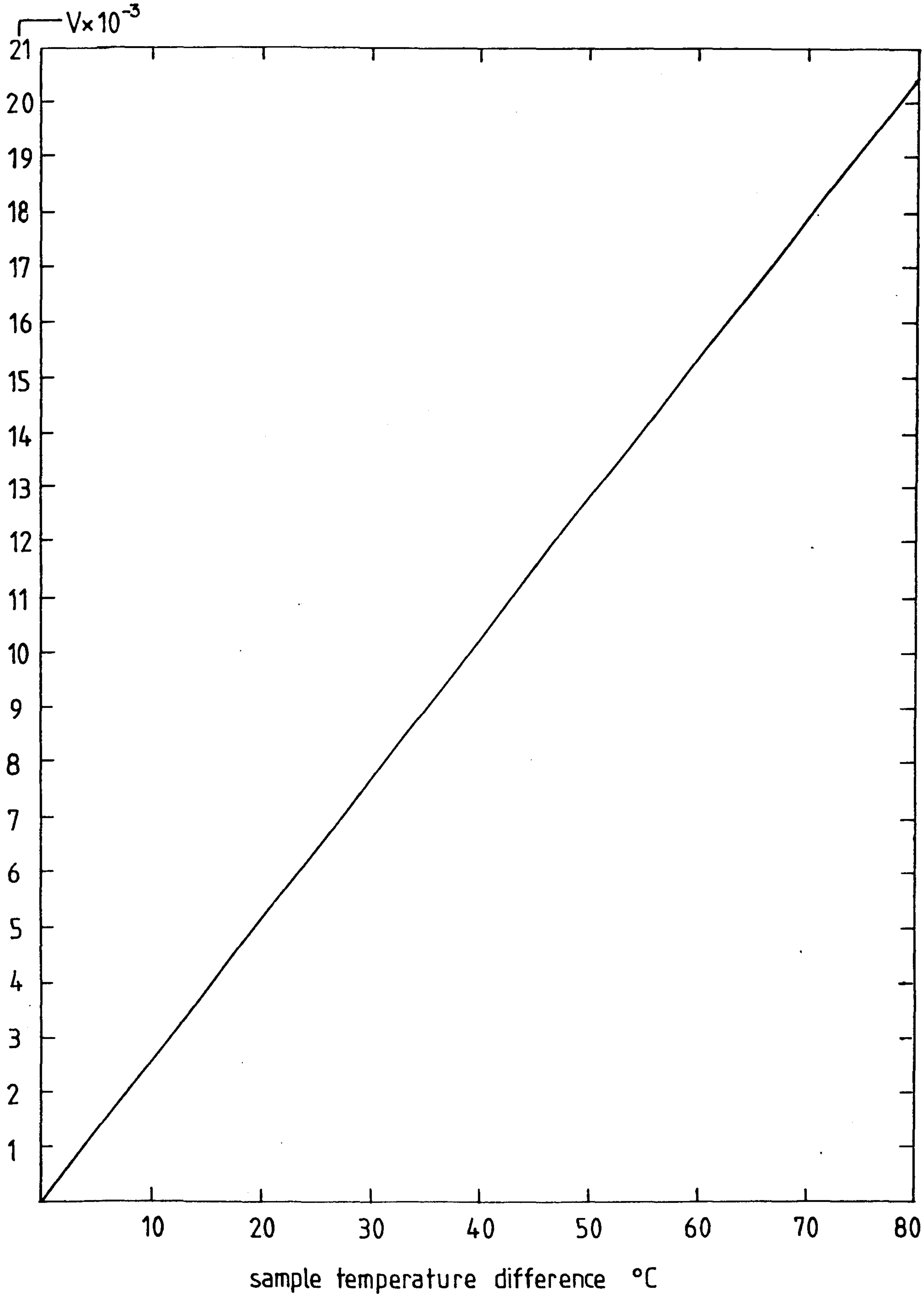
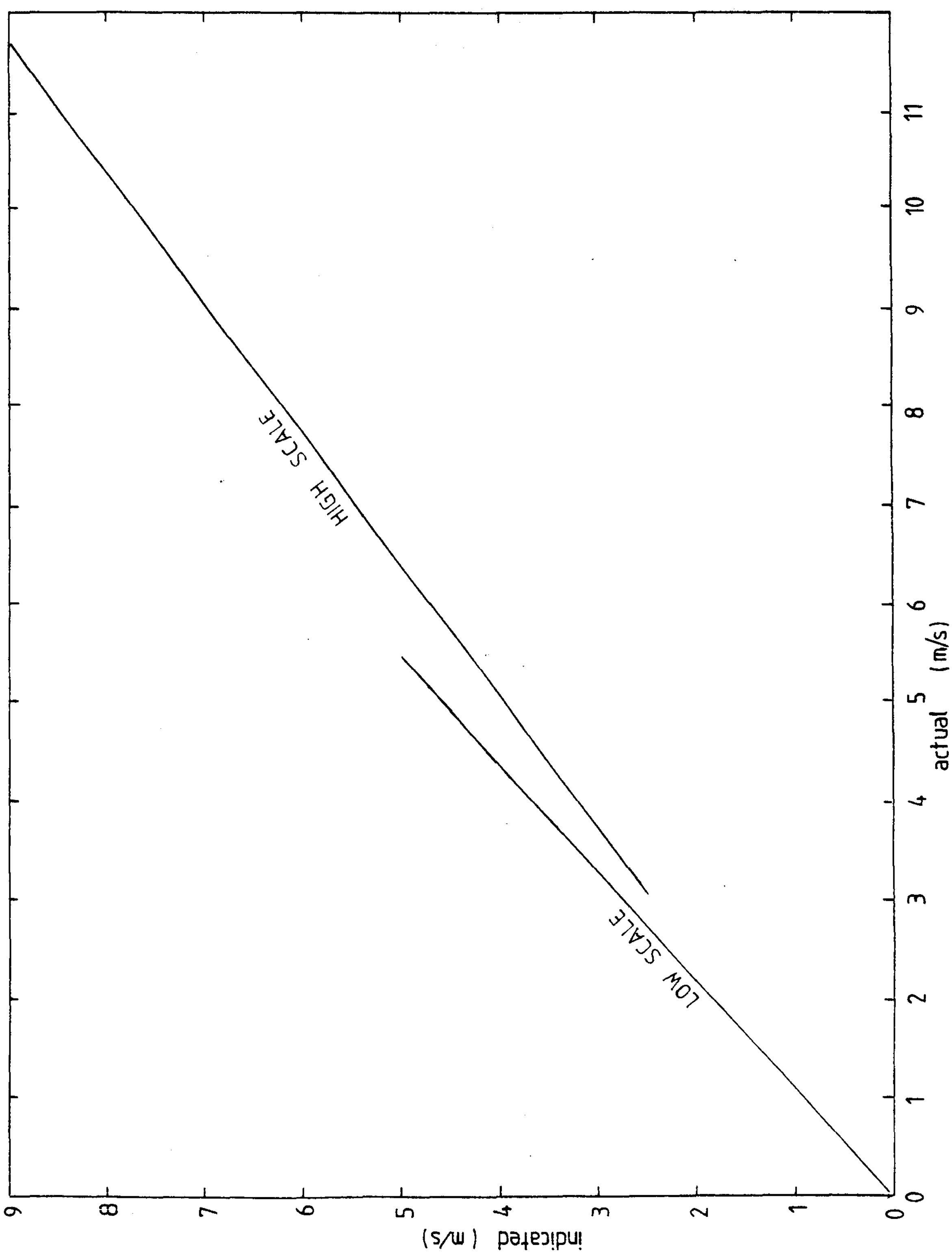


FIG A3b THERMOCOUPLE CALIBRATION CURVE

FIG A3c 'WALLAC' ANEMOMETER CALIBRATION



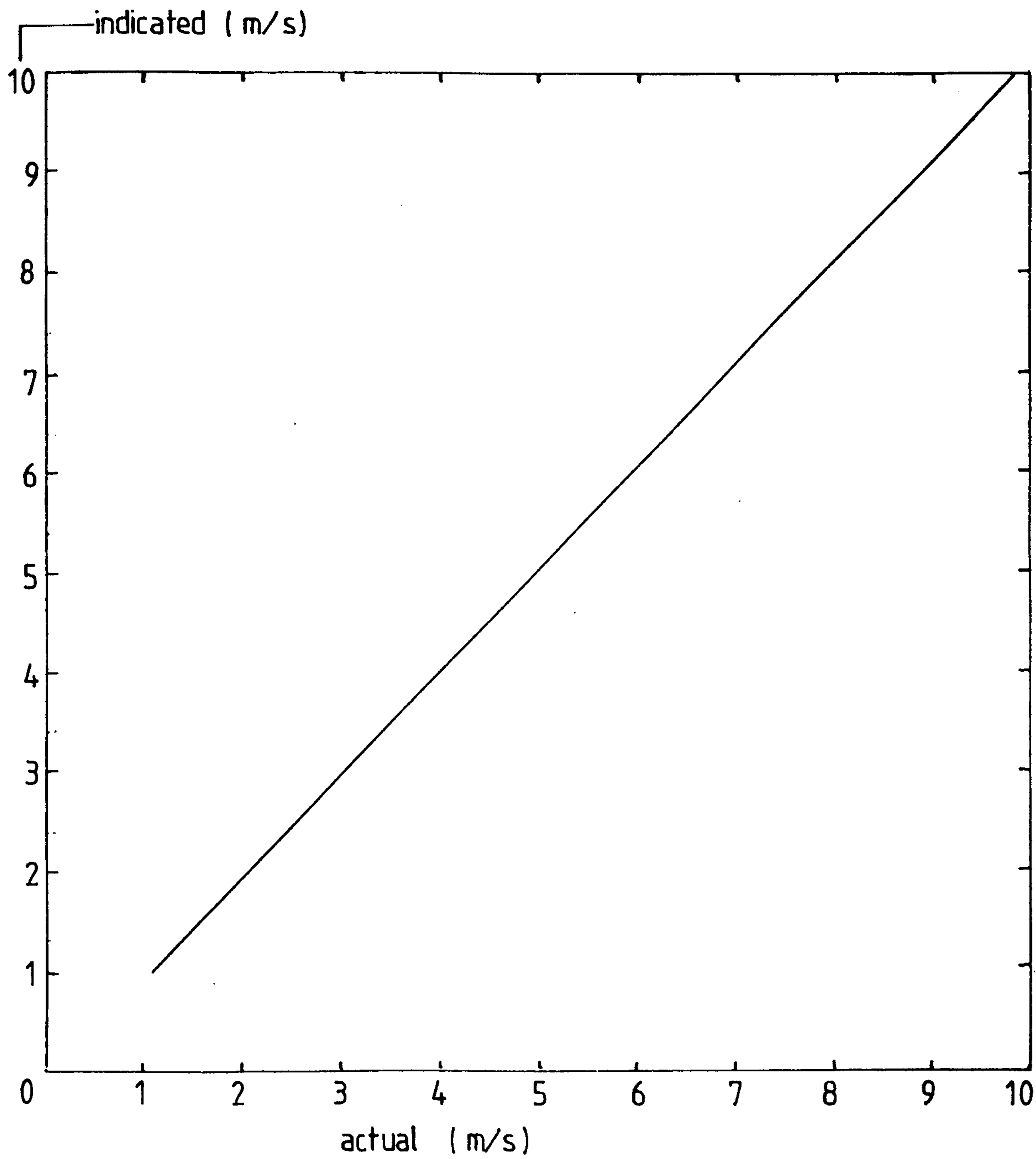


FIG A3d 'ALNOR' ANEMOMETER CALIBRATION

APPENDIX 4

SPECIFICATIONS AND OPERATING INSTRUCTIONS FOR 'INFRATRACE'

INDEX

SPECIFICATION	INFRATRACE 1000	INFRATRACE 2000
Applications	General Industrial Use Energy Conservation Measures glass temperature	High Temperature Processes Metal Foundry Measures through glass
Readout	12.7mm Digital Liquid Crystal Display	
Full Scale Measurement Range	0–1 000°C (Temperatures outside this range are displayed at reduced accuracy)	600–2 000°C (Automatic out-of-range indication)
Resolution	1°C	1°C
Accuracy (Mean) Notes 1,2	± 0.7% of reading ± 0.4% of full scale	± 1% of reading ± 0.4% of full scale
Repeatability Note 1	± 0.3% of full scale	± 0.3% of full scale
Emissivity Compensation	0.2 to 1.0	0.2 to 1.0
Spectral Range	8–14 μm	2–2.5 μm
Speed of Response	3 readings per second. <100 ms at recorder output	
Minimum Target Size	20mm diameter at 1 metre range	
Field of View	25 milliradians (1½°)	
Distance/Target Dia.	40:1	
Optical Sight	Integral. Parallax corrected	
Ambient Temperature (Operating) (Storage)	0°C to 45°C –20°C to 60°C	
Battery Type	9 volt PP3, TR146X, 6F22 (Heavy duty type recommended)	
Battery Life	30 hours, representing 3 months typical use	
Dimensions (excluding handle)	235 × 90mm (9¼ × 3½ in)	
Weight	980gms (2lb 2½ oz)	

Notes: 1. Applies at ambient temperature of 18°C to 28°C
2. Assuming black body target emission. Emissivity = 1.

Getting the best results from INFRATRACE

1. Introduction

INFRATRACE incorporates an optical system capable of producing highly consistent and accurate measurements. This is matched by automatic processing features, readily selected by pressing the appropriate push button. Errors due to atmospheric attenuation and extraneous heat sources, such as the sun, are rendered negligible in most applications.

In fact operation could not be simpler—

AIM the instrument

PRESS the trigger, and

READ the temperature immediately in °C.

The following notes are presented for the guidance of the user in obtaining the optimum performance from INFRATRACE in a host of diverse applications.

2. Basic Principle of Operation

Any surface at a temperature above absolute zero emits heat in the form of radiated energy. At temperatures above 600°C some of this energy is visible, but much of it is of longer wave length than visible light (i.e. infra-red) although it behaves in the same manner as visible light. It travels in straight lines, and may be reflected by mirrors and focused by a system of lenses. The radiant flux from the surface is directly related to its temperature by known physical laws. In a non-contact thermometer, such as INFRATRACE, it is this energy which is measured, the value being processed to display temperature directly in degrees Celsius.

A more detailed knowledge of these physical principles is necessary only in a minority of applications and a more comprehensive description is therefore presented in the Appendix

3. General Operating Instructions

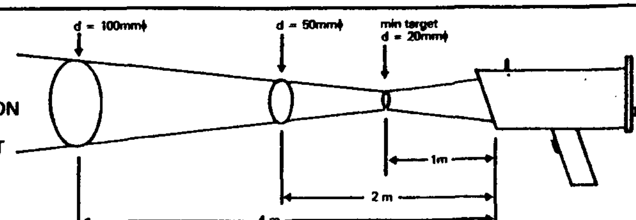
3.1. Aim INFRATRACE at the target

Remember to loop the wrist strap around the wrist. INFRATRACE may be held and aimed like a pistol. The aperture of the front sight should be aligned with the appropriate aperture of the rear sight forming an extension to the Display/Control Panel. The upper of these two apertures provides parallax correction when the measured object is at the optimum distance of 1 metre from the instrument. The lower aperture provides a parallel sight line suitable for aiming at targets at a distance greater than 5 metres.

INFRATRACE will effectively measure surface temperatures at literally any distance — from a position almost touching the instrument to infinity — provided that the target fills the small field of view of the instrument. (See Appendix for explanation of atmospheric attenuation effects.)

A fixed focus optical system defines a minimum target size of 20mm diameter at 1 metre distance, and a DISTANCE/TARGET SIZE ratio of 40:1 at greater distances. This is illustrated in figure 1. In practice it is desirable to have a target diameter twice that specified, or to operate at a maximum of half the specified distance (but not less than 1 metre) in order to facilitate correct aiming, and to minimise errors due to hand movement

Fig. 1.
SPOT SIZE DEPENDS ON DISTANCE TO TARGET (1 : 40)



3.2. Measure

Press the trigger button in the handle and the temperature reading is displayed almost immediately. The reading is updated 3 times per second — as fast as the eye can easily read it. Provided the setting of the emissivity control (Section 3.6) is correct the reading is displayed digitally in °C. Nothing more is required.

3.3. Memory

If it is inconvenient to read the digital display while concentrating on accurate aiming of the instrument, then the Memory function may be employed. Press the appropriate button [M] on the Control Panel. Aim the instrument and press the trigger as before. The reading updates, as before, 3 times per second. When the trigger is released the final reading is held for 10 seconds — giving ample time to note it without ambiguity, after relaxing from the aimed position. Normal operation is restored by pressing the right-hand reset button [·] (The Memory may be used together with the Average and Peak functions by pressing two buttons simultaneously.)

3.4. Average

It is often necessary to measure the temperature of a moving surface — a rolling drum, or a continuous belt or strip process — where the temperature is not uniform. A similar situation exists with a stationary surface when the operator scans the instrument. This would often produce rapid fluctuations in the high resolution digital display which would sometimes be difficult to interpret. Simply press the "Averaging" button [AV] and INFRATRACE performs the mental arithmetic accurately and immediately, smoothing out the peaks and troughs to give a mean reading.

A typical situation is illustrated graphically in figure 2. The solid curve shows the change in energy detected by the instrument over short passage of time, as would be shown on a chart recorder. The effect on consecutive digital readings is shown both with and without the averaging facility in operation.

The average function is cancelled by operation of the reset button [·]

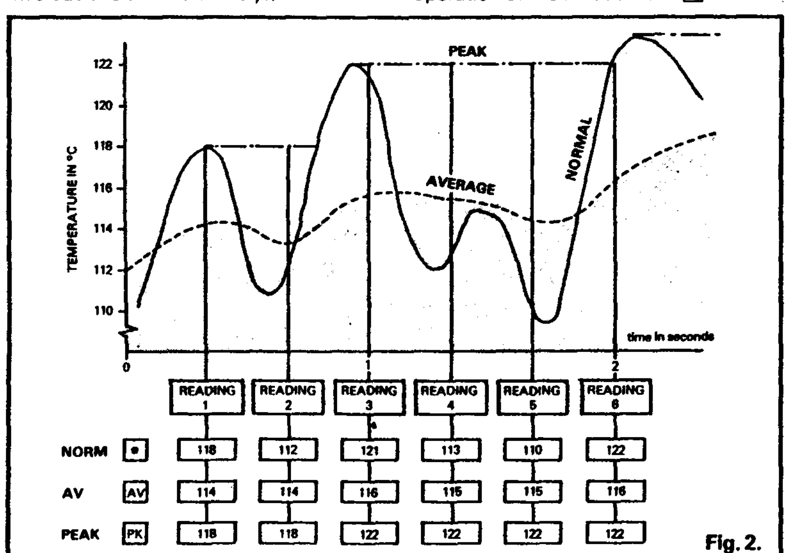


Fig. 2.

3.5. Peak Reading

Conversely, it may be useful to highlight the fluctuations in a dynamic process — in fact to seek and measure the hottest spot. Since INFRATRACE responds in about 50 milliseconds (much faster than the eye) a hot spot can be found very quickly. Press the "Peak" button [PK] and operate INFRATRACE normally. The highest temperature seen by the instrument is held indefinitely until an even higher temperature is recorded — at which point the display updates. This also is shown in figure 2.

When the trigger is released, the circuit will reset to below zero, so as to be ready to record a new peak temperature (which might be lower than the previous peak) when the trigger is pressed again. The reset button [·] restores INFRATRACE to the normal mode.

3.6. Emissivity

For a full explanation of the meaning of emissivity turn to Section 4. In practice the Emissivity Compensation Control should be set to the value established for a particular material and surface, before each measurement is made. The instrument automatically applies any necessary correction.

The value of emissivity may be established by reference to tables, as in the Appendix, or by a special measurement which need be done only once, on the surface concerned.

3.7. Fixed Operation

The wrist strap may be removed by unscrewing from the base of the handle, to expose a standard ¼" BSW Camera Tripod Bush. INFRATRACE may then be supported on even a lightweight camera tripod to monitor a fixed position.

An analogue output signal may be taken from the right-hand 2.5mm jack socket to feed a chart recorder or remote digital display. An ordinary digital voltmeter may be used for this latter purpose since the output is calibrated LINEARLY such that 0 volts represents 0°C and, for example, 100mV represents 100°C. The external equipment should have a high input resistance (not less than 100 kilohms). The output signal has a time constant of only 50mS, or approximately 1 second when the [AV] button is pressed. The output is not affected by operation of the [PK] button or [M] button.

External power may be provided for continuous use through the left-hand 3.5mm jack socket in which the tip is positive and the sleeve is negative. The power supply required is nominally 9 volts at up to 15mA, and a standard Kane-May adaptor (as used for DIGITHERM and other instruments) is perfectly suitable. The power supply *must* be floating, i.e. not electrically connected to earth or to the Recorder output.

3.8. Battery

When the loaded battery voltage falls below 7 volts, all the decimal points show in the digital display. The instrument may be used for a few further measurements before the accuracy of measurement may become affected. The battery should be replaced immediately.

The battery is contained in a small cartridge at the base of the handle which slides out after the large dome-headed screw (use a small coin if a screwdriver is not available) is removed from the handle. Replace with a suitable 9 volts transistor battery type PP3 (IEC designation 6F22) or a manganese alkaline equivalent for even longer life.

4. Establishment of Emissivity (see also Appendix A.1)

The table of emissivities on page 11 is provided as an approximate guide to the correct emissivity settings for a variety of materials. Most non-metallic substances have an emissivity of around 0.9, i.e. they are 90% "efficient" in emitting energy, and no difficulty should be experienced in obtaining accurate measurement. Metal surfaces which are efficient reflectors are less efficient emitters of energy and consequently exhibit lower values of emissivity.

It should be noted that, due to the optical design, INFRATRACE 2000 is less sensitive to errors in estimating emissivity. As an example consider a metal surface at 1000°C with an emissivity of 0.5. If the emissivity was set erroneously to 0.6, then INFRATRACE 1000 would read approximately 930°C, an error of 70°, and INFRATRACE 2000 would read 965°C, an error of only 35°C.

Where convenient, if the table gives a low value of emissivity, the surface may be coated with a thin layer of carbon black, or matt black paint. This will not interfere with the heat distribution of the material, but will render the emissivity greater than 0.9 facilitating measurement.

Alternatively the tabulated value may be used. Since it is not possible to give precise values for all the materials and surface conditions encountered in industry, it is desirable to establish the value by direct measurement. When this has been done – only once – the established value may be used for all future measurements.

Using a surface contact thermometer if necessary halting a moving process, measure the surface temperature directly. The measuring instrument must be allowed to stabilise, since the speed of response is significant using this technique. Take away the probe and immediately point INFRATRACE at the same area of the surface. Adjust the Emissivity Control to obtain the same reading as for the contact measurement. Read off from the Emissivity Control the value appropriate to the material – this may be used for all future measurements.

If only qualitative indications are required, e.g. temperature differentials and localised hot spots, then the value given in the table is quite adequate. Alternatively the Emissivity Control may be set to 1 for simplicity. INFRATRACE will, of course, perform consistently from day to day provided that the same setting of the control is employed on every occasion.

5. Further hints for best results

(a) Do not point the instrument at heat sources hotter than the stated measurement range of the instrument. The sensitive infra-red detector may become damaged.

(b) After measuring high temperatures at the upper end of the quoted temperature range, it may be necessary to wait several minutes before measuring very low temperatures. The sensitive detector in INFRATRACE 1000 may absorb considerable heat when exposed to 1000°C which must be dissipated before precise measurement of the very small energy from room temperature sources can be undertaken.

(c) When searching for small temperature differences at room temperature (e.g. locating faults in under-floor heating), and where absolute measurement of temperature is not required, the sensitivity of the instrument may be increased by setting the Emissivity Control to 0.2. Temperature differences of 0.2°C may then be resolved, (as a change in the displayed reading of 1°C).

(d) INFRATRACE 2000 will not measure temperatures below 600°C or above 2000°C. Out of range indications of respectively $\boxed{000}$ and $\boxed{1\quad}$ are displayed.

INFRATRACE 1000 will display negative temperatures, which are repeatable, but are subject to errors greater than temperatures within the specified range.

APPENDIX

The foregoing instructions provide all the basic information to enable the successful application of INFRATRACE to a great variety of measurement tasks. This appendix presents further background information, as an introduction to infra-red measurement techniques; which may prove useful in ensuring the best approach to new measurement problems.

A.1. Radiant Heat Emission from a surface

To understand the mechanism of infra-red transmission we must answer a fundamental question – what is heat? In any heated material – that is a substance having a temperature above absolute zero (–273°C) – the molecules are not stationary but are vibrating. Energy is stored in this vibrating motion, and when more energy is added to the material (when it is heated to attain a higher temperature) the vibration increases. In effect when we measure its temperature by any method we are measuring the magnitude of this vibration.

This molecular motion in turn produces electromagnetic waves, similar in nature to radio waves and visible light, so that some of the energy in the material is radiated in straight lines away from the surface of the material. INFRATRACE is an instrument which uses normal optical principles of reflection to sample this radiated energy, which, being a function of the molecular vibration provides an indirect measurement of the surface temperature.

Infra-red energy behaves in just the same way as visible light. It may be refracted in a lens or prism, and reflected by mirrors.

The instructions for using INFRATRACE describe a procedure for establishing the emissivity of a specific material. The concept of emissivity may be better understood by considering all the sources of radiant energy acting over the area of the surface viewed by INFRATRACE (see figure 3).

1. Transmitted energy –
   most materials are opaque at infra-red wavelengths, so this contributes negligible energy.
2. Reflected energy –
   generally this is energy reflected from the walls or floor of the building and is insignificant.
3. Emitted energy –
   this is the component we are interested in measuring to establish temperature.

The efficiency with which these contributions are generated by the surface is denoted by Transmissivity, Reflectivity and Emissivity respectively, and when expressed as fractions of unity, the sum of the three terms is unity. Since we can assume in most cases that the transmissivity is zero then it is clear that

Reflectivity + Emissivity = 1
for any surface

Thus a material which is a very poor reflector has an emissivity of approximately 1. Almost all non-metallic solids and liquids are poor reflectors and typically have an emissivity between 0.8 and 1. Highly polished metal surfaces however are good reflectors, and a mirror-like surface with a reflectivity of 0.9 (reflection efficiency of 90%) will have a very low emissivity of 0.1.

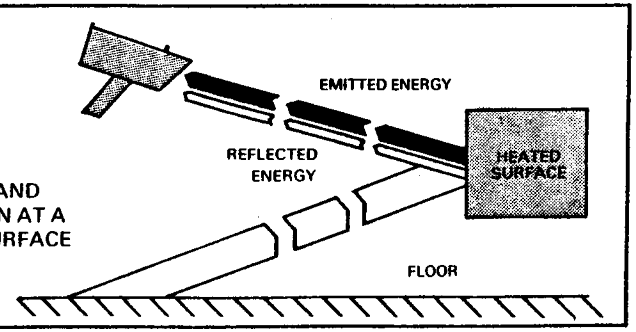


Fig. 3.
EMISSION AND REFLECTION AT A HEATED SURFACE

A rough surface will generally have an effective emissivity which is higher than a smooth surface of the same material. The reason for this is apparent in Figure 4. The reflected energy at the point viewed by the instrument originates at another point on the surface of the material. The effect of multiple reflections of this kind is to enhance the effective emissivity. The same applies when measuring the inside surface of a tube or cavity, or in woven or tufted fabrics.

A.2. Spectral Distribution of Energy

The human body makes the distinction between heat which can be felt, and light which can be seen. In scientific terms we say that infra-red energy has a wavelength longer than that of visible light.

When a material is heated, the molecular vibration increases both in magnitude and in frequency – it vibrates faster. When the resulting electro-magnetic energy is transmitted from the surface, at the speed of light, its wavelength is shorter.

This is well illustrated by the radiant electric heater, or 'bar-fire' consisting of a heater element and metal reflector. When first switched on a small amount of radiant heat may be felt when the hand is held close to the reflector. This is long wavelength energy. As the temperature rises, the radiant heat increases steadily, and eventually the heater is seen to glow with a red colour – shorter wavelength visible light. The distribution of energy of different wavelengths is referred to as the energy spectrum. Figure 5, below, shows the relationship between visible light wavelengths and infra-red wavelengths. It may be useful to consider infra-red energy as yet other "colours" which happen to be invisible to the human eye.

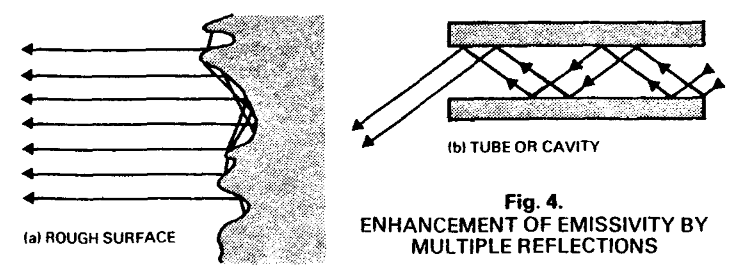


Fig. 4.
ENHANCEMENT OF EMISSIVITY BY MULTIPLE REFLECTIONS

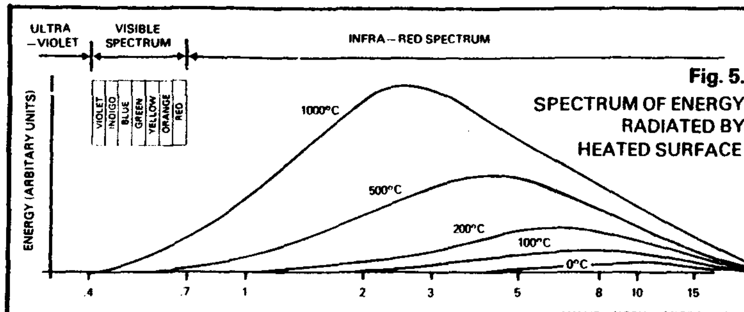


Fig. 5.
SPECTRUM OF ENERGY RADIATED BY HEATED SURFACE.

Figure 5 shows the division of the spectrum by wavelengths into Visible light, infra-red and ultra-violet. Radio waves (with a much longer wavelength) and X-Rays (with a much shorter wavelength) cannot be shown on this scale.

The curves show how the radiated energy from the surface of a heated solid or liquid is distributed. The total area under the curve is a measure of the total energy which is radiated, while each point on the curve represents the energy radiated at a specific wavelength.

It will be noted, for example, that when the temperature reaches 600°C a small amount of energy is radiated at the red end of the visible spectrum ("red-hot") while bodies at room temperature radiate most of their energy at wavelengths above 6 microns. It is also apparent that the total energy radiated is not proportional to the temperature of the surface. A body of 1000° radiates 250 times the energy it would radiate when at 50°C (Mathematically the energy is proportional to the fourth power of absolute temperature $E \times T^4$).

In order to measure temperatures as low as 0°C it is clearly necessary for the non-contact thermometer to respond to long wavelength variations. The spectral response of INFRATRACE 1000 satisfies this requirement. For measuring high temperatures INFRATRACE 2000 senses short wavelength radiation. In both cases the restricted spectral range of the instruments provides significant advantages in temperature measurement which will now be explained.

A.3. Selective Filters

In conventional photography coloured filters are commonly used to enhance or eliminate the camera's response to specific colours in the visible spectrum. Similarly the accurate measurement of temperature is simplified in INFRATRACE by embodying in its design filters which substantially eliminate the major sources of error often encountered in infra-red measurements.

Figure 6 represents the effect of the earth's atmosphere on infra-red radiation. As it passes from the heated surface to the instrument, some of the energy is absorbed, principally in the molecules of water vapour and carbon dioxide which are always present. (Often present in abundance in many industrial atmospheres.) It may be seen that more than half the energy emitted by the surface has been absorbed while travelling over 300 metres. The effect over just a few metres is significant when measuring accurate temperatures, and the attenuation will vary from day to day and from place to place according to the concentrations of water vapour and carbon dioxide which are present.

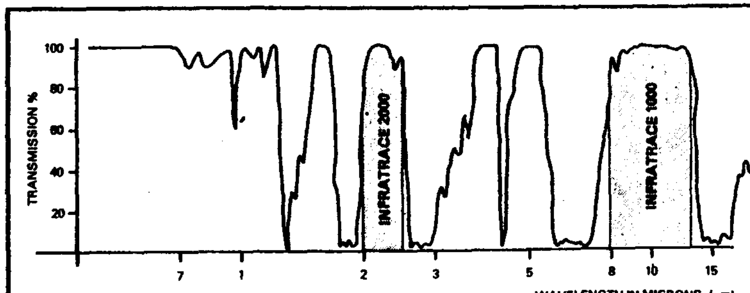


Fig. 6. ATMOSPHERIC TRANSMISSION OVER 300 METRE PATH. SHADED AREAS REPRESENT SPECTRAL RANGE OF INSTRUMENTS.

It will be noted that the spectral response of each INFRATRACE is matched to an "atmospheric window", that is a band of wavelengths over which the atmosphere transmits energy with negligible attenuation — even over long distances.

This means that INFRATRACE will measure temperature accurately at any distance, unlike the Total Radiation Pyrometer which responds to all wavelengths.

When measuring low temperatures (below 200°C) using INFRATRACE 1000 an even more significant source of error is eliminated. When the measured surface is slightly reflective, the instrument may receive energy reflected at the surface, which originates at an extraneous high temperature heat source such as the sun. It would not, of course, be affected by such a heat source at a similar temperature to that of the measured surface. However, as illustrated in figure 4, the energy from such high temperature sources is predominantly of short wavelength and is rejected by the selective filters in INFRATRACE 1000.

A.4. Measurements on or through Glass

The different spectral responses of INFRATRACE 1000 and INFRATRACE 2000 are especially significant when measuring glass temperature, or when measuring a surface behind a glass window. Ordinary soda-lime glass transmits energy of a wavelength less than 2.5 microns and absorbs energy of longer wavelengths.

Thus when measuring the temperature of a glass plate of minimum thickness 0.5mm, INFRATRACE 1000 may be used (with an emissivity setting of 0.95). A heat source behind the glass will have minimal influence on the reading, since its radiated energy will be absorbed by the glass.

Conversely, when a high temperature furnace is viewed through a glass window, the spectral response of INFRATRACE 2000 corresponds with the transmission through the glass so that the internal furnace temperature may be measured. In practice, some absorption will occur in the glass window, but this may be compensated using the Emissivity Control on INFRATRACE 2000, to allow for the transmission coefficients tabulated below.

Soda Lime Glass	
Plate thickness mm	Approximate Transmission
0.5	0.90
1	0.82
2	0.67
3	0.55
5	0.37
10	0.16

e.g. A measurement of fire brick temperature through 3mm plate glass.
From the table:

Glass transmission = 0.55
Emissivity of Fire Brick = 0.85
Multiply the two factors 0.55 × 0.85 = 0.47

Set emissivity control to 0.47 and measure in the normal way.

TYPICAL EMISSIVITIES — METALS

Surface	Emissivity
Iron and Steel	
Cast Iron, polished	0.2
Cast Iron, turned at 100°C	0.45
Cast Iron, turned at 1000°C	0.6 to 0.7
Steel, ground sheet	0.6
Mild steel	0.3 to 0.5
Steel plate, Oxidised	0.9
Iron plate rusted	0.7 to 0.85
Cast iron (rough) rusted	0.95
Rough ingot iron	0.9
Molten cast iron	0.3
Molten mild steel	0.3 to 0.4
Stainless steels polished	0.1
Stainless steels, various	0.2 to 0.6
Aluminium	
Polished Aluminium	0.1 *
Aluminium, heavily oxidised	0.25
Aluminium oxide at 260°C	0.6
Aluminium oxide at 800°C	0.3
Aluminium Alloys, various	0.1 to 0.25
Brass	
Brass, polished	0.1 *
Brass, roughened surface	0.2
Brass, oxidised	0.6
Copper	
Copper, polished	0.05 *
Copper plate, oxidised	0.8
Molten copper	0.15
Lead	
Lead, pure	0.1 *
Lead, oxidised at 25°C	0.3
Lead, oxidised, heated to 200°	0.6
Nickel and its alloys	
Nickel, pure	0.1 *
Nickel plate, oxidised	0.4 to 0.5
Nichrome	0.7
Nichrome, oxidised	0.95
Various	
Zinc, oxidised	0.1 *
Galvanised iron	0.3
Tin plated steel	0.1 *
Gold, polished	0.1 *
Silver, polished	0.1 *
Chromium, polished	0.1 *

TYPICAL EMISSIVITIES — NON-METALS

Surface	Emissivity
Refractory & Building Materials	
Red brick, rough	0.75 to 0.9
Fire clay	0.75
Asbestos	0.95
Concrete	0.7
Marble	0.9
Carborundum	0.85
Plaster	0.9
Alumina, fine grain	0.25
Alumina, coarse grain	0.45
Silica, fine grain	0.4
Silica, coarse grain	0.55
Zirconium silicate up to 500°C	0.85
Zirconium silicate at 850°C	0.6
Quartz, rough	0.9
Carbon, graphite	0.75
Carbon, soot	0.95
Glass (using INFRATRACE 1000)	0.95
Glass (using INFRATRACE 2000)	See text
Timber (various)	0.8 to 0.9
Miscellaneous	
Enamel (any colour)	0.9
Oil paint (any colour)	0.95
Lacquer	0.9
Matt black paint	0.95 to 0.98
Aluminium lacquer	0.5
Water	0.98
Rubber, smooth	0.9
Rubber, rough	0.98
Plastics, various (solid)	0.8 to 0.95
Plastic films (.05mm thick)	0.5 to 0.95
Polythene film (.03mm thick)	0.2 to 0.3
Paper & cardboard	0.9
Silicone polish (.03mm thick)	0.7

*For highly reflective materials, a film of black must be applied (see text).

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