HEAT AND MASS TRANSFER BY CONVECTION THROUGH LARGE RECTANGULAR OPENINGS IN VERTICAL PARTITIONS.

by

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SUMMARY

The primary objective of this study was to define the variables that influenced the isolation efficiency of areas such as isolation rooms, treatment rooms and operating theatres, mainly from the point of view of air movement through doorways due to convection. Areas where doors were constantly left open, such as cold storage rooms, were also considered.

The movement of air through an open doorway, with and without the influence of temperature was determined and the amount of air required to prevent this movement was also found. Original theory for combined natural convection and forced air flow across a rectangular opening in a vertical partition has been postulated and generalised to include both heat and mass transfer. Experiments for natural convection, and combined natural convection and forced air flow, were carried out with openings 2.05m high and from 0.10m to 1.40m wide. Temperature differentials were in the order of 0 to 12^{o}C and the supply and extract volumes in the range 0 to 0.30m^3/s. Natural convection results are quoted in the range $10^8 < Gr < 10^{11}$ while the combined natural convection and forced air flow results for the Nusselt number are expressed as a function of a dimensionless group which was found to include both Reynolds and Grashof numbers.

The isolation effectiveness of doors when they are shut or slightly open was also considered, and to extend the above work to include practical situations such as hospitals where doors are opened and closed, tests were carried out to establish the actual temperature differences and the door opening habits in existing hospital areas. These results were then utilised in tests carried out to determine how the efficiency of a room was effected by a door (both sliding and swing) being opened and closed and a person passing through it. Theory and experimental results are presented.

Finally, the full implications of the results obtained from all the tests are discussed and design recommendations are presented which may be used to achieve more efficient systems.

Several papers relating to the work of this thesis have been published by the author. These additional papers are submitted under separate cover.

This study was initiated by Mr. W. Whyte of the B.S.R.U. who also advised on the general running of the project. Theory, experimental work and analysis of the results were carried out by the author. The field tests to establish temperature differences and door opening habits were carried out by a final year student at the University of Strathclyde (Ward, 1970).

NOMENCLATURE

C	coefficient of discharge
C_T	coefficient of temperature
C_V	ceofficient of fictitious velocity
c_p	specific heat of fluid
c_1, c_2	concentration eg. gas or particles.
D	diffusion coefficient of mass transfer
D_h	hydraulic diameter of doorway = 2WH/(W+H)
G_a	rate of mass flow of air supplied to room
G_b	rate of mass flow of the air entering the room through the door.
g	acceleration due to gravity
H	height of opening
H_t	total, sensible or latent heat gain to the room, at any time t.
h	heat-transfer coefficient
h_m	mass-transfer coefficient
h_a	total, sensible or latent heat content of air supplied to the room.
h_b	total, sensible or latent heat content of air flowing into the room through the door.
k	thermal conductivity of fluid.
M	mass of air contained in the room.
$\dot{m}$	mass transfer rate through opening
P_1, P_2	pressures in rooms 1 and 2.
P_o	absolute pressure at the level of the neutral zone in the opening.
P_T, P_X	pressure due to temperature differential and excess supply ventilation pressure.
Q	volumetric fluid flow rate i.e. inflow air volume to room due to temperature differential between room and outside.
Q_L	leakage transfer volume into an area which is under positive pressure.
Q_e	extract air volume from room

Q_s	supply air volume to room.
Q_r	volume of room
S	total, sensible or latent heat content, depending on the context, of air within the room at any time.
S_0	total sensible or latent heat content of air initially in the room.
s	small increment of heat content
T_1, T_2	temperatures in rooms 1 and 2
t	time
t	thickness of partition
V	velocity
W	width of opening
α	thermal diffusivity
β	coefficient of thermal expansion
η	efficiency
μ	dynamic viscosity
ν	kinematic viscosity
ρ	fluid density

Dimensionless Groups

Fr_Δ	Densimetric Froude Number, $V/\left[gH(\Delta\rho/\bar{\rho})\right]^{1/2}$
Gr	Grashof Number based on densimetric differences, $g\,\Delta\rho\,H^3/\bar{\rho}\,\nu^2$
Nu	Nusselt number, hH/K
Pr	Prandtl number, $C_p\mu/k$
Re	Reynolds number, $\bar{\rho}V_b D_h/\mu$
Sc	Schmidt number, ν/D
Sh	Sherwood number, $h_m H/D$
St	Stanton number, $h/V\bar{\rho}C_p$
Sw	$\frac{Re^3}{Gr}\cdot\frac{H^3}{D_h{}^3} = \mu\,V_b{}^3/\nu^2 g\,\Delta\rho$

1. INTRODUCTION

The laws of heat and mass transfer find application in many fields of engineering. Mechanical, chemical and process engineering, and manufacturing and metallurgical industries are examples. In addition, the civil and constructional engineer and environmental control engineer need considerable knowledge of the subject. Buildings and factories must be economically heated and insulated, and air conditioning is increasingly necessary. Not only is ventilation and air conditioning used to achieve comfort standards but it is also used in many cases to isolate certain areas with respect to heat or mass transfer.

Heat transfer processes are described by equations which relate the energy to be transferred in unit time to the physical area involved. Other factors entering the equations are the temperatures or the temperature gradient, and some co-efficient which depends on various physical properties of the system and on the particular mechanism of heat transfer involved. The three basic mechanisms of heat transfer are convection, conduction and radiation, and may occur separately, or simultaneously. The subject matter of this thesis is solely based on the individual mechanism of convection.

Convection is the name given to the gross motion of the fluid itself, so that fresh fluid is continually available for heating or cooling. Apart from the bulk movement of the fluid, there is generally a smaller motion of eddies which further assists in distributing heat energy. Convection heat transfer is sub-divided into two different kinds, natural and forced. Heat transfer by natural convection occurs between a solid and a fluid or fluid and fluid,undisturbed by other effects when there is a temperature difference between the two. It is not often that a fluid can be regarded as entirely at rest, so frequently there is a small amount of forced convection as well. But true forced convection requires a major applied motion of the fluid in relation to the source or sink of heat, so that natural convection effects are negligible. An important aspect of natural convection is that the fluid motion which does occur is due entirely to natural buoyancy forces arising from a changing density of the fluid in the vicinity of the surface. Within the realms of both natural and forced convection there are two sub-divisions of laminar and turbulent flow convection.

It is thus evident that many factors enter into heat convection, including the shape and magnitude of the solid-fluid or fluid-fluid boundary, characteristics of the fluid flow, such as the magnitude of turbulent eddies, and the conductivity of the fluid itself. Because of these complexities many convection problems are not amenable to mathematical solutions, and recourse is made to techniques of dimensional analysis and experiment. Thus many empirical dimensionless relationships are now available in the literature to enable the engineer to design heat transfer apparatus, whether it be an industrial heat exchanger or the prevention of heat and mass transfer across an opening as in this case.

This thesis describes research that was carried out to define the variables that influenced the isolation efficiency of areas such as isolation rooms, treatment rooms, and operating theatres, mainly from the point of view of air movement through doorways due to convection. Areas where doors were constantly left open, such as cold storage rooms, were also considered. Section 2 presents an historical background of the subject, where previous research and theoretical considerations are discussed. The fundamental theory of natural and forced convection is dealt with in detail in section 3.

The object of section 4 is to consider the movement of air through an open doorway with and without the influence of temperature, and to determine the amount of air required to prevent this movement. Generalised theory for heat and mass transfer across the opening is quoted, along with actual results from experimental tests. This information is primarily applicable to areas in which doors are left open constantly, i.e. cold storage rooms, etc. but may also be used for all doors or openings in general.

The isolation effectiveness of doors when they are shut or slightly open was also considered, (section 5), the primary reason for conducting these tests being to obtain figures for the area of crack round a shut door which could result in complete isolation under different air supply volumes and temperature differentials across the doorway. Both single door and multiple door areas are discussed.

To extend the above work to include practical situations such as hospitals where doors are opened and closed, section 7 deals with the question; how much is the efficiency of a room against bacterial contamination effected by a door (both sliding and swing) being opened and closed and a person passing through it. Theory and experimental results are presented. The above tests were however dependent on either the temperature difference across the door or the number of times the door is used or they are dependent on both factors. It was therefore necessary to relate the results to be obtained in section 7 to the actual hospital situation in order that their full relevance could be obtained. Towards this end, tests were carried out to establish the actual temperature differences and the door opening habits in existing hospital areas (section 6). These results were then utilised in the tests carried out in section 7.

The final section of the thesis, section 8, discusses the full implications of the results obtained from all the tests and presents design recommendations which may be utilised to achieve more efficient designs and systems in all the fields concerned.

2. HISTORICAL BACKGROUND

The problem of transfer of air through doorways or hatches due to natural convection has been brought to the fore in the last 20 years by the evolution of air-conditioning and refrigeration engineering practice. There exists two major fields of study concerned with this problem. These are the medical aspect of sterile areas within hospitals where airborne cross infection may be significant and the engineering and economic aspects of cold storage room design.

2.1 (a) Sterile areas within hospitals

In contrast with the contact and endogenous routes of infection, the role of the air-borne route is very difficult to define. The areas in hospitals which have been considered to possibly gain some benefit from a reduction in the number of bacteria in the air are those in which airborne infection has been considered to play some part in the spread of infection. These areas include operating theatres, treatment rooms, isolation rooms and even wards. The principles laid down by earlier workers who studied hospital infections had a major influence on current practice in the design of operating theatres, and these principles have recently been applied to the design of other parts of the hospital.

For many years, surgical operating theatres have been provided with some form of ventilation equipment. This was usually designed with the sole object of providing comfortable working conditions. In 1946 Bourdillon and Colebrook drew attention to another important function of ventilation in burns dressing-rooms and operating theatres. They showed that serious sepsis of burns and wounds could be caused by bacterial contamination from the air and that well-designed ventilation equipment could play a large part in preventing this.

Air borne bacteria in an operating, treatment or isolation room can come from sources inside and outside the room. Inside the room contaminated particles may be shed from the coverings of septic wounds, from blankets, from the respiratory tracts, skin and clothing of the occupants of the room, and may be raised from the floor as the nurses move about their duties. From the outside sources, air-conditioned supply air to the room may be contaminated due to bacterial growth on certain types of humidification equipment. Also, air in hospital wards and corridors is contaminated with pathogenic bacteria and air may be transferred from the dirty area into the clean area through openings such as doors. This may occur by an excessive pressurisation of the dirty area or by natural convection due to a temperature differential between the two areas.

To reduce these risks, Bourdillon and Colebrook recommended ventilation of burns dressing-rooms by forcing a supply of filtered air into the room instead of sucking air out, which was the normal practice at that time. This positive pressure system reduces the amount of contaminated air flowing into the dressing-room from other parts of the hospital and helps to carry away organisms that are liberated inside the room. The importance of the control of airborne bacteria can be shown by the results of the following studies. Lowbury (1954) reported that 12 out of 69 (17.4%) patients with burns dressed in filtered air, and 25 out of 71 (35.2%) control patients became infected. Shooter et al (1956) showed that a reduction of sepsis rate from 9% to 1% occurred after changing the ventilation system in their theatre from a basic extract system to a positive pressure system. Blowers and Crew (1960) indicated that the frequency of wound sepsis fell from about 11% to 5% of all operations by improved ventilation.

b) Previous research and publications

Various papers on the subject of convective transfer through doorways of bacteria such as Staph. aureus, an important group of bacteria which causes infection, have been published in the last ten years. Wolf, Harris and Hall-(1961), carrying out bacteriological tests on open operating doors, recorded that the air entering the room through the doorway carried 12 colonies per $ft.^3$ of air when the room was not in use. Blowers and Crew (1960) suggested that to prevent ingress of contaminated air, the operating room should be pressurised by a flow of filtered air, this being in the region of $0.57m^3/s$ (1200 c.f.m.). They also suggested that to obtain this, mechanical exhaust fans should be replaced by pressure relief dampers and that other rooms of the suite should be ventilated in a similar manner, with pressure gradients great enough to cause air flow from clean to dirty zones. Ma (1965) put forward the view that if an operating theatre air-conditioning system pressurises the theatre and introduces ample "clean" air (20 air changes/hour), it has done more than 90% of what is possible in the way of airborne bacterial control with an air-conditioning system. The actual movement of air within the suite is also critical and this has been discussed in papers by Blowers, Ma and Heckert (1968).

The effect of opening and closing of doors upon the contamination level within a room has also been discussed by various workers. Results obtained by Beck (1966) showed an increase in contamination of a room when a swing door was opened and closed, even when positive pressure was used. He therefore advocated the use of sliding doors along with positive ventilation. However, results of Baird and Whyte (1968) showed no difference between swing doors and sliding doors so far as isolation efficiency was concerned. The number of times the door was opened and closed was also stated as being of no significance. The total time that the door was left open was however found to be significant. Whether positive pressure can fully isolate a room and how much air would be required has been discussed in several papers. A large variety of answers have emerged, probably due to the fact that only isolated tests were carried out and not a full research program such as this. Baird and Whyte put forward prediction equations in their study, but omitted the effect of temperature differential. Their equations were also only applicable to a specific air supply rate to the room. A general equation involving all parameters concerned was therefore required, to express the specific "mechanism" involved whether positive or balanced.

2.2 (a) Cold Storage rooms and public buildings

The problem concerned with cold storage rooms is that of heat and mass transfer through the access doorway resulting in greater running costs. Due to there being an increase in the amount of short term storage and consequently faster turnover of goods, the "structure" of the doorway has changed radically. The "lock" doors formerly used to shield the rooms against the penetration of air from outside had to be removed, as they hindered the traffic too much.

In these conditions the doors of cold rooms and deep freeze rooms must inevitably stand open several hours a day, and high cold losses must be allowed for due to the air exchange between the cold room and the surroundings. With large temperature differentials in the region of 30^{o}C between the two areas, vast quantities of air will be transferred. Increased door measurements in the interest of mechanical transport have also meant increased cold losses, as have the lower temperatures in the deep freeze stores.

To counteract these losses, air screens and mechanically operated doors are used, yet they still form a large part of the heat balance of many cold storage depots, whose actual amount should be a matter of precise knowledge both for the planning engineer and the manager of cold stores. There is also the problem in public buildings such as shops, supermarkets and restaurants of convective air currents causing unpleasant draughts and loss of heat at doorways where there is normally a heavy concentration of pedestrian traffic.

Convective transfer may be combatted by air curtains or revolving doors, but these also have their problems. There is a difference in air curtain design, depending upon where the air curtain is to be used. In supermarkets, department stores and other public buildings, the air speed must be low to avoid objectionably high velocities on pedestrian traffic, so the air curtains must be two to three feet thick to achieve adequate holding power. In non-public doorways such as those in factories or warehouses, the traffic is mostly vehicular rather than pedestrian. Consequently, the thickness of the jet can be reduced to several inches, and the air curtain units can be made more compact. These units are much simpler than the low velocity type and can often be mounted above a conventional doorway, with no special construction required.

The major criterion for the effectiveness of an air curtain is the rate of heat transfer through the air curtain compared with that of the same opening with no air curtain. If the outlet velocity is very high, it is possible that an air curtain could increase the rate of heat transfer. On the other hand, if the outlet velocity is too low, the pressure forces created by the difference in air densities will cause the curtain to break contact with the floor and bend back towards the cold side, leaving the bottom portion of the opening unprotected. It is therefore apparent that at some outlet velocity the rate of heat transfer will be a minimum. Various workers such as Cadiergues (1956), Hetsroni and Hall (1964),Hayes and Stoecker (1969) have carried out such tests to determine the heat transfer characteristics of an air curtain.

The problem relating to revolving doors is that of door size, mechanically operated doors may be employed, revolving the doors at walking pace and large enough for pedestrian and luggage to transfer easily. Whether revolving doors would be able to cope with heavy traffic is debatable, and the use of air curtains with mechanically operated sliding doors is probably a more suitable solution.

b) Previous research and theoretical considerations

Up to 1960 studies of natural convection were primarily concerned with problems of heat transfer involving vertical and horizontal plates and bodies of varying shape. These studies and their application to practical situations, ranging from heating equipment to the cooling of turbine blades, were reviewed by Schmidt (1961) who mentioned a type of natural convection that up till then had received very little attention. This was the situation occurring at openings in partitions, for which Schmidt reported an optical investigation of the transient mixing of two fluids of different densities (carbon dioxide and air) separated by an opening in a vertical partition.

Apart from the transient case, the two basic aspects of natural convection through openings are those of steady conditions with vertical and horizontal partitions. Emswiler in 1926 treated the case of multiple openings in a wall and obtained an expression for the rate of flow of air in terms of temperature difference and Bernoulli's equation for ideal flow. He did not consider the case of a single opening nor did he treat the heat and mass transfer aspects of the problem which can be generalised for all fluids.

No direct measurements had been made to substantiate and extend the theory and this may partly be explained by measurement difficulties and by the fact that opening sizes of practical importance were rather large to be investigated in a laboratory: However, in the late 1950's, a large test unit, apparently the first of its kind, was built at the National Research Council of Canada to allow direct measurement of the heat transmittance of 8 ft. square walls. This apparatus was ideally suited to the problem of convection transfer. Brown and Solvason (1962) generalised the simple basic theory of natural convection across openings in vertical partitions to include both heat and mass transfer. Their experiments, using the apparatus stated above, were carried out with openings from 3 to 12 in. high and with air as the convecting fluid. The test results for Nusselt number were expressed as a function of the Grashof number based on opening height in the range of $10^6 < Gr < 10^8$ and due to the high thermal resistance of the partition would also be expected to be directly applicable to mass transfer. Due to the fact that the openings were small, ratio of wall thickness to opening height was taken into consideration in the analyses, this ratio tending to be large (0.19 - 0.75) and therefore of significance.

A theoretical approach to the problem was made by Graf (1964), his theory being similar to Brown and Solvason but omitting a discharge coefficient of approx. 0.65 which they had used. In his paper he also attempted to calculate the amount of air that would be required to 'screen' a room by means of high pressure, i.e. to supply excess pressure to the room which would be high enough to compensate for the thermal pressure. Since no discharge coefficient had been taken into account, this volume of air supplied to the room was greater than actually required for complete screening.

Tamm (1966), also put forward similar theory, i.e. did not consider a discharge coefficient, but differed slightly from Graf in that he used the density of the cold air as a reference. Both Solvason <u>et al</u>, and Graf used the mean density of the hot and cold air, $\frac{(\rho_c + \rho_H)}{2}$, as a reference. Tamm also makes reference to tests carried out by Franke, who measured the air velocities at various heights in a cold room doorway. The mean air volume flowing in or out was ascertained by planimetry of the surface contained by the velocity curve and the co-ordinate axis of the doorway. This showed the test volume to be approx. 10% less than the theoretical one. A more accurate result would be obtained using the mean density as applied by Solvason <u>et al</u> and Graf. This result, however, does not mean that a coefficient of discharge should not be included, as the value of this

Table 2.1.: Comparison of variables as studied in previous research.

Source	Convection	Area m^2	Height m	Range of ΔT^oC
Brown and Solvason (1962)	Natural Natural plus Forced Air Flow	0.00581 to 0.09190	0.0762 to 0.3048	8 - 47
Graf (1964)	Natural Natural plus Forced Air Flow	-	-	Theory
Tamm (1966)	Natural	-	-	Theory
Fritzsche & Lilienblumm (1968)	Natural	4.5	2.5	12 - 41.5
Shaw (1971)	Natural Natural plus Forced Air Flow	0.205 to 1.845	2.05	0 - 12

coefficient may be shown to be in the order of unity at approx. 40 - 50^{o}C temperature differential.

The existence of such a coefficient was made evident by Fritzsche and Lilienblumm (1968) when they showed that the coefficient did not remain constant but varied with temperature differential, i.e. $C_T = \emptyset \Delta T$. This fact was determined experimentally, using a cold storage room doorway, 1.8m wide x 2.5m high. Air velocities in the doorway were determined by means of a grid of 108 wing wheel anemometers. By using so many measuring points, it was possible to obtain a good average value of velocity and hence air transfer volumes inwards and outwards.

A three dimensional diagram of the velocity distribution could also be constructed giving a good visual representation of the air movement. From the results which they obtained, Fritzsche and Lilienblumm came to the conclusion that a discharge coefficient had to be added to the basic equation of Tamm, this coefficient being a function of the temperature differential between the two areas.

There are a number of variables such as type of convection, area of opening, height of opening, temperature differential and condition of opening, the combinations of which may be considered in any particular analysis. Table 2.1. compares these pertinent variables as studied by each source, and it may be seen that no previous research has been carried out with small temperature differentials. Also there were no results for effect of excess pressure within a room acting on the natural convection. The forced air flow used by Brown and Solvason was in fact a horizontal velocity parallel to the opening surface and acted as a type of air curtain. This thesis does however consider these variables and as a result considerably widens the knowledge of convection through openings in vertical partitions.

3. FUNDAMENTAL THEORY OF NATURAL AND FORCED CONVECTION (after Schmidt)

If heat flows in more than one direction, the three components of the heat flux have to be added and the following differential equation is obtained:

$$\frac{\partial \theta}{\partial t} = \propto \left(\frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2} + \frac{\partial^2 \theta}{\partial \gamma^2} \right) \qquad (3.1)$$

Let us divide all variables in the differential equation (3.1) by certain fixed reference values, each of which is selected separately for each variable. We divide all lengths by L, all times by t_0, and all temperatures by Θ. Eqn. (3.1) then assumes the following dimensionless form:

$$\frac{\partial(\theta/\Theta)}{\partial(t/t_o)} = \frac{\propto t_o}{L^2}\left[\frac{\partial^2(\theta/\Theta)}{\partial(x/L)^2} + \frac{\partial^2(\theta/\Theta)}{\partial(y/L)^2} + \frac{\partial^2(\theta/\Theta)}{\partial(\gamma/L)^2} \right] \qquad (3.1a.)$$

It will be easily verified that the factor $\propto t_o/L^2$ is also dimensionless magnitude.

Thus a solution found for a specified set of boundary conditions is also valid for all similar cases, provided the reference values of length and time are correctly selected.

The principle of similarity in heat transfer

The transfer of heat between a solid surface and a liquid or gaseous medium is a problem of hydrodynamics, except that heat flows are superimposed on the mechanical phenomena of motion. We must therefore combine the fundamental equations of hydrodynamics with eqn. (3.1) for heat transfer.

If we restrict ourselves to the study of steady flow problems, and if we stipulate velocities which are small with respect to the velocity of sound and consequently allow only small density variation, the following equations are valid:

1. The equation of continuity expresses that the algebraic sum of all quantities of fluid crossing the surfaces of a volume ele-

ment must be equal to zero and may be written down as

$$\frac{\partial}{\partial x}(\rho V_x) + \frac{\partial}{\partial y}(\rho V_y) + \frac{\partial}{\partial \eta}(\rho V_\eta) = 0 \qquad (3.2)$$

where Vx, Vy and V_η are the three components of the velocity vector V and ρ is the density.

2. The equations of motion of viscous fluids apply the fundamental laws of dynamics to fluids and assume that the acceleration force is the sum of lift, pressure gradient, and viscosity forces. The lift per unit of volume is $-\hat{g}\rho\beta\theta$, where $\hat{g}$ denotes the vector of the acceleration due to gravity with the absolute 'normal' value

$$g_o = 9{\cdot}80665\ m/sec^2 = 32{\cdot}1719\ ft/sec^2$$

and ρ is the density of the fluid before heating; β is the coefficient of thermal expansion, and θ the excess of temperature of the heated parts of the fluid over the parts which remain cold. It is generally agreed to direct the x-axis vertically upwards so that the lift then has only one component, which is directed in opposition to the vector of gravitational acceleration and therefore receives a negative sign. The pressure gradient on the other hand has, generally speaking, three components $\partial P/\partial x$, $\partial P/\partial y$ and $\partial P/\partial \eta$. The viscosity forces have also three components and depend on the first and second derivatives of velocity $\dot{V}$, which we resolve into its three components, V_x, V_y and V_η. The conditions for the equilibrium of forces in the three co-ordinate directions lead to the following equations of motion:

$$\rho\left(V_x\frac{\partial V_x}{\partial x} + V_y\frac{\partial V_x}{\partial y} + V_\eta\frac{\partial V_x}{\partial \eta}\right) = -g\rho\beta\theta - \frac{\partial P}{\partial x} +$$

$$+\mu\left[\frac{\partial^2 V_x}{\partial x^2} + \frac{\partial^2 V_x}{\partial y^2} + \frac{\partial^2 V_x}{\partial \eta^2} + \frac{1}{3}\frac{\partial}{\partial x}\left(\frac{\partial V_x}{\partial x} + \frac{\partial V_y}{\partial y} + \frac{\partial V_\eta}{\partial \eta}\right)\right]$$

$$(3.3)$$

contd.

$$\rho\left(V_x\frac{\partial V_y}{\partial x} + V_y\frac{\partial V_y}{\partial y} + V_\eta\frac{\partial V_y}{\partial \eta}\right) = -\frac{\partial P}{\partial y} +$$

$$+\mu\left[\frac{\partial^2 V_y}{\partial x^2} + \frac{\partial^2 V_y}{\partial y^2} + \frac{\partial^2 V_y}{\partial \eta^2} + \frac{1}{3}\frac{\partial}{\partial y}\left(\frac{\partial V_x}{\partial x} + \frac{\partial V_y}{\partial y} + \frac{\partial V_\eta}{\partial \eta}\right)\right] \tag{3.3}$$

$$\rho\left(V_x\frac{\partial V_\eta}{\partial x} + V_y\frac{\partial V_\eta}{\partial y} + V_\eta\frac{\partial V_\eta}{\partial \eta}\right) = -\frac{\partial P}{\partial \eta} +$$

$$+\mu\left[\frac{\partial^2 V_\eta}{\partial x^2} + \frac{\partial^2 V_\eta}{\partial y^2} + \frac{\partial^2 V_\eta}{\partial \eta^2} + \frac{1}{3}\frac{\partial}{\partial \eta}\left(\frac{\partial V_x}{\partial x} + \frac{\partial V_y}{\partial y} + \frac{\partial V_\eta}{\partial \eta}\right)\right] \tag{3.3}$$

3. The equation of energy expresses that in steady conditions the quantity of energy flowing into an element of fluid is equal to the quantity of heat being conducted away from it. In connection with this we neglect the pressure energy against sensible heat and disregard the generation of heat through friction in the flow. With these simplifications, which are almost always admissible, the energy equation becomes

$$V_x\frac{\partial \theta}{\partial x} + V_y\frac{\partial \theta}{\partial y} + V_\eta\frac{\partial \theta}{\partial \eta} = \alpha\left(\frac{\partial^2\theta}{\partial x^2} + \frac{\partial^2\theta}{\partial y^2} + \frac{\partial^2\theta}{\partial \eta^2}\right) \tag{3.4}$$

where α is again the thermal diffusivity i.e. $\alpha = \frac{k}{C_p\rho}$

In vector symbols the above equations can be rewritten thus:

$$\mathrm{div}(\rho \mathbf{V}) = 0 \tag{3.2a}$$

$$\rho(\mathbf{V}\,\mathrm{grad})\mathbf{V} = \hat{g}\rho\beta\theta - \mathrm{grad}\,P + \mu(\Delta \mathbf{V} + \tfrac{1}{3}\,\mathrm{grad.div}\,\mathbf{V}) \tag{3.3a}$$

$$(\mathbf{V}\,\mathrm{grad})\theta = \alpha\Delta\theta \tag{3.4a}$$

In these three equations, each of which consists of three component equations, the independent variables are the three coordinates x, y, z, and the three magnitudes **V**, θ , and P are unknown, the vector **V** having three components. In general, the solution of the above system of equations is given by the following fields:

The velocity field $V_x = f_{V_x}(x, y, \eta)$

$V_y = f_{V_y}(x, y, \eta)$

$V_\eta = f_{V_\eta}(x, y, \eta)$

The temperature field $\theta = f_\theta \; (x, y, \eta)$

The pressure field $P = f_P \; (x, y, \eta)$

By eqn. (3.3) the pressure field is determined by the fields of velocity and temperature. The pressure is not an independent variable, because it can be eliminated from these equations, in that, for instance, the first is differentiated partially with respect to y, the second partially with respect to x, and then one subtracted from the other. In this way, in the place of the three eqns. (3.3) two new acceleration equations are obtained in which the pressure does not appear.

The functions f contain the following five constants of the differential equations:

The acceleration due to gravity	g
The density of the fluid at the initial conditions	ρ
The kinematic viscosity	$\nu = \mu/\rho$
The thermal diffusivity	$\alpha = k/C_P\rho$
The coefficient of thermal expansion	β

The coefficient of thermal expansion $\beta = (1/v)(\partial v/\partial T)_P$ can be calculated from the equation of state; for perfect gases we have $\beta = 1/T$. With not too great temperature differences these characteristic quantities of the fluid can be regarded as constants and they are usually taken for the mean temperature of the fluid.

In order to derive the conditions of similarity of the whole fields, given geometrical similarity of the boundaries and of the boundary conditions, we rewrite eqns. (3.3) introducing dimensionless variables, in the same way as we did with eqn. (3.1) in that we divide the components of velocity V_x, V_y, V_η the temperature difference θ

and the linear coordinates x, y, z by certain characteristic magnitudes V_o, Θ, and L. These characteristic magnitudes are, as a rule furnished by the boundary conditions, in that V_o and Θ, for example, have prescribed values at certain points of the boundary and L is a magnitude, which characterises the size of the body. In most cases Θ is the difference between the temperature of the wall θ_o and of the fluid θ_f at a great distance from it. We multiply eqn. (3.3) by $L/\rho V_o^2$, introduce the kinematic viscosity $\nu = \mu/\rho$, and restrict ourselves to small density variations, for which the derivatives of $\frac{\partial V_x}{\partial x} + \frac{\partial V_y}{\partial y} + \frac{\partial V_\eta}{\partial \eta}$ with respect to the linear coordinated can be neglected. We then obtain eqn. (3.3) in dimensionless form.

$$\frac{V_x}{V_o}\frac{\partial}{\partial(x/L)}\left(\frac{V_x}{V_o}\right) + \frac{V_y}{V_o}\frac{\partial}{\partial(y/L)}\left(\frac{V_x}{V_o}\right) + \frac{V_\eta}{V_o}\frac{\partial}{\partial(\eta/L)}\left(\frac{V_x}{V_o}\right) = -\frac{Lg\beta\theta}{V_o^2} - \frac{\partial}{\partial(x/L)}\left(\frac{P}{\rho V_o^2}\right) +$$
$$+ \frac{\nu}{V_o L}\left[\frac{\partial^2}{\{\partial(x/L)\}^2}\left(\frac{V_x}{V_o}\right) + \frac{\partial^2}{\{\partial(y/L)\}^2}\left(\frac{V_x}{V_o}\right) + \frac{\partial^2}{\{\partial(\eta/L)\}^2}\left(\frac{V_x}{V_o}\right)\right]$$

and two similar equations for V_y/V_o and V_η/V_o, which need not be written down. The pressure is here made dimensionless with the aid of the momentum flux ρV_o^2, which is equal to double the dynamic pressure. Rearranging and introducing, for the sake of simplicity, the following dimensionless parameters

$$\epsilon = \frac{x}{L},\ z = \frac{y}{L},\ \gamma = \frac{\eta}{L},\ \Omega_x = \frac{V_x}{V_o},\ \Omega_y = \frac{V_y}{V_o},\ \Omega_\eta = \frac{V_\eta}{V_o}$$
$$\pi = \frac{P}{\rho V^2},\ \text{and}\ \theta_\Theta = \frac{\theta}{\Theta}$$

we obtain

$$\frac{\partial^2\Omega_x}{\partial\epsilon^2} + \frac{\partial^2\Omega_y}{\partial z^2} + \frac{\partial^2\Omega_\eta}{\partial\gamma^2} = \frac{V_o L}{\nu}\left[\Omega_x\frac{\partial\Omega_x}{\partial\epsilon} + \Omega_y\frac{\partial\Omega_y}{\partial z} + \Omega_\eta\frac{\partial\Omega_\eta}{\partial\gamma} + \frac{\partial\pi}{\partial\epsilon}\right] + \frac{L^2 g\beta\Theta}{V_o\nu}\cdot\theta_\Theta$$

and two similar equations for Ω_y and Ω_η, in which, however, the lift term does not appear. From (3.4) we have in similar way:

$$\frac{\partial^2\theta_\Theta}{\partial\epsilon^2} + \frac{\partial^2\theta_\Theta}{\partial z^2} + \frac{\partial^2\theta_\Theta}{\partial\gamma^2} = \frac{V_o L}{\alpha}\left[\Omega_x\frac{\partial\theta_\Theta}{\partial\epsilon} + \Omega_y\frac{\partial\theta_\Theta}{\partial z} + \Omega_\eta\frac{\partial\theta_\Theta}{\partial\gamma}\right] \tag{3.4b}$$

In the equation of continuity ρ can be omitted owing to the restriction to small density variations, so that we obtain

$$\frac{\partial\Omega_x}{\partial\epsilon} + \frac{\partial\Omega_y}{\partial z} + \frac{\partial\Omega_\eta}{\partial\gamma} = 0 \tag{3.2b}$$

Apart from the variables, three dimensionless parameters occur in these differential equations, namely V_oL/ν, V_oL/α, and $L^2g\beta\theta/V_o\nu$.

If the fields are to be similar throughout their extensions, for similar boundaries and boundary conditions, the dimensionless variables must satisfy the same differential equations, that is, the three parameters of the differential equations must have identical values respectively, for all similar cases. The expression

$$Re = \frac{V_o L}{\nu} = \frac{\rho L V_o}{\mu} \tag{3.5}$$

is the already familiar Reynolds number, and the parameter

$$P = \frac{V_o L}{\alpha} \tag{3.6}$$

is called the Peclet number. Instead of the parameter $L^2 g \beta \Theta / V_o \nu$ it is customary to introduce the so-called Grashof number

$$Gr = \frac{L^3 g \beta \Theta}{\nu^2} = \frac{L^2 g \beta \Theta}{V_o \nu} \cdot Re \tag{3.7}$$

This is exactly equivalent, since if each of the three parameters Re, P, and $L^2 g \beta \Theta / V_o \nu$ has a fixed value, every function of these parameters must also have a fixed value, that is the Grashof number has also a fixed value. In general, in the place of the three numbers above three mutually independent functions of them may be used. In particular we shall make use of the Prandtl number

$$Pr = \frac{\nu}{\alpha} = \frac{c_p \mu}{k} = \frac{P}{Re} \tag{3.8}$$

which is preferable in comparison with the Peclet number because it contains only properties of the substance. These dimensionless magnitudes are called characteristic parameters.

In the general case, that is, for arbitrary values of the characteristic parameters, the fields of velocity and temperature depend on the characteristic parameters, apart from the linear coordinates, and we can write the solution in the form

$$\begin{aligned} \frac{\hat{V}}{V_o} &= f_{\hat{V}}\left(\frac{x}{L}, \frac{y}{L}, \frac{\eta}{L}, Re, Pr, Gr\right) \\ \frac{\theta}{\Theta} &= f_\theta\left(\frac{x}{L}, \frac{y}{L}, \frac{\eta}{L}, Re, Pr, Gr\right) \end{aligned} \tag{3.9}$$

where the first equation stands for three component equations.

Fields of similar boundaries can be represented by the same functions $f_{\hat{V}}$ and f_{θ} when the characteristic parameters have the same numerical values. In these cases the parameters $\hat{V}$ and θ differ only by constant factors, which have the same values at all points. All fields, which can be described by the same functions, are called similar.

The similarity of fields does not stipulate that all properties of the substances are respectively equal ; on the contrary, they can have widely differing values, provided that their combinations into characteristic parameters have the same values.

In practical applications it is seldom required to know the temperature and velocity fields with every detail. It is sufficient to know the coefficient of heat transfer on certain boundaries of the temperature field. In terms of the difference of the temperature of the wall and fluid at a greater distance, the heat flux passing through area A is $hA\Theta$. In the layer of fluid whose coefficient of conduction is k and which clings to the wall, heat is transferred only by conduction and the flux can also be expressed as $kA(\partial\theta/\partial n)_0$ where n is a coordinate measured on the perpendicular to the surface through which heat is transferred. Equating the two expressions and introducing dimensionless variables , we obtain by differentiation of the dimensionless temperature field with respect to a coordinate n, normal to the surface, through which heat is transferred, the following equation:

$$\frac{hL}{k} = \frac{\partial(\theta/\Theta)}{\partial(n/L)} = \frac{\partial}{\partial(n/L)} f_{\theta}\left(\frac{x}{L}, \frac{y}{L}, \frac{\eta}{L}, Re, Pr, Gr\right) \quad (3.10)$$

The dimensionless coefficient of heat transfer is also known as the Nusselt number.

$$Nu = \frac{hL}{k} \quad (3.11)$$

It can be interpreted as follows:

Imagine that the resistance to heat transfer is caused by a stationary layer clinging to the surface whose thickness is δ and which accommodates the whole temperature gradient. We then have

$$\frac{1}{h} = \frac{\delta}{k} \tag{3.12}$$

so that the film must have a thickness $\delta = k/h$. The magnitude

$$\frac{1}{Nu} = \frac{k}{hL}$$

is then the ratio of the thickness of the film to the characteristic length.

Instead of the Nusselt number it is sometimes convenient to use the Stanton number

$$St = \frac{Nu}{Re\,Pr} = \frac{h}{V\rho C_p} \tag{3.13}$$

As a rule the differences in the value of the coefficient of heat transfer at different places are disregarded and mean values are much used, which corresponds to an integration over the surface under consideration. The linear coordinates are thus cancelled, and for all problems with similar boundaries there remains only the relationship between the characteristic parameters.

$$\frac{hL}{k} = \phi\,(Re, Pr, Gr) \tag{3.14}$$

Similarity considerations cannot give any clues as to the form of the function ϕ. It must be found separately for each particular case, mostly by experiment.

The theory of similarity, however, sets a frame to those experiments in that it reduces the number of measured parameters to a minimum, and shows how results of one experiment can be extended to include a whole range of similar cases. Sometimes eqn. (3.9) is extended by the introduction of further ratios, so that it can include cases which are not necessarily geometrically similar. For example, when considering the flow of a fluid through a rectangular opening in a vertical partition due to natural convection, the thickness t is used as a characteristic length. Various opening heights are taken into account in that the ratio of thickness to height t/H is introduced as a further characteristic parameter, so that

$$\frac{hH}{k} = \phi\,(Re, Pr, Gr, t/H) \tag{3.15}$$

In actual fact the dependence of the coefficient of heat transfer on all three characteristic parameters of eqn. (3.9) is only seldom required, since in each particular problem it is sufficient to take into account its variation with one or two of them.

In numerous applications the state of motion created by external forces is so strong that it is not influenced by the small thermal lift forces. The last term of the differential equation (3.3b) falls out, the velocity field becomes independent of the temperature field, and we speak of forced convection. The dependence on the Grashof number vanishes and eqn. (3.15) simplifies to

$$\frac{hL}{k} = \phi\,(Re, Pr) \tag{3.16}$$

In another group of problems the motion is caused only by lift forces. We then speak of natural convection. The Reynolds number ceases to influence the result and we have

$$\frac{hL}{k} = \phi\,(Gr, Pr) \tag{3.17}$$

If the state of motion created by external forces can in fact be influenced by the thermal lift forces, then all three characteristic parameters of equation (3.9) must be included, so that

$$\frac{hL}{k} = \phi\,(Re, Gr, Pr) \tag{3.18}$$

4. THE INFLUENCE OF TEMPERATURE AND THE CONTROL OF AIR MOVEMENT THROUGH DOORWAYS BY FORCED AIRFLOW

4.1 Theory

4.1.1 The theory of the volumetric exchange of air due to natural convection through a rectangular opening in a vertical partition.

Consider a large sealed enclosure consisting of room 1 and 2 as shown in Figure 4.1. The rooms are separated by a vertical partition with a rectangular opening of height H and width W. The temperature in the rooms are T_1 and T_2 respectively. Since the enclosure is sealed, there is no net flow of air across the opening. The absolute pressure P_0 at the elevation of the centre line of the opening is everywhere equal.

In room 1, the pressure P, at level Z below the centre line will be

$$P_1 = P_0 + \rho_1 g Z \tag{4.1}$$

then the pressure at the same level in room 2 will be

$$P_2 = P_0 + \rho_2 g Z \tag{4.2}$$

g being the acceleration due to gravity and ρ_1 and ρ_2 being the densities of air in room 1 and 2 respectively.

The pressure difference in these two rooms at the same level is

$$P_2 - P_1 = (\rho_2 - \rho_1) g Z \tag{4.3}$$

This pressure difference can be expressed as the height (h_a) of a column of air where

$$h_a = \frac{\rho_2 - \rho_1}{\bar{\rho}} Z = \frac{\Delta\rho}{\bar{\rho}} Z$$

where $\bar{\rho}$ is the mean density

$$\bar{\rho} = \frac{\rho_1 + \rho_2}{2} \tag{4.4}$$

As there is only limited information available for the relation between pressure head and velocity V for rectangular orifices at low flow rates, the flow will, in this case, be assumed to be ideal (i.e.

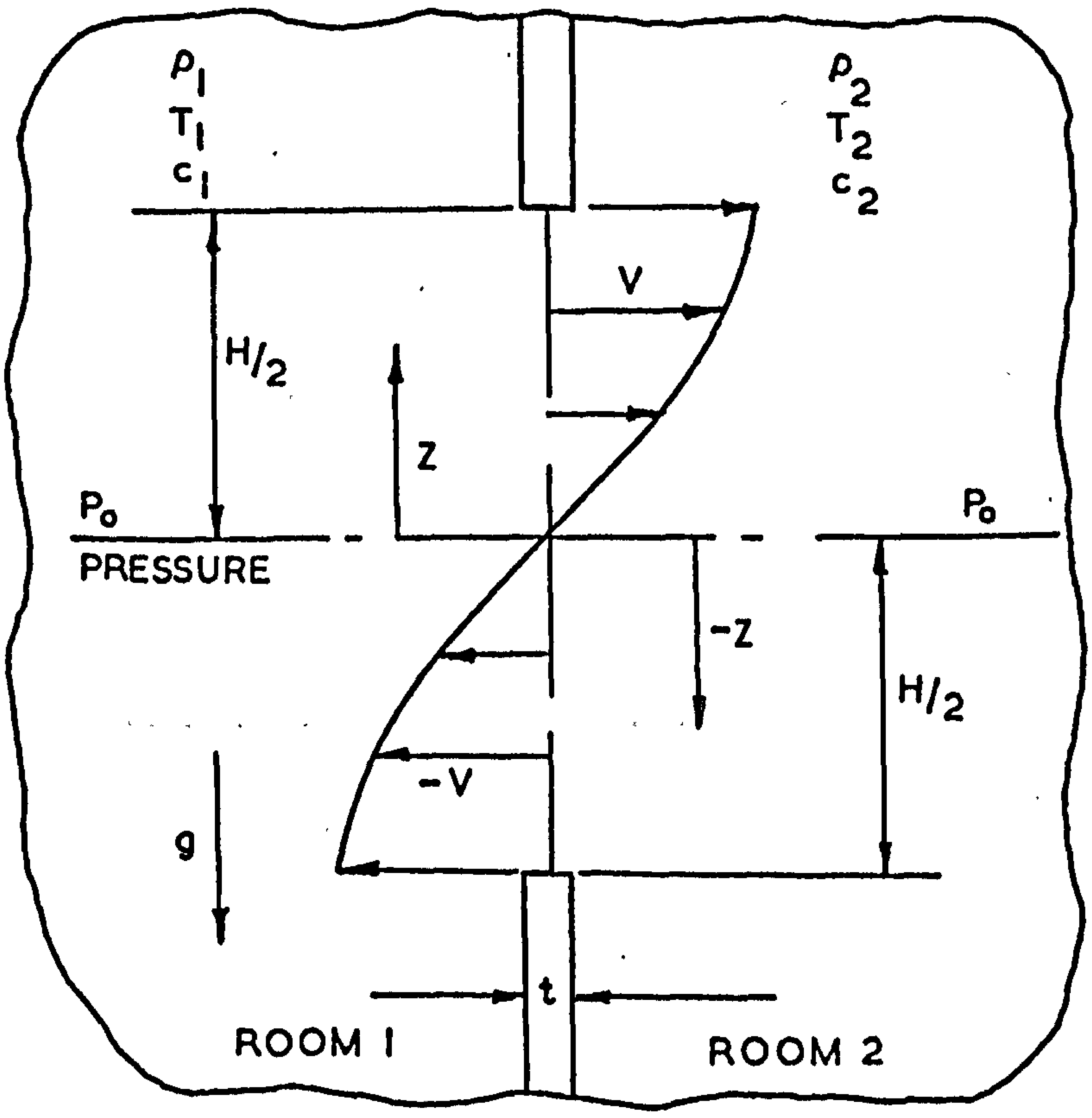


FIG.4.1. SCHEMATIC REPRESENTATION OF NATURAL CONVECTION ACROSS AN OPENING IN A VERTICAL PARTITION.

frictionless). For ideal flow the Bernoulli equation can be assumed, i.e.

$$V = (2gh_a)^{1/2} = \left[2g\frac{\Delta\rho}{\bar{\rho}}z\right]^{1/2} \tag{4.5}$$

where V = air velocity

Now Q = CAV

where Q = rate of volumetric discharge

C = coefficient unknown as yet; to be determined from tests)

A = area of opening.

(Note: The coefficient C is normally referred to as the coefficient of discharge and has been taken by various sources as 0.65 for a door opening).

The total volumetric discharge through half of the opening can be written as

$$Q = C\int_0^{H/2} W\left[2g\frac{\Delta\rho}{\bar{\rho}}z\right]^{1/2}.dz$$

On integrating this expression, the total volumetric discharge through one half of the opening will be

$$Q = C\frac{W}{3}\left[g\frac{\Delta\rho}{\bar{\rho}}\right]^{1/2}. H^{3/2} \tag{4.6}$$

With the flow Q is now associated the heat-transfer rate

$$\dot{q} = Q\bar{\rho}C_p\,(T_1 - T_2) \tag{4.7}$$

and the mass transfer rate, the transfer of gas or particles due to diffusion as opposed to bulk air movement,

$$\dot{m} = Q\bar{\rho}\,(c_1 - c_2) \tag{4.8}$$

where C_p is the specific heat.

Introducing now the heat-transfer coefficient h and the mass transfer coefficient h_m, defined as

$$h = \dot{q}/\left[WH\,(T_1 - T_2)\right] \tag{4.7a}$$

and

$$h_m = \dot{m}/\left[WH\bar{\rho}\,(c_1 - c_2)\right] \tag{4.8a}$$

equation 4.7 and 4.8 lead to the following equation in terms of dimensionless variables:

for heat transfer

$$Nu = \frac{hH}{k} = \frac{C}{3}\left(\frac{g\Delta\rho H^3}{\nu^2\bar{\rho}}\right)^{1/2} \cdot \frac{c_p\mu}{k}$$

$$= \frac{C}{3} Gr^{1/2} . Pr \qquad (4.9)$$

for mass transfer

$$Sh = \frac{h_m H}{D} = \frac{C}{3}\left(\frac{g\Delta\rho H^3}{\nu^2\bar{\rho}}\right)^{1/2} \cdot \frac{\mu}{\bar{\rho}D}$$

$$= \frac{C}{3} Gr^{1/2} . Sc . \qquad (4.10)$$

where the symbols are as defined in the Nomenclature.

Equation (4.9) and (4.10) cannot be exact for all conditions owing to neglect of viscosity in equation (4.5) and neglect of thermal conductivity and diffusivity in equations (4.7) and (4.8). The effect of these properties is considered in detail by Brown and Solvason. However it is adequate to state that if air is considered to be the convecting fluid over the tested temperature differential range, the pure conduction heat transfer would be quite negligible compared with that of convection. Hence, for air in this general range, the exponents of the Grashof, Prandtl and Schmidt numbers will not vary appreciably from those stated in equations (4.9) and (4.10).

4.1.2. The theory of the volumetric exchange of air due to the combined effect of natural convection and forced air flow through a rectangular opening in a vertical partition.

As far as the author is aware, no theory for the above conditions has as yet been written. The problem may be approached in a similar manner to that of natural convection, the only difference being that one of the rooms is under positive pressure due to air being supplied to it from an external source, Figure 4.2. In this case the enclosures are not sealed, air being supplied to one extracted from the other. In room 1, the pressure P, at a level Z below the centre line will be

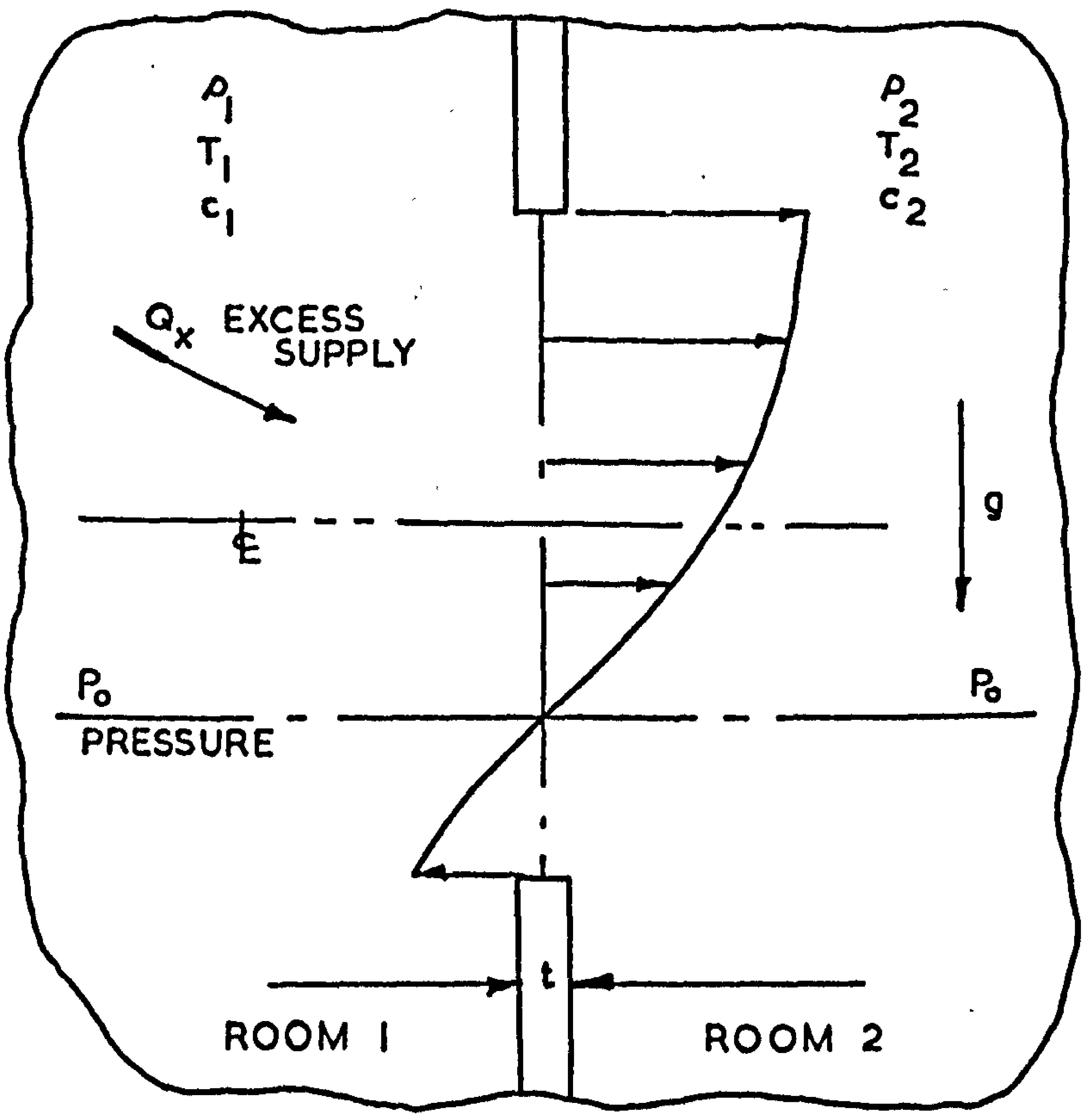


FIG. 4.2. SCHEMATIC REPRESENTATION OF COMBINED NATURAL CONVECTION AND FORCED-AIR FLOW ACROSS AN OPENING IN A VERTICAL PARTITION.

$$P_1 = P_{\mathfrak{c}1} + \rho_1 gZ \tag{4.11}$$

The pressure at the same level in room 2 will be

$$P_2 = P_{\mathfrak{c}2} + \rho_2 gZ \tag{4.12}$$

where $P_{\mathfrak{c}1}$ and $P_{\mathfrak{c}2}$ are the pressures at the centre line in rooms 1 and 2 respectively. The difference in these centreline pressures is equal to the additional pressure within room 1 due to the excess supply ventilation i.e. $P_{\mathfrak{c}1} - P_{\mathfrak{c}2} = P_x$

The pressure difference in these two rooms at the same level is

$$P_2 - P_1 = (\rho_2 - \rho_1) g Z - P_x \tag{4.13}$$

The pressure difference and supply pressure can be expressed as the height (h_a) of a column of air where the pressure due to temperature differential

$$h_1 = \frac{\rho_2 - \rho_1}{\bar{\rho}} Z = \frac{\Delta\rho}{\bar{\rho}} Z$$

and the supply air pressure

$$h_2 = \frac{P_x}{\bar{\rho} g} = \frac{V_x^2}{2g}$$

therefore from (4.13)

$$h_a = h_1 - h_2 \tag{4.14}$$

Similar limitations to that of the theory of natural convection regarding viscosity, thermal conductivity and diffusivity must also be considered in this analysis.

The Bernoulli equation may once again be assumed i.e.

$$V = (2gh_a)^{1/2}$$

$$= \left[2g\left(\frac{\Delta\rho}{\bar{\rho}} Z - \frac{V_x^2}{2g}\right)\right]^{1/2}$$

$$= \left[2g \frac{\Delta\rho}{\bar{\rho}} Z - V_x^2\right]^{1/2} \tag{4.15}$$

Now $Q_L = CAV$ where Q_L is the leakage inflow against the forced air flow.

$$\therefore Q_L = C \int_{L_2}^{L_1} W \left[2g \frac{\Delta\rho}{\bar{\rho}} Z - V_x^2\right]^{1/2} dZ$$

where limit L_1 represents the bottom or top of the door and has the value H/2 since the centre line of the door has been taken as the reference point, and L_2 is the neutral zone where supply pressure equals convective pressure occurring when

$$V_T^2 - V_x^2 = 0$$

i.e. the pressure due to the temperature differential equals the excess supply ventilation pressure

$$P_T - P_x = 0$$

On integrating the above expression, the leakage inflow through the door will be

$$Q_L = CW \cdot \frac{1}{2g(\Delta\rho/\bar{\rho})} \cdot \frac{2}{3}\left[2g\frac{\Delta\rho}{\bar{\rho}} \cdot \frac{H}{2} - V_x^2\right]^{3/2}$$

$$\therefore Q_L = CW \cdot \frac{1}{3} \cdot \frac{1}{g(\Delta\rho/\bar{\rho})}\left[g\frac{\Delta\rho}{\bar{\rho}}H - V_x^2\right]^{3/2} \qquad (4.16)$$

It has already been shown in (a) that with natural convection on its own, the Nusselt number may be expressed as a function of the Grashof and Prandtl numbers i.e. Nu = ϕ(Gr,Pr). This result is consistent with existing theory on natural convection. Existing theory on forced convection states that the Nusselt number may be expressed in terms of Reynolds and Prandtl numbers. As far as is known, no relationship yet exists for the combined effect of natural convection and forced air flow.

It may therefore be assumed that Nusselt number could be expressed in terms of both natural and 'forced' convection

$$\text{i.e. } Nu = \phi(Re, Gr, Pr)$$

The following theory proves this to be true and in the process introduces a dimensionless group, which for convenience will be called Sw.

With the flow Q_L is now associated the heat transfer rate

$$\dot{q} = Q_L \bar{\rho} c_p (T_1 - T_2) \qquad (4.17)$$

and the mass transfer rate

$$\dot{m} = Q_L \bar{\rho} \, (c_1 - c_2) \tag{4.18}$$

where c_p is the specific heat.

Introducing now the heat transfer coefficient h and the mass transfer coefficient h_m, defined as

$$h = \dot{q} / \left[WH(T_1 - T_2) \right] \tag{4.17a}$$

and

$$h_m = \dot{m} / \left[WH \bar{\rho} \, (c_1 - c_2) \right] \tag{4.18a}$$

equations (4.17) and (4.18) lead to the following equations in terms of dimensionless variables:

for heat transfer

$$Nu = \frac{hH}{k} = \frac{c}{3} \cdot \frac{c_p \mu}{k} \left[\frac{\mu V_b^3}{\nu^2 g \Delta\rho} \right] = \frac{c}{3} \cdot Pr . Sw . \tag{4.19}$$

where V_b is the equivalent velocity in the brackets of equation (4.16) and Sw is a dimensionless group.

From equation (4.19) it is found that the group, Sw, is in fact equal to the value

$$Sw = \frac{Re^3}{Gr} \cdot \frac{H^3}{D_h^3} \tag{4.20}$$

where D_h is the hydraulic diameter of the doorway. Thus the dimensionless group Sw, is a function of the Reynold and Grashof numbers, the height of the opening and the hydraulic diameter of the opening. No physical meaning can be attached to this group, as can be done, for instance, with Reynolds number (ratio of inertia forces to viscous forces), however it is none the less a dimensionless grouping. It can therefore be seen from this analysis that with combined natural convection and forced air flow, the Nusselt number can be represented by

$$Nu = \frac{c}{3} \cdot Pr \frac{Re^3}{Gr} \cdot \frac{H^3}{D_h^3} \tag{4.21}$$

For mass transfer, a similar analysis may be carried out leading to the following expression,

$$Sh = \frac{h_m H}{D} = \frac{c}{3} . Sc . Sw = \frac{c}{3} . Sc . \frac{Re^3}{Gr} . \frac{H^3}{D_h^3} \tag{4.22}$$

4.2 EXPERIMENTAL PROCEDURE

4.2.1 Test Area

The test area was situated at the Experimental Ward Unit at Hairmyres Hospital, East Kilbride, these tests being carried out in the isolation rooms of the intensive care area. A detailed plan of the ward area is given in Figure 4.3 and a plan of the two isolation rooms and their associated vestibule is given in Figure 4.4.

These rooms open into a common air lock or vestibule. The air was supplied to room No. 'A' by a high level grille and to room No. 'B' and the vestibule by ceiling diffusers. Both diffusers were shielded in order to prevent interference of the air movement pattern through the door by the supply air-stream. Each room had a low extract grille. All other extract grilles and bypass dampers were sealed.

Water filled cast iron radiators were positioned in each of the three rooms to supply additional heating to that of the supply air. These were controlled by a contact mercury thermometer. In addition to these heating facilities, a sheet of expanded polystyrene was placed over the window in room No. 'A' to reduce any heat loss through the window.

4.2.2 Instrumentation

The variables which were measured in the air movement tests were as follows:-

1. Air volumes being supplied and extracted to each room by the air-conditioning system.
2. Air temperatures in the rooms and doorways.
3. Air velocities in the doorways.
4. Airflow direction through the doorways.
5. Gas concentrations within each room to determine isolation efficiencies.

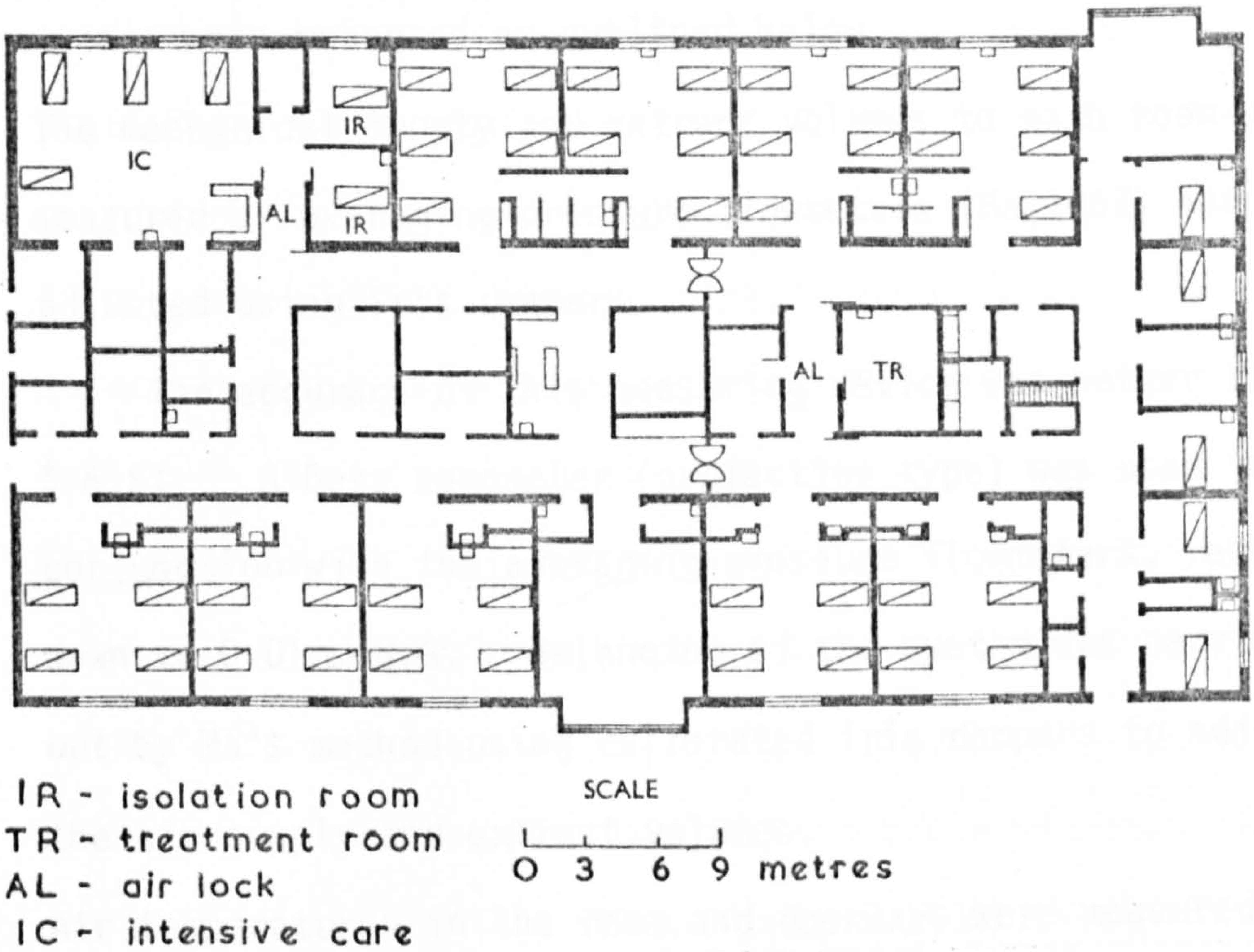


FIG. 4.3. PLAN OF EXPERIMENTAL WARD UNIT

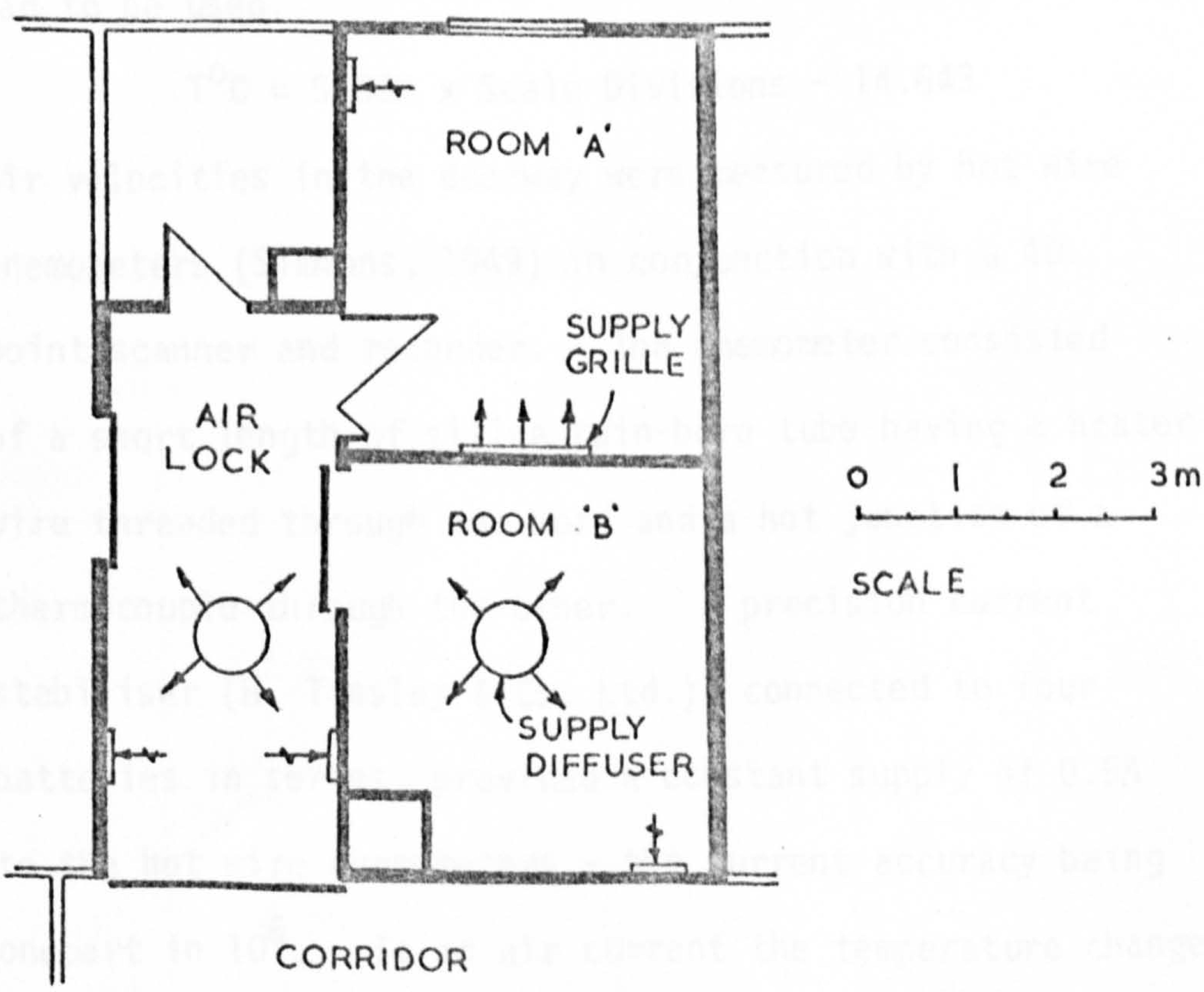


FIG. 4.4. PLAN OF TEST AREA.

These parameters were measured as outlined below.

1. The mechanical supply and extract volumes to each room were measured by averaging pressure flowmeters (Ma,1967) and adjusted using iris dampers.

   The accuracy of this measuring device was better than $\pm$ 6.5%. A Betz manometer (projection type) was used in conjunction with the averaging pressure flowmeters, reading down to 0.01mm W.G. Balancing of the system was carried out by Ma's method using calibrated iris dampers to adjust the air supply and extract volumes.

2. Air temperatures in the room and doorways were measured using copper-constantan thermocouples in conjunction with a multi-point recorder. This instrument had a range of 0 - 1 mV for temperatures in the range -18 to +27^{o}C (accuracy $\pm$ 0.3^{o}C.). As the recorder did not give a direct reading of temperature, the conversion formula shown below had to be used.

$$T^{o}C = 5.339 \times \text{Scale Divisions} - 14.643$$

3. Air velocities in the doorway were measured by hot wire anemometers (Simmons, 1949) in conjunction with a 40 point scanner and recorder. The anemometer consisted of a short length of silica twin-bore tube having a heater wire threaded through one bore and a hot junction of a thermocouple through the other. A precision current stabiliser (H. Tinsley & Co. Ltd.), connected to four batteries in series, provided a constant supply of 0.5A to the hot wire anemometers - the current accuracy being onepart in 10^{6}. In an air current the temperature change caused by heat loss was registered by the thermocouple. The e.m.f. developed served as a measure of the speed of flow of the air (accuracy $\pm$ 3%).

4. Airflow direction through the doorways was determined initially by using cotton wool swabs soaked in titanium tetrachloride which produced smoke when exposed to air. It was found however that this propogated rusting of the anemometers and anything metallic in the rooms. Cigarette smoke was subsequently used to determine the air direction.

5. To determine the isolation efficiency of a room, the nitrous oxide concentration within each room was measured by means of an Infra-Red Gas Analyser. The concentration of nitrous oxide was shown on a microameter, the scale of which was graduated in percentage gas by volume. The sample of gas to the analyser could be either neat or diluted with nitrogen as required by means of two flowmeters (1/16" and 1/8"). This system gave a maximum accuracy range of 1:1000.

4.2.3 Scope of Tests and Procedure

The 4-bed ward door and corridor door in the vestibule were closed and the mechanical ventilation supply and extract air volumes to each room were adjusted to the values shown below and maintained at these throughout the test series.

Table 4.1: Supply and extract air volumes (m^3/s)

Test Series	Ward 1		Ward 2		Vestibule	
	Supply	Extract	Supply	Extract	Supply	Extract
1 (B)	0.30	0.30	0.05	0.05	0.15	0.15
2 (P)	0.30	0.00	0.05	0.00	0.15	0.50
3 (P)	0.25	0.00	0.10	0.00	0.15	0.50
4 (P)	0.20	0.00	0.15	0.00	0.15	0.50
5 (P)	0.20	0.00	0.15	0.00	0.15	0.50
6 (B)	0.20	0.20	0.15	0.15	0.15	0.15
7 (P)	0.45	0.00	0.00	0.00	0.15	0.60

B - Balanced System. P - Positive System

Balanced ventilation systems (natural convection) had equal supply and extract volumes while the positive ventilation systems (combined natural convection and forced air flow) had only supply air.

All measurement and adjustment of these volumes were made from the roof plant room. Heating or cooling of the supply air to each room was also controlled from there.

If additional heating was required the radiator in that particular room was switched on from two to three hours before the beginning of a test to enable the radiator to come up to maximum temperature and the air temperature in the rooms to stabilise.

Since the correct or best positions to take the air temperatures were not known beforehand, it was decided to take vertical temperature grids in both rooms, vestibule and doorways. These grids consisted of a "meccano" strip with ten hot wires, anemometers and thermocouples fixed at equal intervals down its length. The grids were suspended either in the rooms close to the door yet away from the direct influence of the airstream through the doorway, or in the doorways themselves.

Different door areas were to be tested and these were set up by blanking off the door openings with wooden boards. The height of the doors were kept constant at 2.05m while the door widths were varied as follows:

Room 1	Room 2
1.20m ($2.46m^2$)	1.40m ($2.87m^2$)
0.90m ($1.845m^2$)	0.90m ($1.845m^2$)
0.50m ($1.025m^2$)	0.50m ($1.025m^2$)
0.10m ($0.205m^2$)	0.10m ($0.205m^2$)

Each test began by releasing nitrous oxide at a constant rate into the vestibule and taking the temperature at the grids in both isolation rooms and vestibule. The grids were then suspended in the doorways in such a manner that the air velocities and temperatures at any vertical section could be measured. Airflow direction at each point on the grid was then determined and the anemometer heads adjusted accordingly to face the oncoming airflow. If the direction was not definite, i.e. in the neutral zone, the anemometer head was placed sideways.

The thermocouple and anemometer readings were then recorded for that particular vertical section. Five sets of readings were taken and averaged. When recording had finished the grids were moved to their next position and the procedure repeated.

In order to obtain a useful picture of the air movement through the door and the temperature at the doorway, 5 vertical grid position readings were obtained for the 1.20m and 1.40m openings, 3 positions for the 0.90m, 2 for the 0.50m and 1 position for the 0.10m. Gas samples were then taken in each room, diluted by nitrogen in the ratio of approximately one part to eight. If the sample in the ward was too low, a neat sample of the gas/air mixture was taken to give a more accurate reading.

Once this procedure had been completed for a specific door area, the wooden boards were placed in the doorways to reduce the area to the required dimensions. The whole test procedure was then repeated for the new door area.

4.2.4 Treatment of data.

The velocity readings which had been recorded during the tests were transferred to punched cards to be used in conjunction with a trend surface analysis program.

This program calculated the volume flowing in and out through the door by fitting the best curve (linear, quadratic and cubic) to the results. The program also printed out isovel diagrams of the air movement in the doorway. The reference contour for these diagrams was taken as 0 and represented zero velocity, i.e. neutral zone.

```
EXPERIMENT    3


EQUATION COEFFICIENTS
LINEAR            Z1=        0.22488 +      -0.09967 X +      -0.01380 Y
LINEAR + QUAD Z1=      0.38894 +      -0.27743 X +      -0.34332 Y +      -0.01889 X2 +      0.18973 XY +      0.11892 Y2
LIN + QUAD + CUB, Z1=      0.19816 +      2.03634 X +      -0.39227 Y +      -6.81832 X2 +      0.85156 XY +      -0.00358 Y2
    +      5.28579 X3 +      -0.32770 X2Y +      -0.17871 XY2 +      0.06590 Y3

ERROR MEASURE LINEAR TREND SURFACE                         I=      0.00
PERCENT TOTAL SUM SQUARES LINEAR SURFACE                   I=     -9.77
PERCENT TOTAL SUM SQUARES QUADRATIC SURFACE             I=     77.69
PERCENT TOTAL SUM SQUARES CUBIC SURFACE                 I=     93.09

VOLUME BENEATH LINEAR SURFACE                I=     -0.3129
VOLUME BENEATH LIN+QUAD SURFACE              I=     -0.3080
VOLUME BENEATH LIN+QUAD+CUB SURFACE          I=     -0.3091
ARITH. MEAN Z, I= SUM OF Z VALUES/ N         I=      0.1659
AVERAGE Z VALUE, LINEAR SURFACE              I=      0.1655
AVERAGE Z VALUE,LIN+QUAD SURFACE             I=      0.1630
AVERAGE Z VALUE, LIN+QUAD+CUB SURFACE I=      0.1636
AREA OF MAP IN SQUARED UNITS                 I=     -1.8900

Z VALUES ORIG DATA
      0.225          0.253          0.228
      0.163          0.171          0.162
      0.150          0.142          0.135
      0.105          0.132          0.115
      0.093          0.133          0.096
      0.117          0.118          0.119
      0.185          0.122          0.126
      0.265          0.176          0.128
      0.315          0.194          0.116
      0.334          0.209          0.129


NO OVERPRINT VALUES


EXPERIMENT    3.


EQUATION COEFFICIENTS
LINEAR            Z1=      -0.31675 +      0.09967 X +      0.26183 Y
LINEAR + QUAD Z1=     -0.48543 +      0.74553 X +      0.37593 Y +      -0.49889 X2 +      -0.19178 XY +      -0.01354 Y2
LIN + QUAD + CUB, Z1=     -0.32058 +      -0.85609 X +      0.19116 Y +      4.10962 X2 +      -0.63417 XY +      0.35605 Y2
    +      -3.48261 X3 +      0.09061 X2Y +      0.17576 XY2 +      -0.14570 Y3

ERROR MEASURE LINEAR TREND SURFACE                         I=      0.00
PERCENT TOTAL SUM SQUARES LINEAR SURFACE                   I=     86.98
PERCENT TOTAL SUM SQUARES QUADRATIC SURFACE             I=     91.22
PERCENT TOTAL SUM SQUARES CUBIC SURFACE                 I=     94.12

VOLUME BENEATH LINEAR SURFACE                I=     -0.0057
VOLUME BENEATH LIN+QUAD SURFACE              I=      0.0006
VOLUME BENEATH LIN+QUAD+CUB SURFACE          I=      0.0035
ARITH. MEAN Z, I= SUM OF Z VALUES/ N         I=     -0.0031
AVERAGE Z VALUE, LINEAR SURFACE              I=      0.0030
AVERAGE Z VALUE,LIN+QUAD SURFACE             I=     -0.0004
AVERAGE Z VALUE, LIN+QUAD+CUB SURFACE I=     -0.0018
AREA OF MAP IN SQUARED UNITS                 I=     -1.8900
```

FIG. 4.5.A.   TREND SURFACE ANALYSIS PRINTOUT.

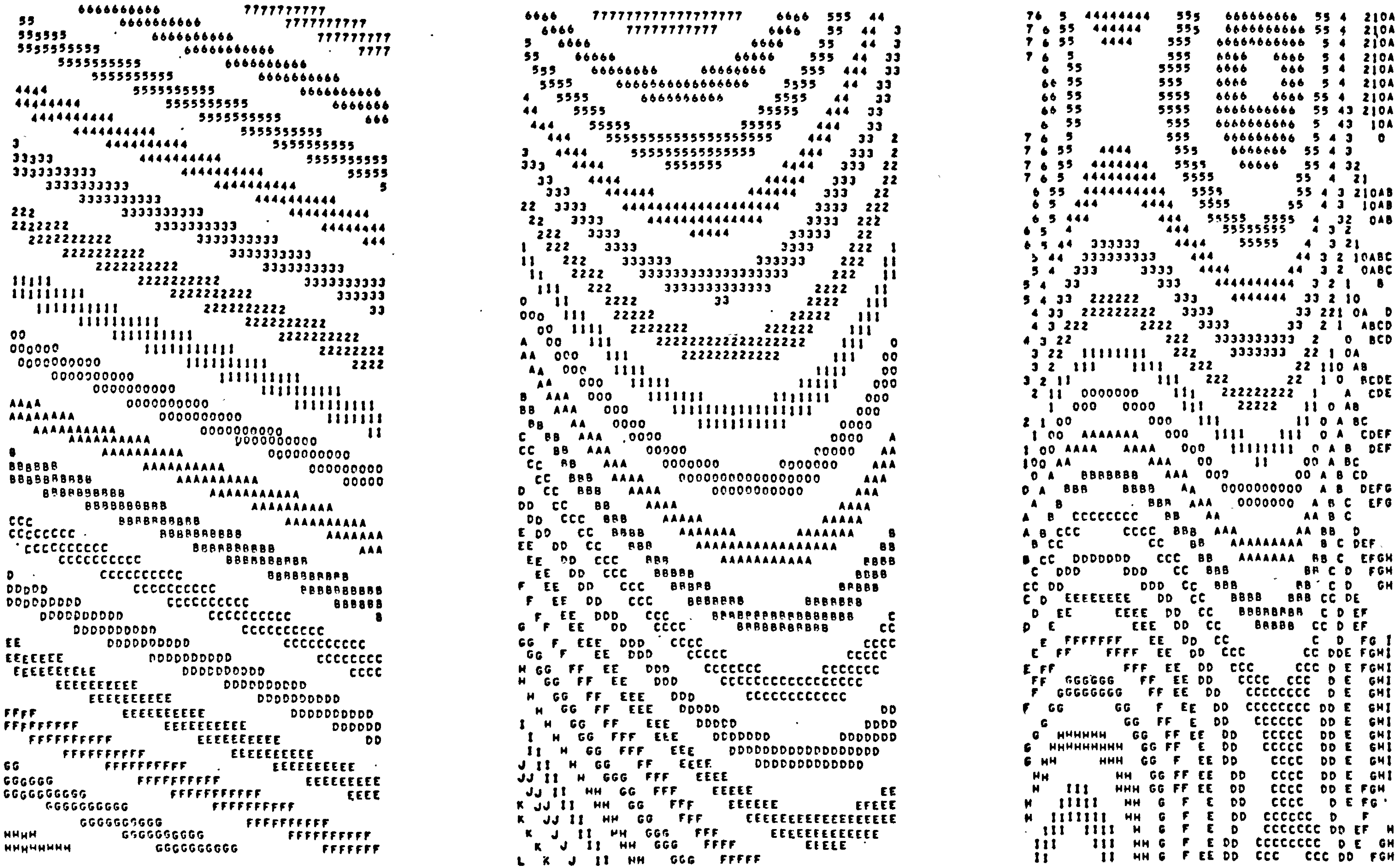


FIG. 4.5.B. ISOVEL DIAGRAMS : LEFT, CONTOURS OF LINEAR TREND SURFACE; MIDDLE, CONTOURS OF LINEAR PLUS QUADRATIC TREND SURFACE; RIGHT, CONTOURS OF LINEAR PLUS QUADRATIC PLUS CUBIC TREND SURFACE.

Contour intervals were taken as 0.02 representing 0.02m/s. A typical printout for a 0.90m wide door opening is shown in Figure 4.5. In this case there was a 2.40°C temperature differential across the opening.

Suppose that the outflow volume is represented by A, and the inflow volume by B. Then the first section of Figure 4.5A adds the inflow and outflow volumes and the second section substracts the two volumes.

i.e. A+B = 0.3129 (volume beneath linear surface - first section)

A-B = 0.0057 (volume beneath linear surface - second section)

$\therefore$ 2A = 0.3186

$\therefore$ A = 0.1593 and B = 0.1536

Average transfer volume in or out = <u>0.1565m^3/s</u>

Figure 4.5B shows the isovel diagrams drawn by the computer for the linear, quadratic and cubic trend surfaces.

The velocities from the printouts can be determined from the list below.

Printout Value	Velocity Range	
3	+ 0.11	+ 0.13
2	+ 0.07	+ 0.09
1	+ 0.03	+ 0.05
0	- 0.01	+ 0.01
A	- 0.03	- 0.05
B	- 0.07	- 0.09
C	- 0.11	- 0.13

The KDF9 computer at the University was used for these computations.

4.3 The choice of measuring points to determine the temperature differential across an opening.

Simple though it may appear at first thought, the decision as where to measure the difference in temperature between two rooms proved to be a difficult problem. This diffuculty was caused by the fact that the temperature in rooms varied both from floor to ceiling and wall to wall. In the case in question these changes in temperature were emphasised by the smallness of the room, the quantity of the air supply volume and the large localised heat output of the radiators.

Shown in Figure 4.6 is a diagrammatic representation of one probable situation. As described, radiators were installed in the room but these would give off convective currents which could rise and force their way out of the top of the door. Measurement at the mid-point of the room would fail to take into consideration the effect of the radiators.

It has been postulated in the theory that the mid-point of the door would be the neutral zone, i.e. the point at which the direction of airflow changes. A situation as described in Figure 4.6 would possibly generate an air flow profile through the door as shown. If this was so the implications would not effect the practical application of the results but it could complicate the theory and at times invalidate the explanation of the results. Both these problems are discussed in the following sections.

4.3.1 Measurement of temperature.

Where to measure temperature was a problem to which there was no satisfactory answer. It was resolved therefore to measure what was considered sufficient temperatures to cover all eventualities. These were 10 readings at equal intervals from the floor to the top of the door taken in the doorway and in each room just to the side of the opening, far enough away from the opening not to be influenced by it, but near enough to give a representative reading.

It was therefore possible to measure the following temperatures as shown in Figure 4.7.

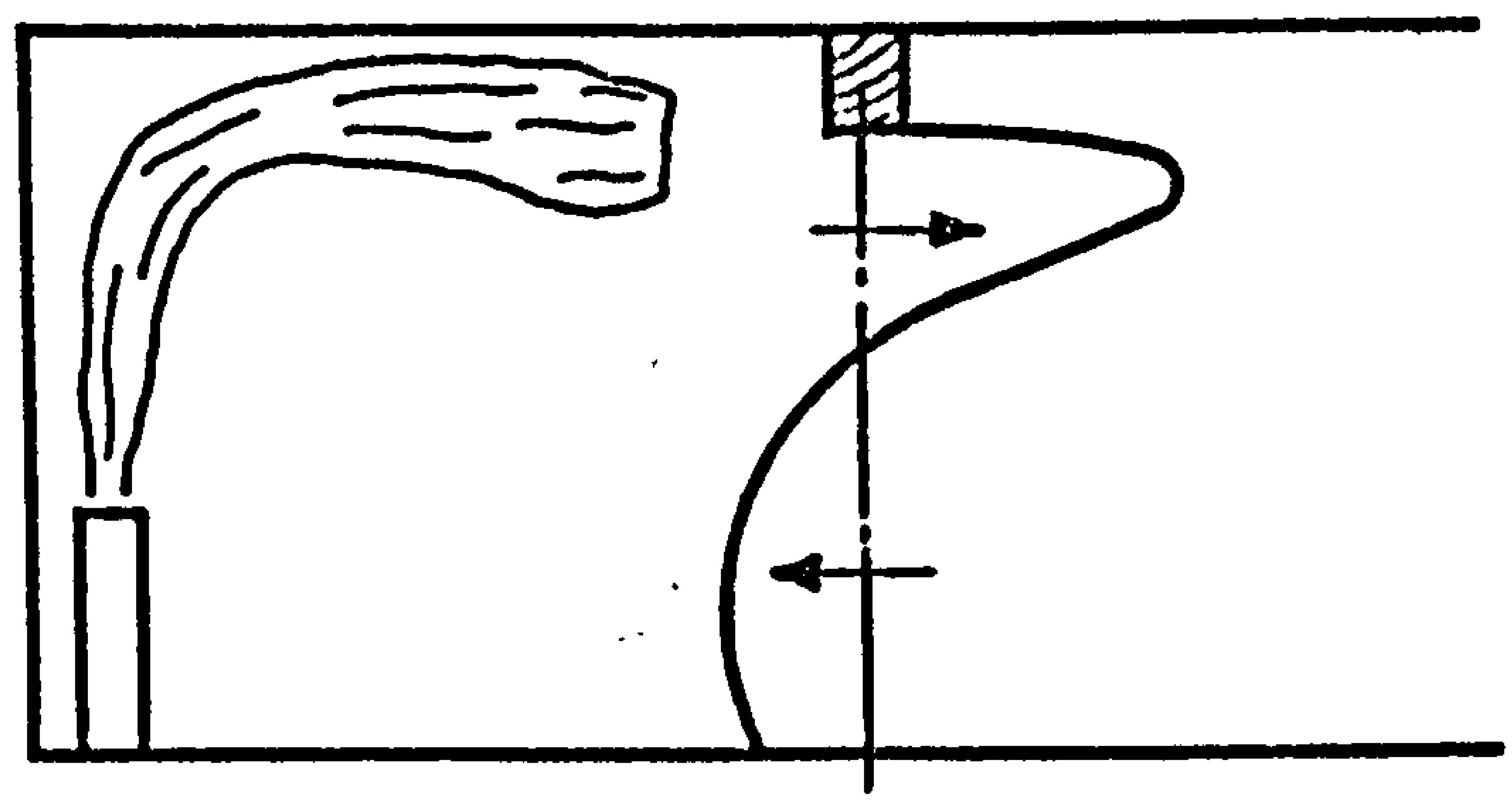


FIG.4.6. STRATIFICATION DUE TO THE USE OF A RADIATOR IN ONE OF THE ROOMS.

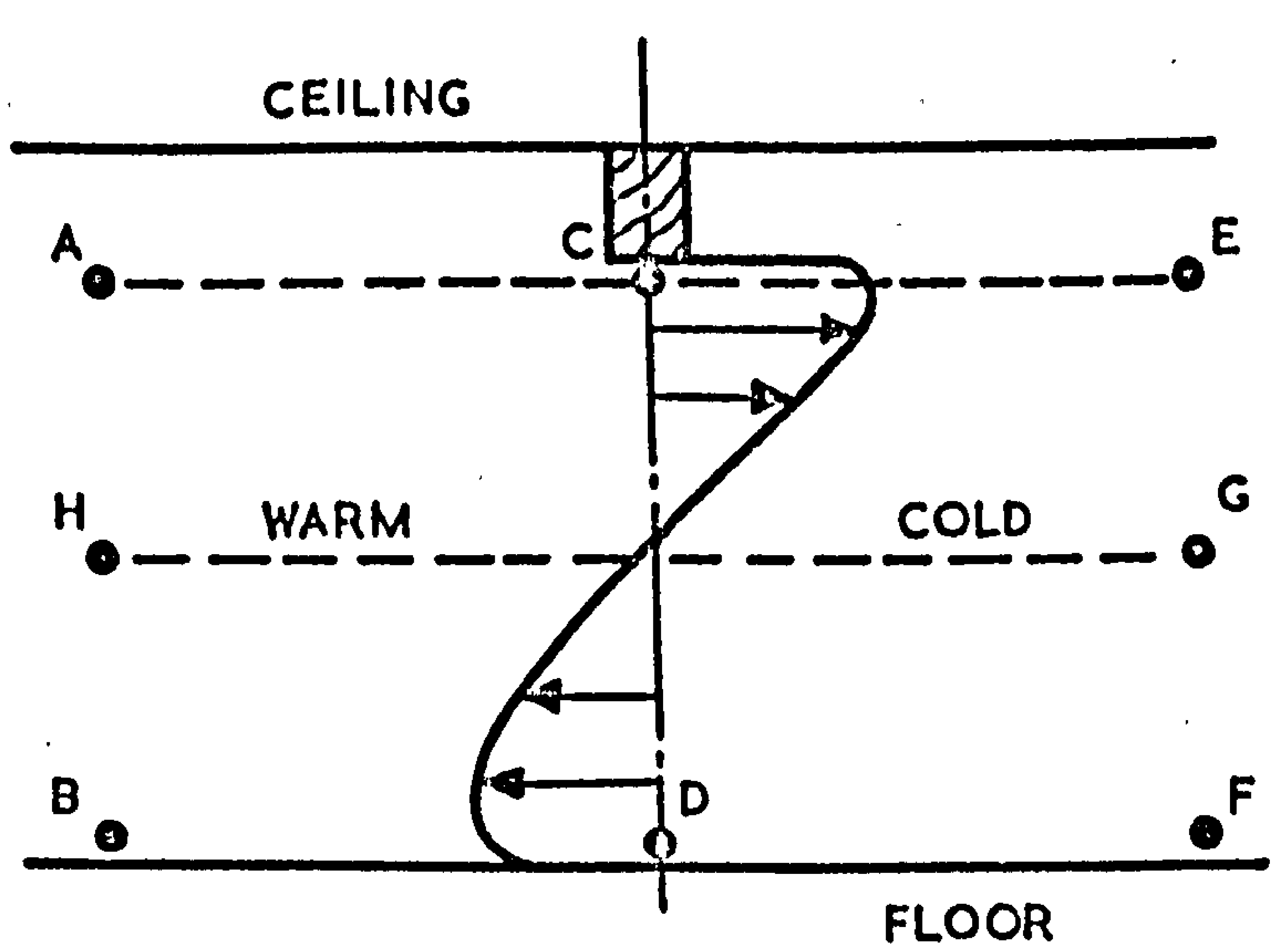


FIG.4.7. TEMPERATURE MEASURING POSITIONS.

1. Temperature difference between top and bottom of the doorway (C - D).
2. Difference between top of warm room and bottom of cold room (A - F).
3. Difference between the mean of the top and bottom temperature in each room.

$$\frac{(A + B)}{2} - \frac{(E + F)}{2}$$

4. Difference between the two centre temperature in each room H - G.
5. Difference between the mean of 10 reading on each room grid.

$$\frac{\Sigma(1 \rightarrow 10)}{10}\text{on grid A/B} - \frac{\Sigma(1 \rightarrow 10)}{10}\text{on grid E/F}$$

6. Difference between the mean temperature of the air flowing out of the warm room - the mean temperature of the air flowing into the warm room. (Volume weighted mean)

This decision as to what was the best temperature measuring point was resolved by determining the temperature which most accurately reflected the amount of air flow through the door. This was done by regression analysis.

A multiple regression analysis was used where full account was taken off:-

a) the six temperature difference as stated above
b) door area
c) room number
d) volume of air suplied to the room
e) volume of air flowing out of the adjacent door and,
f) whether this volume was adjacent or opposite the other rooms exit air.

This showed that the temperature which reflected most accurately (i.e. by utilising the analysis of variance and comparing the residual sum of squares for each regression analysis) the air flow quantities were in the following order:- 6, 1, 4, 3, 5 and 2.

It may be seen that the temperature differential (6) turned out to be the most accurate. This is however the most complicated differential to measure in practice as it requires a grid of temperature to be taken in the doorway and directional tests to be carried out using tracer smoke. Differential (1) turned out to be the second best choice. This is thought to be the best differential with respect to the theory and is the simple difference between the top and bottom temperatures in the doorway.

In conclusion the temperature differential as measured as the average of the air temperature passing through the doorway would most accurately estimate the flow rate but this temperature differential is too complicated to measure. The next most accurate temperature differential is that of the difference between the top and bottom of the door. The latter temperature differential was chosen but any other could have been used. This would however require a change in the coefficient of discharge, i.e. no matter which of the air temperature measuring positions was used the same answer would be obtained, providing that the appropriate coefficient of discharge for that particular position was employed.

4.3.2 Temperature Stratification

The problem of stratification was considered as shown in Figure 4.6 this being caused primarily by the use of a radiator to achieve high temperature differentials between two areas. This is somewhat unusual in practice as shall be shown in Section 6, where the differentials were found to be a maximum of around $1^{o}C$.

Even with higher differentials however, the interaction of the incoming conditioned air through the ceiling diffuser created turbulence and destroyed the stratification caused by the radiator (Figure

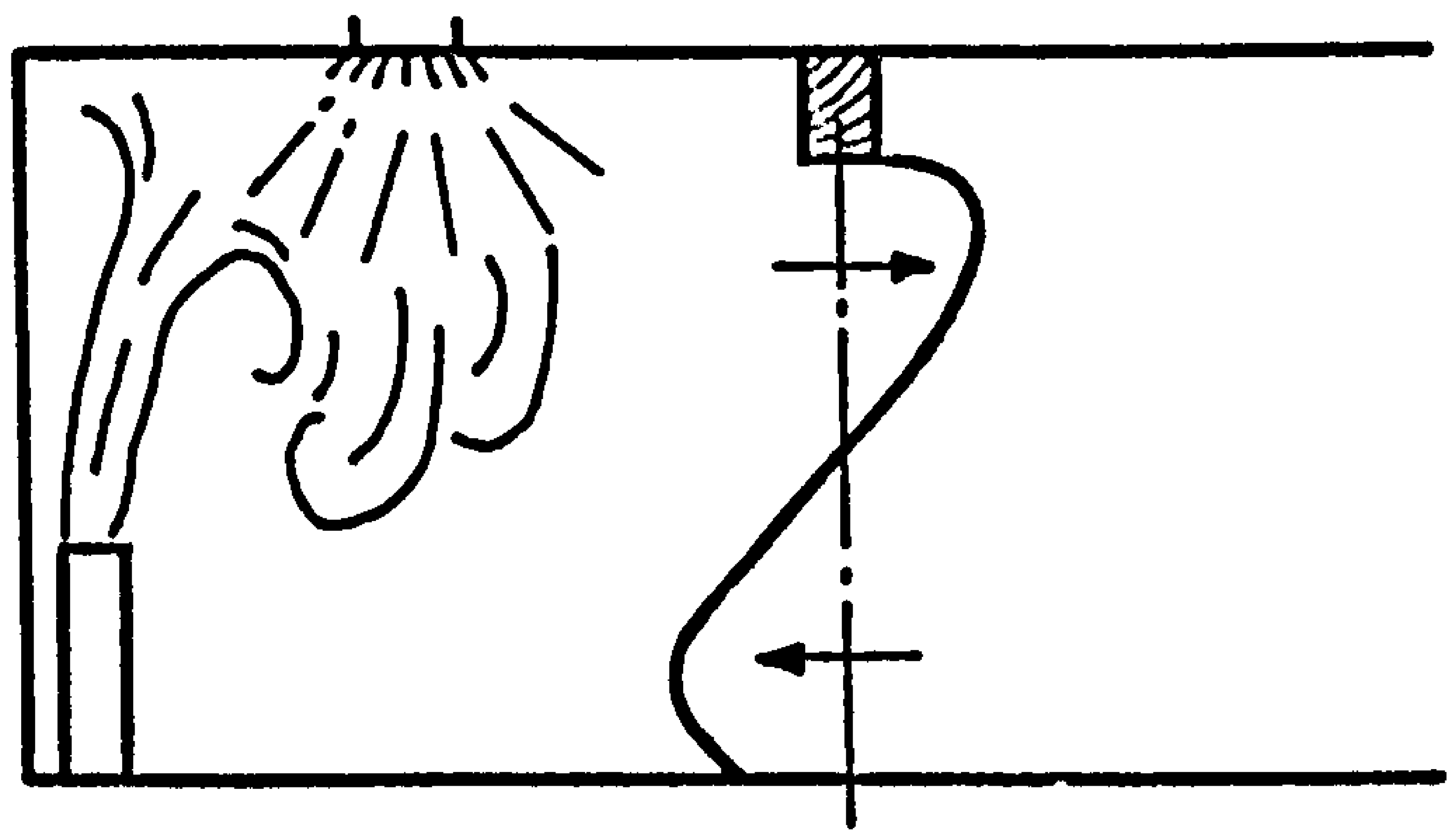


FIG.4.8. DISRUPTION OF STRATIFICATION DUE TO INTERACTION OF INCOMING AIR THROUGH DIFFUSER.

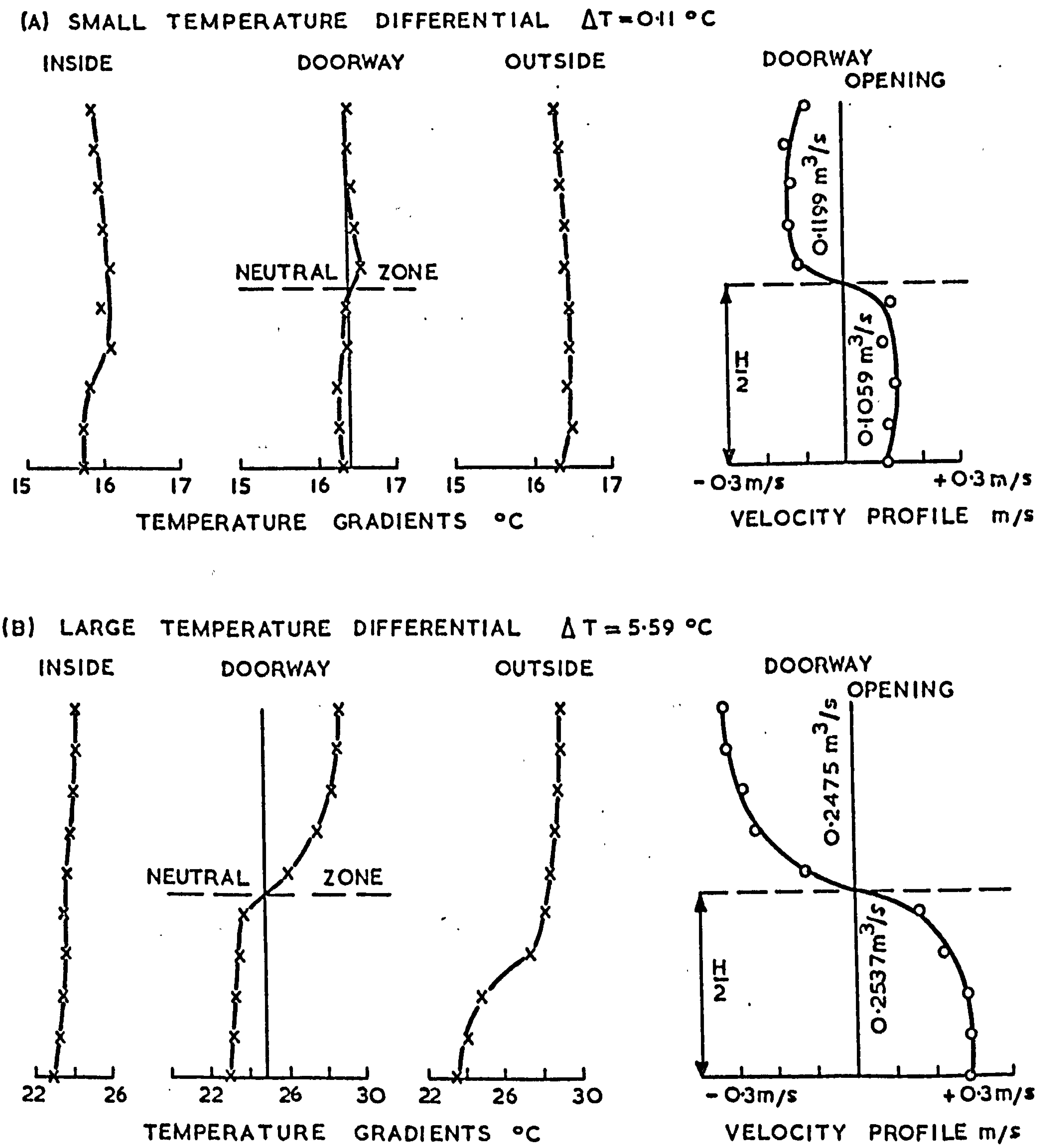


FIG. 4.9. TEMPERATURE GRADIENTS AND VELOCITY PROFILES INDICATING POSITION OF NEUTRAL ZONE.

4.8). This action in turn stabilised the system, ensuring that the neutral zone would be at midpoint of the doorway. This can be seen in Figure 4.9 which shows the temperature gradients and velocity profiles, indicating the neutral zone for both a small temperature differential ($0.11^{o}C$) and a large temperature differential ($5.59^{o}C$). Although all the results are not as good as those presented, inspection of the results and allowing for the screening of one door by air passing through the other, and a tendency at times for air to stream out at the corners, shows that the point of inflection was found to be at the mid-point of the opening.

4.4 Test Results

4.4.1 Natural Convections.

The temperature difference which was used in the analysis of the results was that of the temperature difference between top and bottom of the opening as stated above in 4.3. This was thought to be the most appropriate differential with respect to the theory. The air temperature used in determining the dimensionless groups was that of the average of the top and bottom temperatures at the opening thus giving a mean heat transfer coefficient.

The experimental transfer volumes for various door openings are presented in Figure 4.10. These lines were found to converge at a false origin and the gradients to be in a linear relationship with area of opening. It was thus possible to also depict an area axis.

Some results for the 1.40m wide door were considerably lower than predicted volumes. This divergence from theory may be explained in two ways, firstly the interaction of the air flow through the two doors when fully opened, and secondly that the co-efficient of discharge could vary with area of opening.

It can be seen from Figure 4.4 that the two test doors were very close to one another, and that when both openings were at their maximum area the air flow of one could influence the amount of air flow out

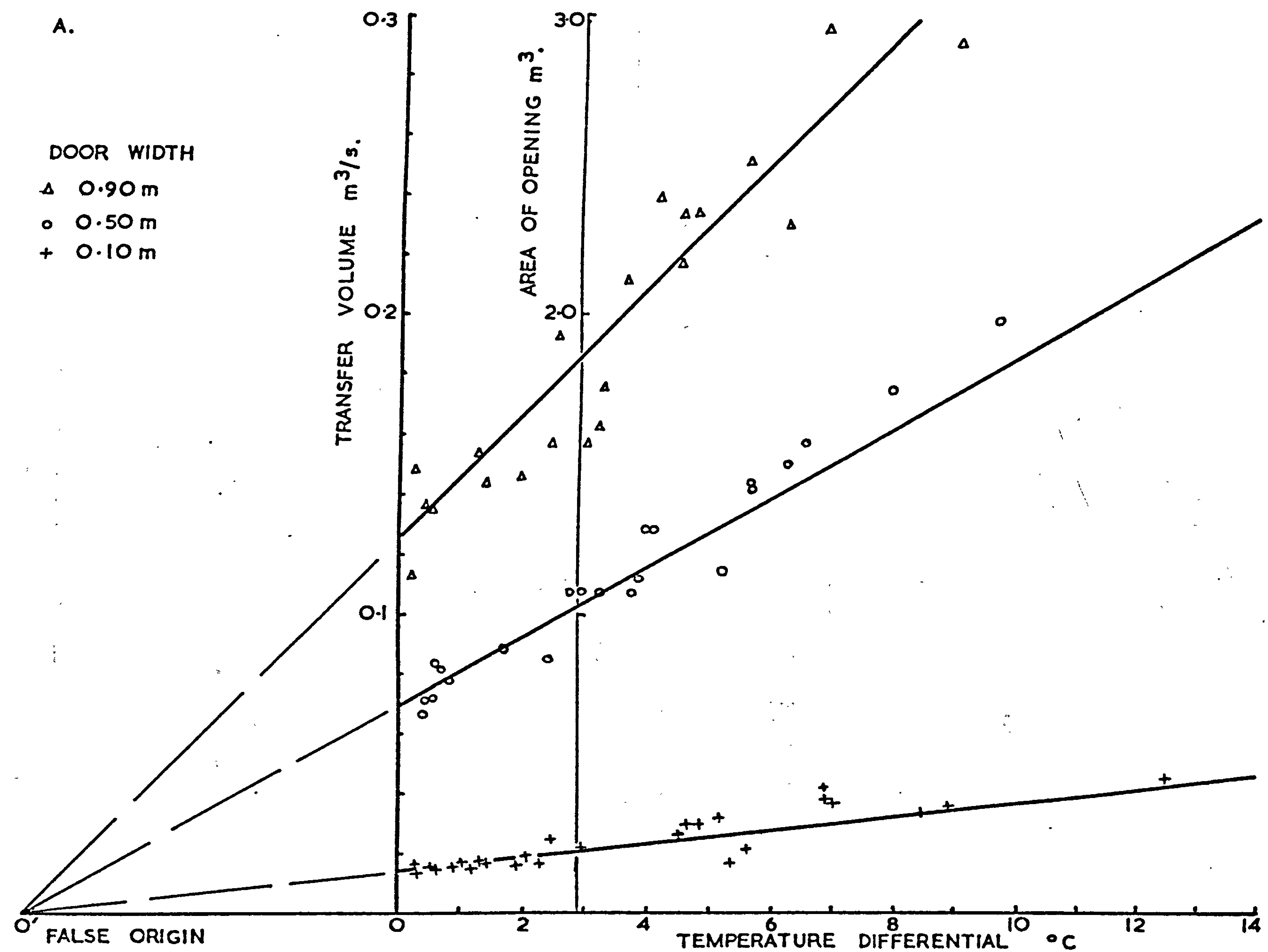
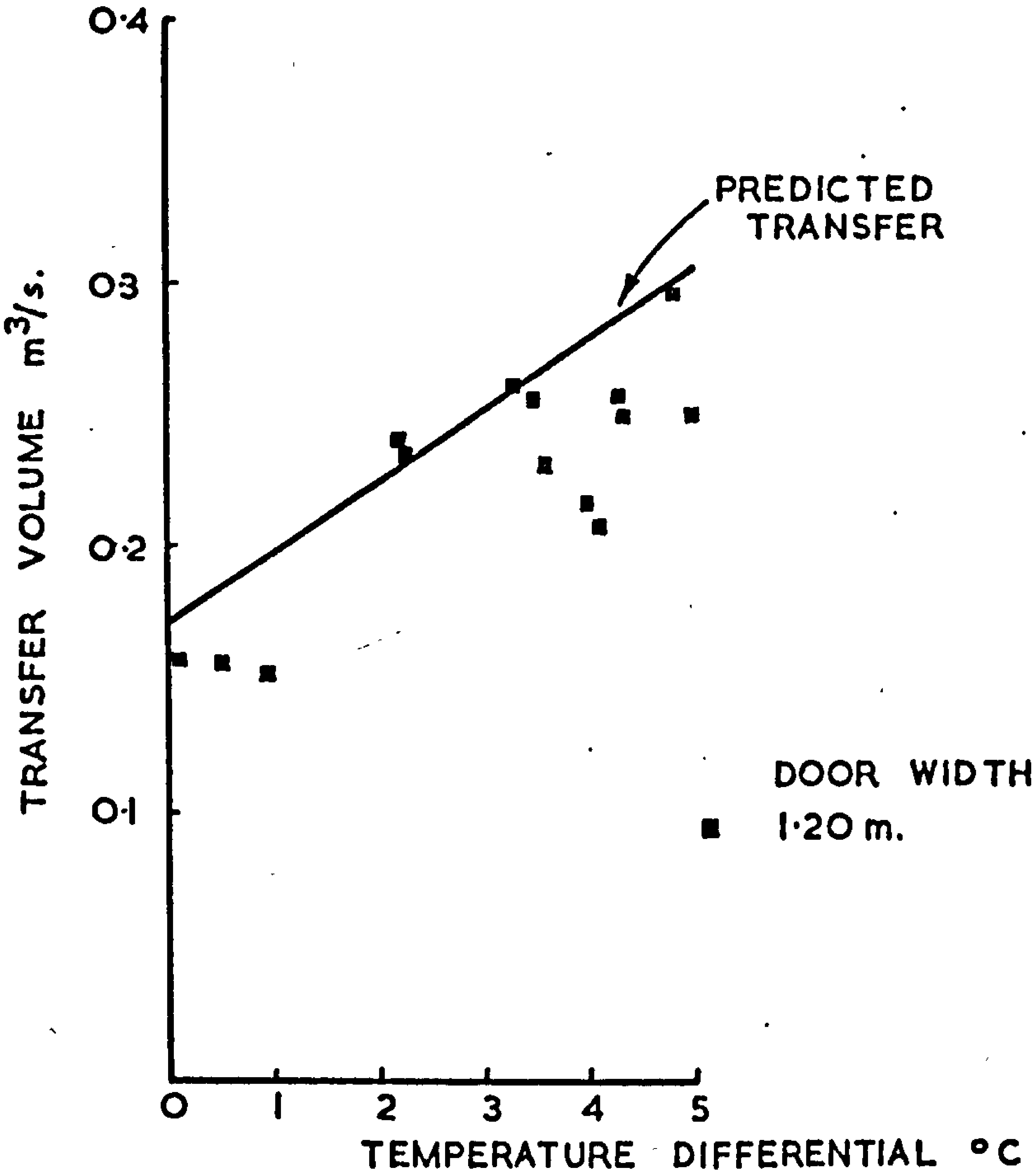


FIG. 4.10. EXPERIMENTAL NATURAL CONVECTIVE TRANSFER VOLUMES (m^3/s).

B.



C.

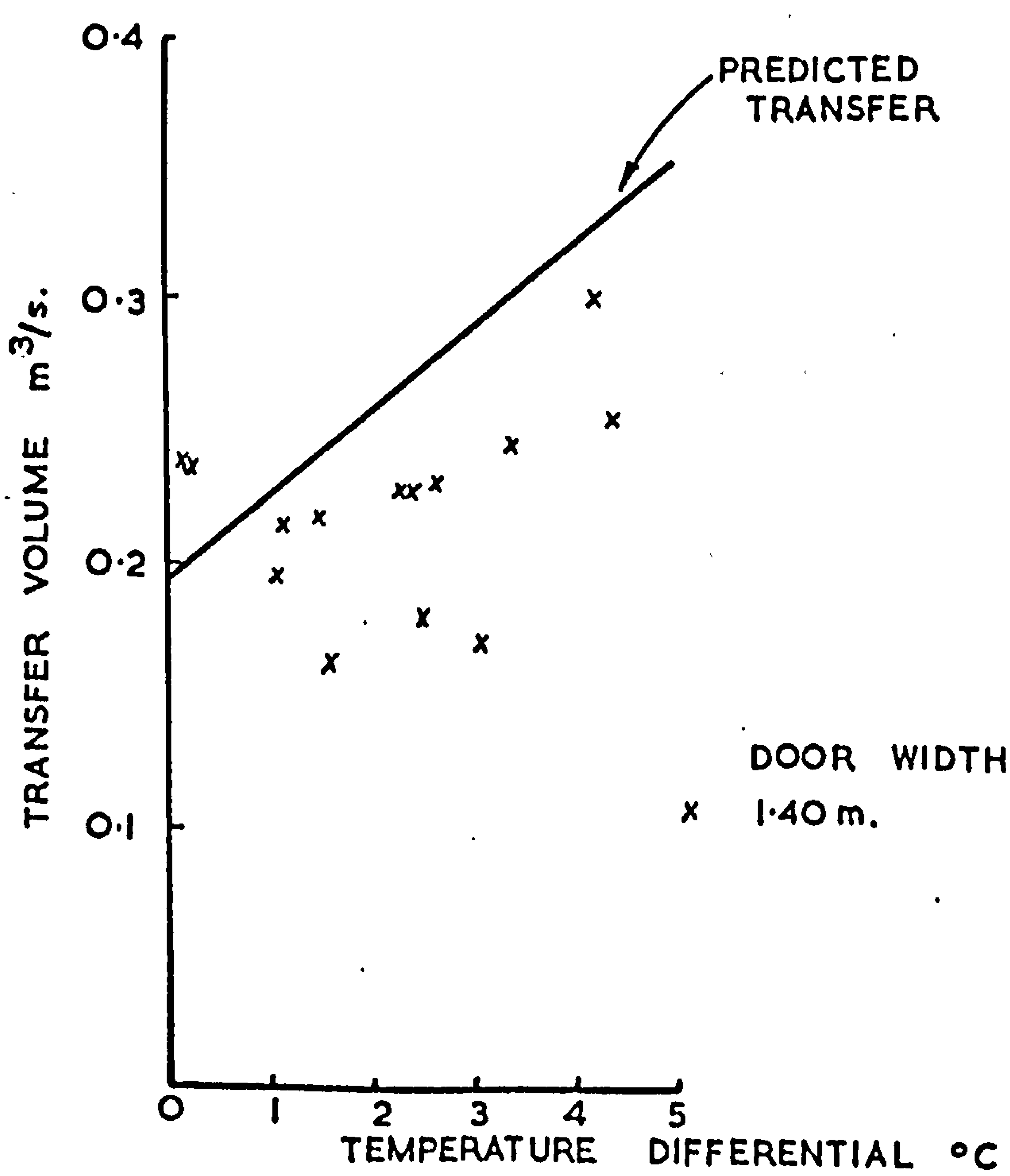


FIG.4.10. CONTD.—NAT. CONV. TRANSFER VOLUMES m^3/s

of the other. For example if both rooms were warmer than the vestibule then a large air flow out of the top half of door 'A' could screen the top half of door 'B', thus influencing the amount of air flowing in and out of that room. The smaller the openings, and hence further apart, the less influence one door would have on the other. This was in fact established when a multiple regression analysis was run on the influence of various variables on the air flow through the door. It was found that if the air flowing out of room 1 was adjacent to the air flowing out of room 'B', i.e. if flowing out of the top of both doors or bottom of both doors then this fact was significant. Under these conditions the air flow out of room 1 masked the air flow out of room 'B' and subsequently reduced the volume of air transfer out of that room. The corrected values for the 1.40m door (room 'B') were replotted.

The results for the 1.20m wide and 1.40m wide doors are shown in Figures 4 . 10.b. and 4.10.C and as can be seen, they are still slightly lower than the predicted volume lines. There is no evidence to support the theory that the coefficient of discharge varies with area of opening (see Figure 4.11) and I am therefore inclined to consider that the discrepancy is still due to influence of the proximity of the doorways and to the interaction of the air flowing through the two doors when fully open.

The coefficient of discharge values for natural convection were obtained by dividing the actual convective transfer volume, from the test results, by the basic theoretical volume, i.e. equation (4.6.) without a coefficient. The coefficient values were found to be primarily a function of temperature differential, the door area not being significant in the range of areas which we studied. (Figure 4.11). It was therefore decided to refer to the coefficient as the coefficient of temperature. Figure 4.11 is interesting in the fact that from about $4^{\circ}C$ differential downwards, the value of the coefficient increases

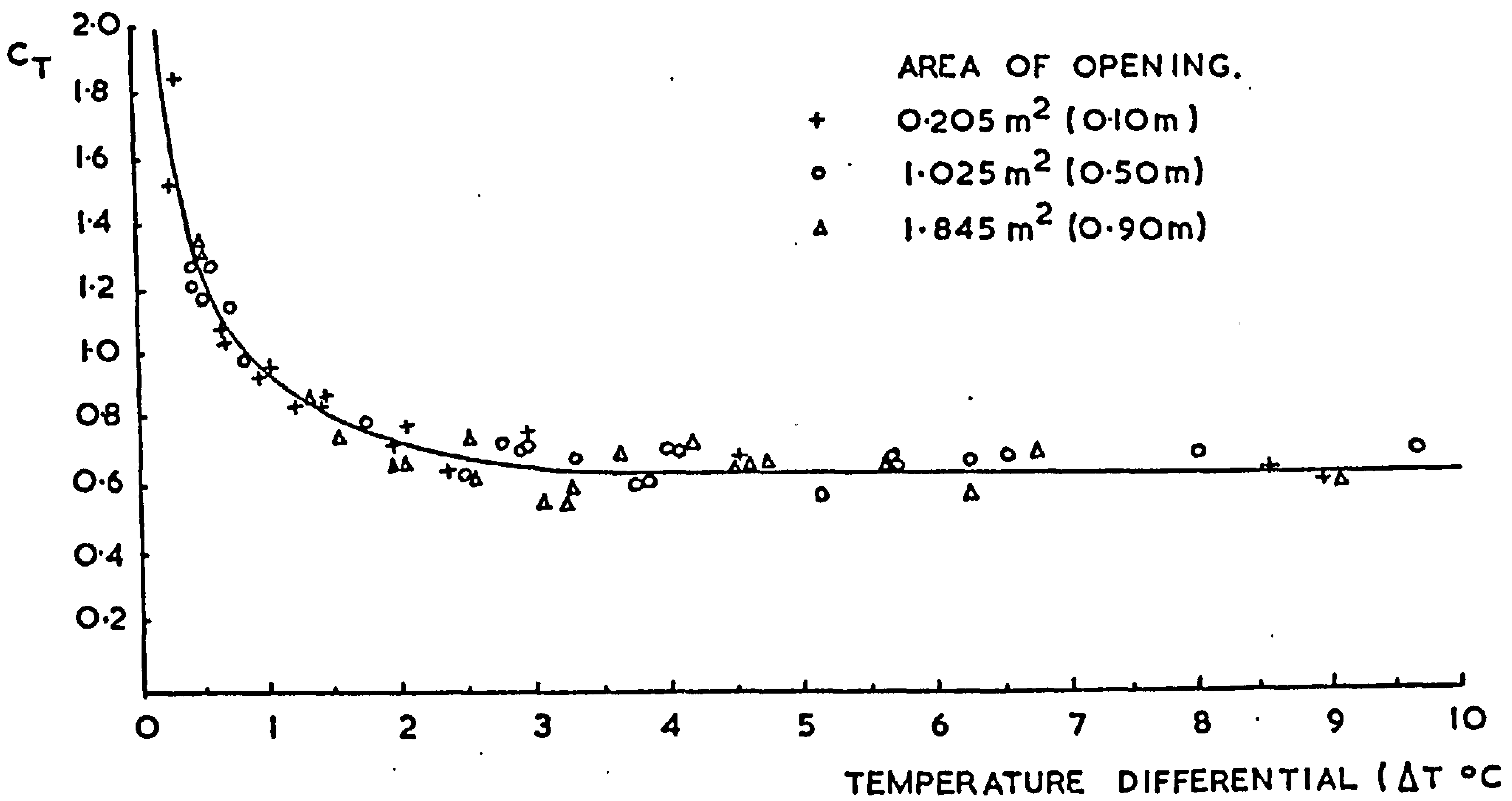


FIG.4.11. COEFFICIENT OF TEMPERATURE (C_T).

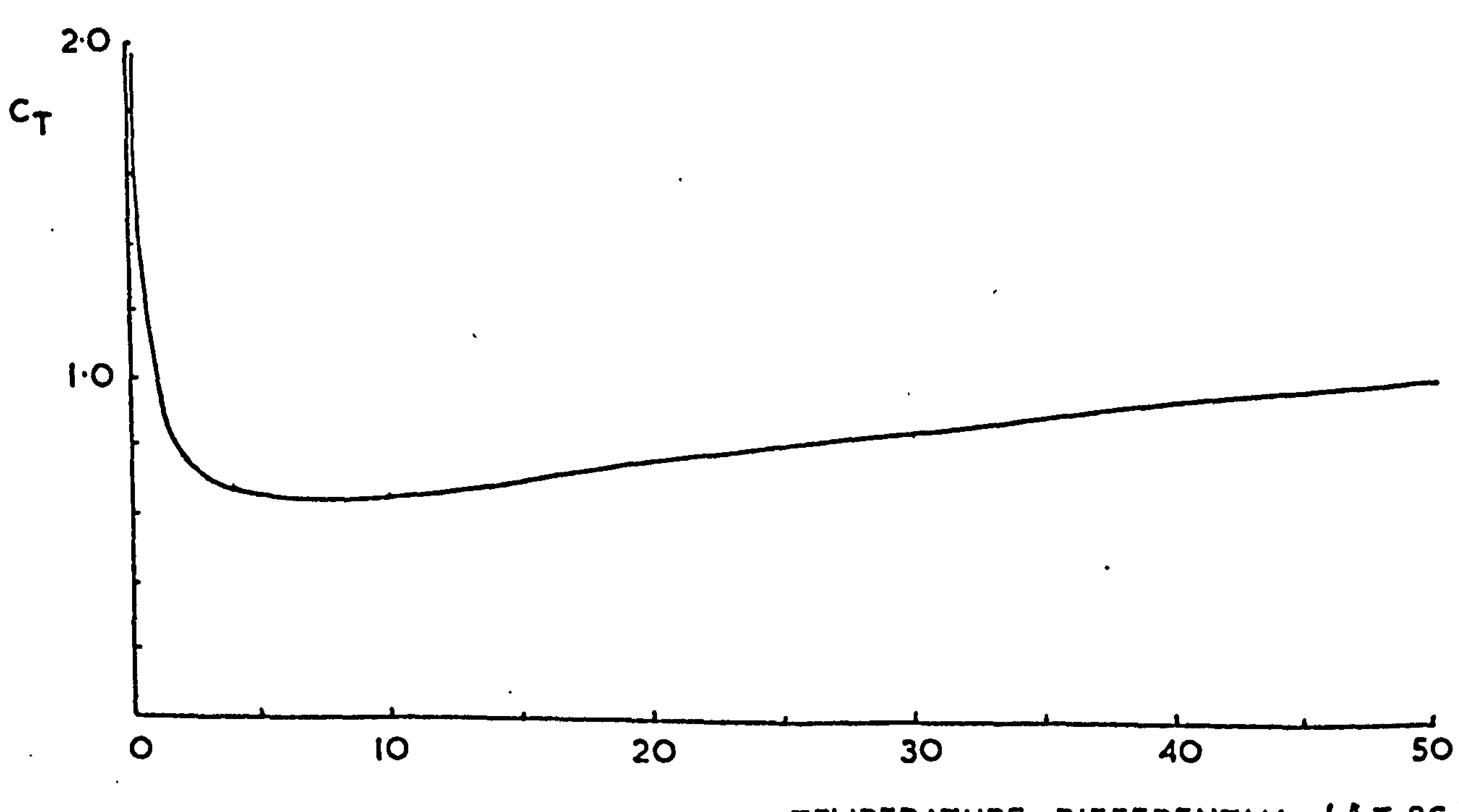


FIG.4.12. COEFFICIENT OF TEMPERATURE FOR LARGE DIFFERENTIALS.

asymptotically with the coefficient axis. The reason for this trend may be explained as follows. By dividing the convective transfer volumes in or out of the doorway at zero temperature differential (by extrapolation) for each door width, by half the correspondong door area, a mean velocity in or out of the doorway may be arrived at.

This results in a mean velocity of 0.1362m/s (27 ft/min) for any door area. As the free air velocity, or turbulence, within a ventilated room is generally quoted as being in the range 0.1016 - 0.1524m/s (20-30 ft/min) this strengthens the validity of the experimental results and explains why the coefficient does not remain constant at 0.65.

Above 10^{o}C temperature differential the coefficient rises again, very slowly this time, reaching a value of unity at about 50^{o}C differential and continuing to rise. This is shown in Figure 4.12. The reason for this trend is not at present apparent, however it compares favourably with limited results of Fritzsche and Lilienblum working in the region of 20-30^{o}C differential. (Figure 4.13).

Experimental results for heat transfer are given in Figure 4.14 where the Nusselt number divided by the Prandtl number is ordinate and the Grashof number is abscissa. For comparison and verification of the theory, results of Brown and Solvason are also shown. With an opening 2.05m high, the upper theoretical curve, and a temperature differential of 10^{o}C, the Grashof number equals 1.3 x 10^{10}. As can

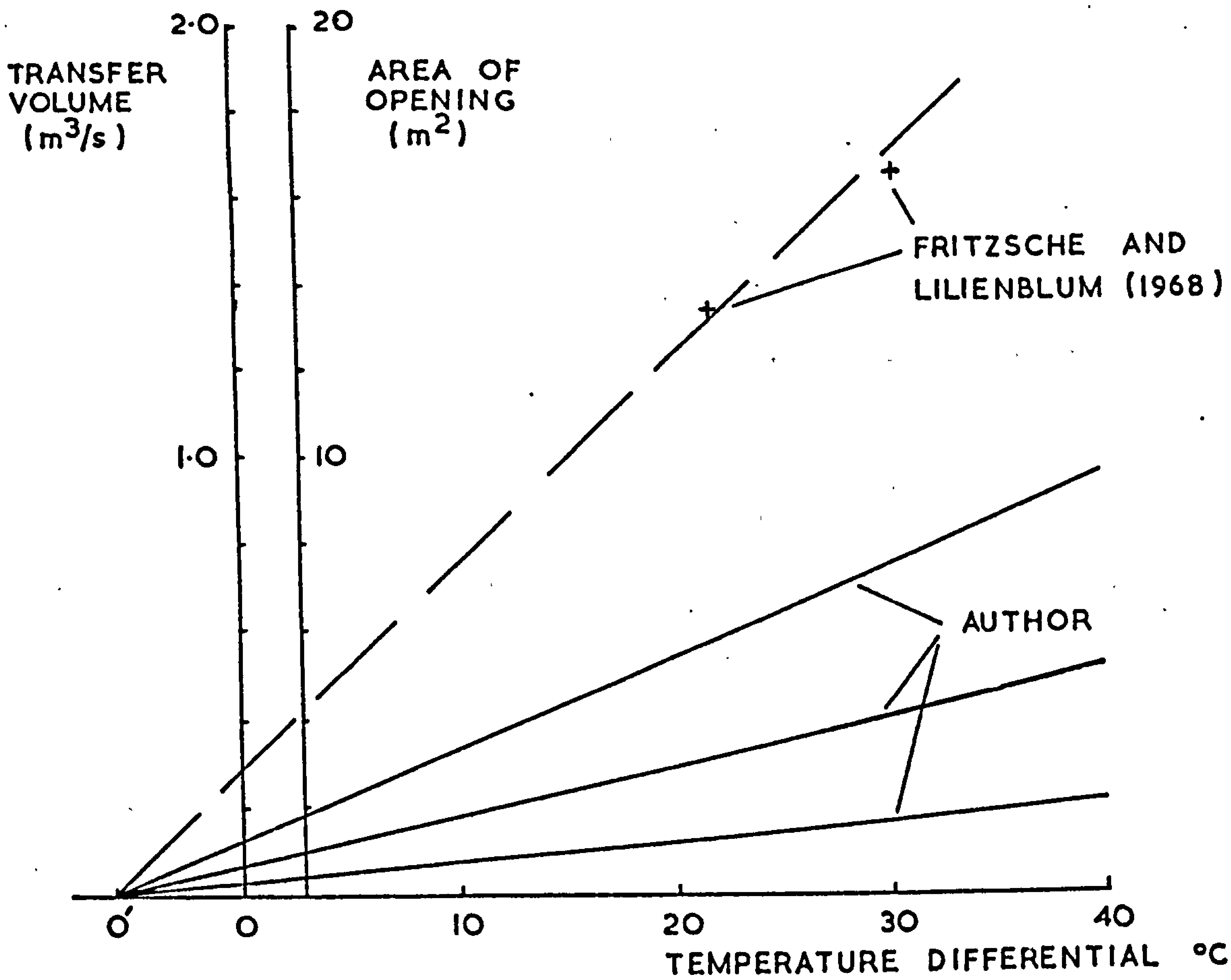


FIG. 4.13. NATURAL CONVECTIVE TRANSFER VOLUMES — LARGE DIFFERENTIALS.

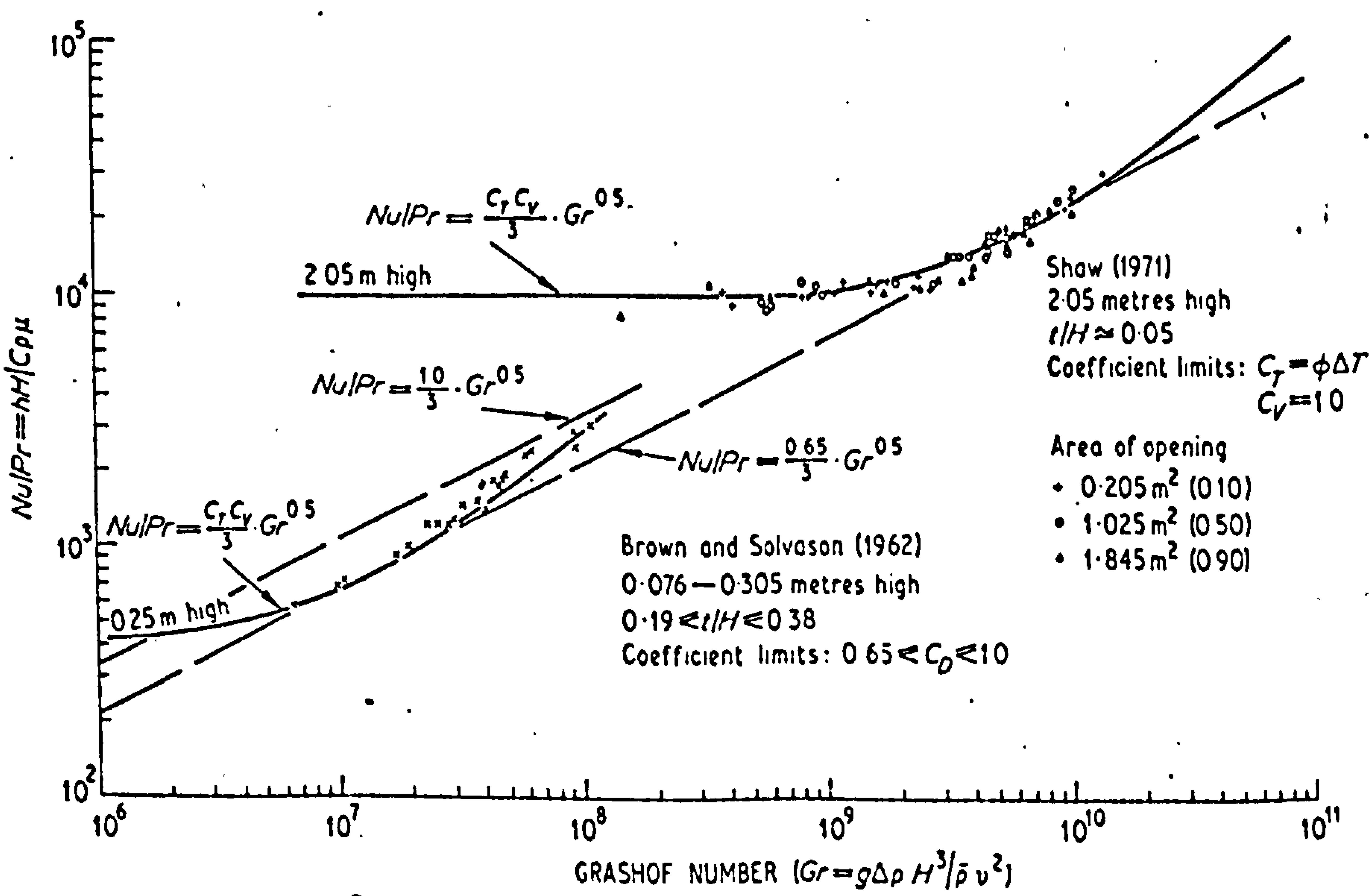


FIG. 4.14. NATURAL CONVECTION ACROSS A RECTANGULAR OPENING IN A VERTICAL PARTITION.

be seen, this is the point where the theoretical curve breaks away from the broken line (coefficient of 0.65). For further reference, at differential 40°C. $Gr = 6.76 \times 10^{10}$. Comparing now these values with the results of Brown and Solvason, it may be seen that a theoretical curve for an opening 0.25m in height correlates favourably with their results. It is now possible to verify the validity of the theoretical curves for higher differentials in the range 10-40°C. as this was the region in which Brown and Solvason were working. With this height of opening and a differential of 10°C., $Gr = 2.35 \times 10^7$ while at 40°C, $Gr = 1.23 \times 10^8$. This is in fact the break away region from the lower broken line and is similar to the upper curve. The two broken lines bounding the lower results are the limits of coefficient values as stated by Brown and Solvason (0.65 - 1.0).

The type of flow for these tests is considered to be turbulent. Also, since the Prandtl number for air in the range of temperature used in the tests was constant at 0.71, it was not possible to investigate its influence as a separate variable.

The theory for natural convection stated above is consistent with the approach of previous workers as can be seen from the list below. There are however two points which differ between sources, these being reference density and the use of a coefficient.

Source	
Brown and Solvason	$Q = C_D . \frac{W}{3} . H^{3/2} (g\,\Delta\rho/\bar{\rho})^{1/2}$
Graf. (1964)	$Q = \frac{W}{3} . H^{3/2} (g\,\Delta\rho/\bar{\rho})^{1/2}$
Tamm (1966)	$Q = \frac{W}{3} . H^{3/2} (g\,\Delta\rho/\rho_c)^{1/2}$
Fritzsche and Lilienblum (1968)	$Q = C_T . \frac{W}{3} . H^{3/2} (g\,\Delta\rho/\rho_c)^{1/2}$
Shaw (1971)	$Q = C_T . C_V . \frac{W}{3} . H^{3/2} (g\,\Delta\rho/\bar{\rho})^{1/2}$

The existence of an excess pressurisation coefficient C_V was not at first apparent as the coefficient had a value of unity for natural convection, i.e. a balanced ventilation scheme with no excess pressure.

4.4.2 Combined natural convection and forced air flow.

The leakage transfer volumes for combined natural convection and forced air flow are presented in Figure 4.15. Positive ventilation tests do not tend to be so consistent as the balanced tests and this may be due to the small samples which were taken. As stated previously the 1.20m and 1.40m wide door results are obviously erroneous, this being due to extraneous influences.

On the analysis of the positive tests in conjunction with the balanced tests, which may be regarded as positive tests with no excess supply pressure, it became evident that another coefficient did in fact exist. By dividing the actual inflow volume by the theoretical inflow volume (equation 4.16, without a coefficient of discharge) an overall coefficient was obtained, this being a function of a fictitious velocity over the area of the opening due to excess supply (Q_x/A) and the temperature differential.

$$\text{i.e. } C = \emptyset\,(Q_x/A, \Delta T)$$

where Q_x is equal to the supply volume minus the extract volume. The discharge coefficient, C, was defined as a product of the temperature coefficient, C_T, as obtained from the balanced tests, and a fictitious velocity coefficient, C_V.

$$\text{i.e. } C = C_T \times C_V$$

By dividing the left hand side of this equation by the temperature coefficient it was possible to find the values of the fictitious velocity coefficients. The value of this coefficient ranged from unity for a system with no excess volume decreasing as the amount of excess volume increased (Figure 4.16).

Experimental results for heat transfer are shown in Figure 4.17, the ordinate once again being Nu/Pr while the abscissa is the dimensionless group Sw. Broken lines in this graph represent temperature differential while full lines represent fictitious air velocity over the opening due to excess supply pressure (Q_x/A). As expected, the excess

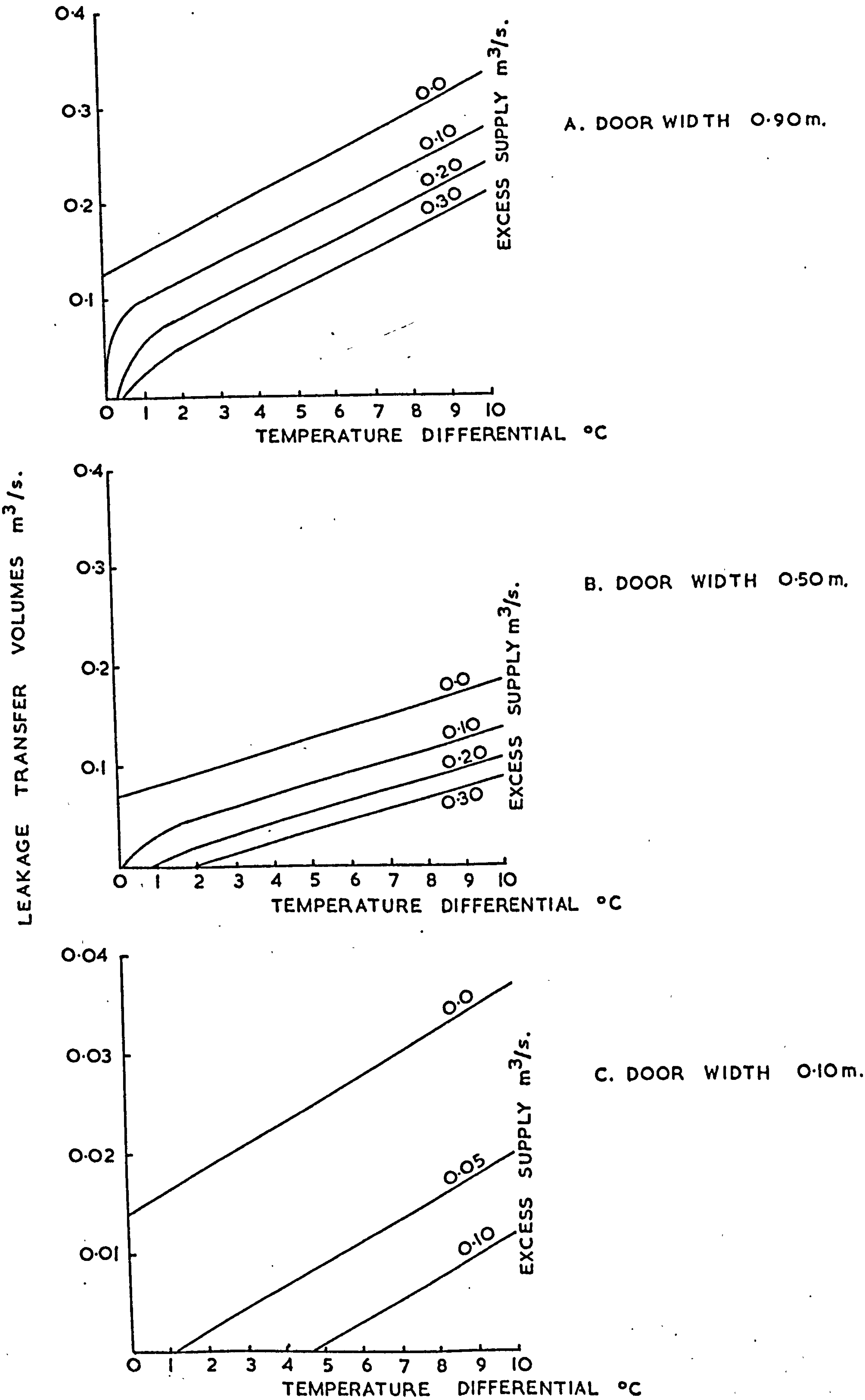


FIG. 4.15. LEAKAGE TRANSFER VOLUME FOR COMBINED NATURAL CONVECTION AND FORCED AIR FLOW.

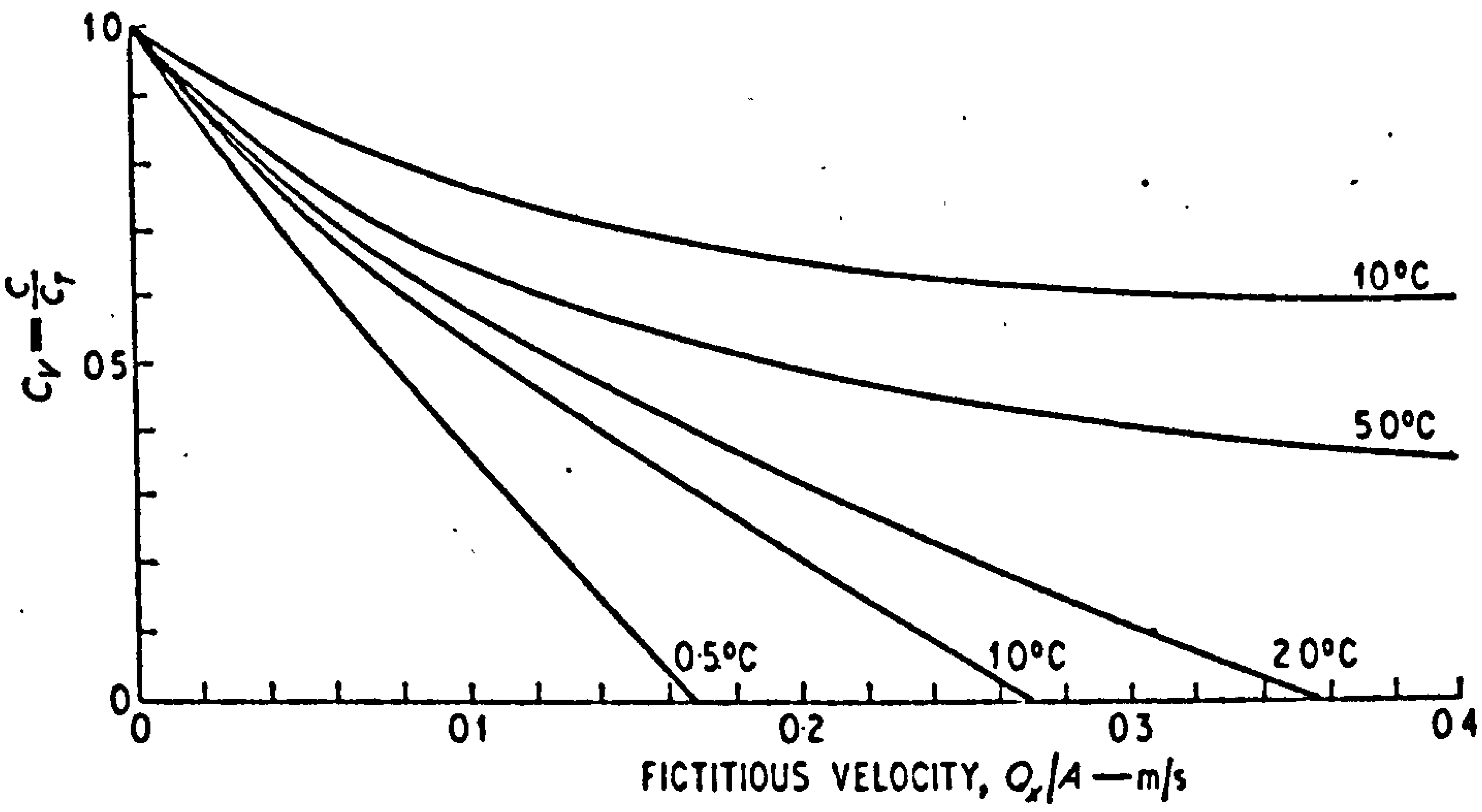


FIG.4.16. COEFFICIENT OF FICTITIOUS VELOCITY (C_V).

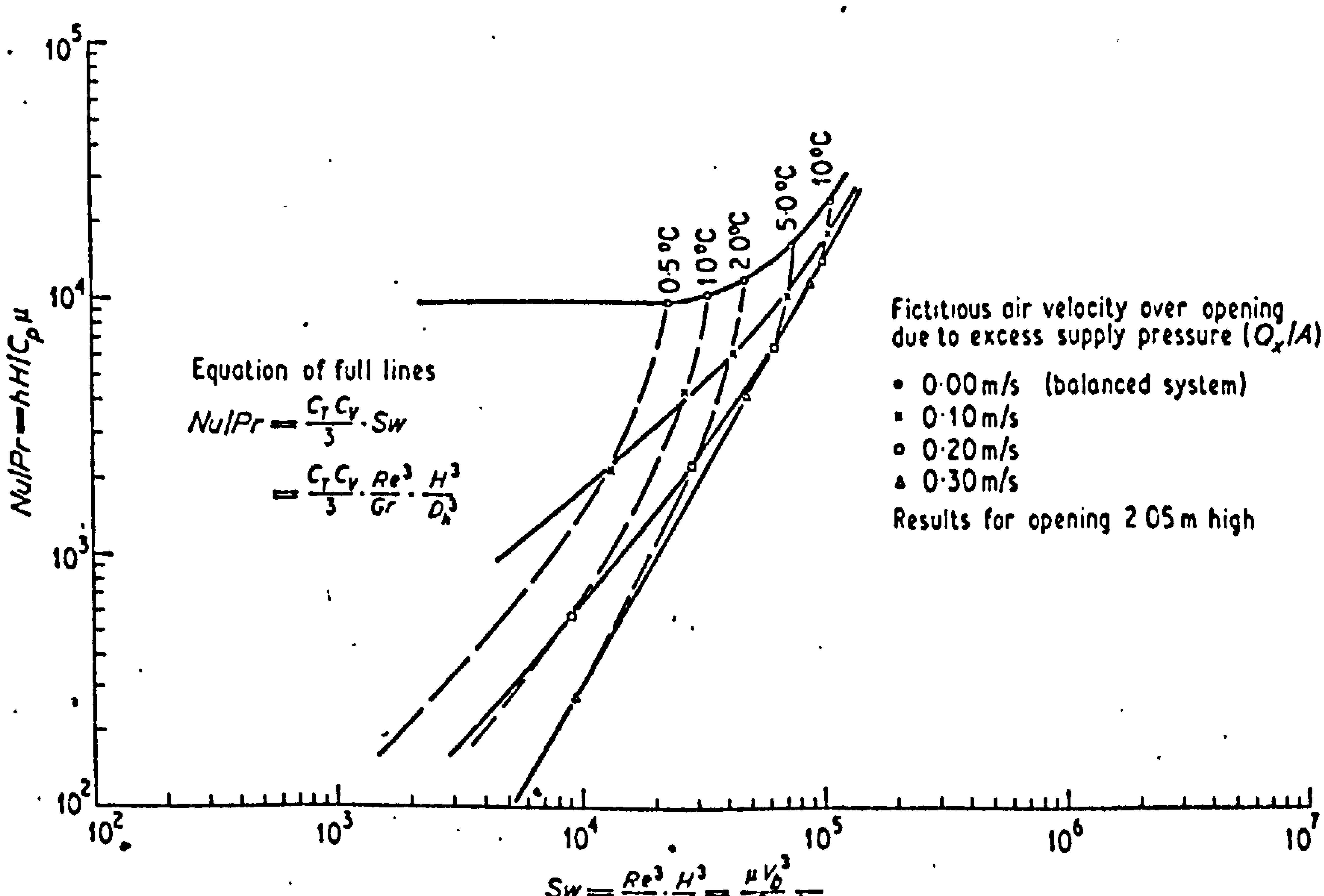


FIG. 4.17. COMBINED NATURAL CONVECTION AND FORCED AIR FLOW ACROSS A RECTANGULAR OPENING IN A VERTICAL PARTITION.

pressure reduces the heat transfer rate across the opening. Once again the type of flow is considered to be in the turbulent regime.

4.5 Discussion and Conclusion

This section has considered the volumetric exchange of air through an open doorway due to the temperature differential across it, both for balanced and positive ventilating systems. It has shown that even without any temperature differential across the opening (Table 4.2) there is still a considerable transfer of air, the volume depending on the dimensions of the opening.

It must be remembered that the theory is not exact for all conditions owing to neglect of viscosity, thermal conductivity and diffusivity in certain equations. However, as stated previously, if air is considered to be the convecting fluid over the tested temperature differential ranges then the theory may be used with confidence. The type of flow is considered in both cases to be turbulent.

Recent unpublished work by Ramsden, 1973, quotes exact theory for convective exchange flow through a rectangular opening in a vertical partition as follows:-

Natural Convection

$$Q = \frac{2}{3} K \, (2g)^{1/2} \, WH^{3/2} \left[\frac{\rho_2 - \rho_1}{\rho_2 \left[1 + (\rho_2/\rho_1)^{1/3}\right]^3} \right]^{1/2} \qquad (4.23)$$

This more rigorous treatment of the problem was necessary as he was considering a dense gas, Halon 1211, which had a gas to air ratio of up to 2.00. It may be seen from the table below that the density ratio for air is still fairly near unity, even at a high temperature differential of 40°C, and gives an error of only ± 8%. In practical cases the air temperature differential is likely to be a lot less than 40°C and could in many cases be as low as one or two degrees.

Temperature Difference ^oC	Density Ratio	Volumetric Exchange $\bar{\rho}$	ρ_1 or ρ_2	%
10	1.037	0.539	0.530	±1.5
20	1.080	0.780	0.753	±3.5
40	1.160	1.135	1.050	±8.0
Halon 1211	2.00	2.400	1.720	±40.0

The non-mean density volumetric exchanges presented above are worked out using the exact theory while the mean density values were worked out with the generalised theory. A coefficient of discharge has been ignored as this will be the same in both cases. There is no doubt that when dealing with high density ratios a more rigorous approach is essential as can be seen by the 40% difference, however the author considers the theory presented in this thesis to be sufficently accurate under the circumstances.

Similar work by Yao and Smith, 1968, has also recently been brought to the authors attention. This research and theory dealt with the convective mass exchange between a "freon" Fe 1301 - air mixture in an enclosure and surrounding air, through openings in a vertical wall. The dense gases, Halon 1211 and "Freon" Fe 1301, were both being used as fire extinguishing agents.

Since the characteristic dimensionless numbers for natural convection and forced convection may be taken as the Grashof and Reynolds numbers respectively it was assumed that the characteristic dimensionless group for combined natural convection and forced air flow would be a function of both Grashof and Reynolds number. This was indeed found to be the case, the relationship being in the ratio, Sw (a dimensionless group) = $\frac{Re^3}{Gr} \cdot \frac{H^3}{D_h{}^3}$, the other significant terms being the height and hydraulic diameter of the opening.

An interesting analogy to the subject of this chapter is the densimetric exchange flow of water in rectangular channels. A lock gate or other such division may separate bodies of still water of the same surface level but which differ slightly in density. This density difference may be due to either temperature or salinity differential. Barr (1963), in a paper on this subject, expressed this mechanism in terms of a densimetric Froude-Reynolds number, which is a criterion involving differential gravitational and viscous forces.

i.e.

$$Fr_\Delta . Re = \left[\frac{g \Delta \rho H^3}{\bar{\rho} \nu^2} \right]^{1/2} \qquad (4.24)$$

This is equivalent to natural convection with air as the convecting fluid and it is interesting to note that the right hand side of equation 4.24 is in fact the Grashof number ($Gr^{1/2}$) of equation 4.3 i.e. the characteristic dimensionless group. Barr does not introduce a forced flow on the natural exchange, but it may be noted that the dimensionless group for this mechanism may also be expressed in terms of the densimetric Froude number and Reynolds number.

i.e.

$$Sw = \frac{Re^3}{Gr} \cdot \frac{H^3}{D_h{}^3} = \left[\frac{\mu V_b{}^3}{\nu^2 g \Delta \rho} \right] = Re . Fr_\Delta{}^2 . \frac{H}{D_h} \qquad (4.25)$$

5. THE ISOLATION EFFECTIVENESS OF DOORS WHEN THEY ARE SHUT OR SLIGHTLY OPEN

5.1. Introduction

The primary reason for conducting these tests was to obtain figures for the area of crack round a shut door which could result in complete isolation under different air supply volumes and temperature differentials across the doorway.

It had been observed on visits to hospitals that pressure relief dampers, which were supposed to function as an air bypass when the door was closed, were lying shut. It was obvious that the cracks round the door would leak a substantial amount of air without resorting to the expense and inconvenience of a bypass damper. It would therefore be useful to know how much air could leak round a door without such a damper. With those doors where a high volume of air was passing round it, it was hoped to determine the width of door undercut, or area of door louvre, which would be necessary to allow free passage of air through the door, yet effecting complete isolation. It was necessary to consider these requirements in two parts. A - Hospital areas that are served by a single door, e.g. isolation and treatment rooms and B - Hospital areas served by multiple doors, e.g. operating rooms.

A second reason for those experiments was that general observations had shown that people do not always close doors properly and that they are often left slightly open. This posed a question - in that situation, is the area beyond the door being isolated, i.e. is the door effectively shut with regard to air transfer? It was necessary to determine at which width of opening the area beyond the door was effectively isolated, and knowing this width, it was possible to adjust door opening switches in the series of experiments which were carried out on door opening habits (Section 6).

5.2 Single Door Areas

5.2.1 Experimental Procedure

Experiments were carried out using the swing door of the isolation room mentioned in Section 4 and the sliding door of the treatment room of the Hairmyres Ward, measurements being made firstly of the change in static pressure over a closing and closed door, and secondly, the volume of air passing through it under these conditions.

The static pressure results are not included in this thesis but may be found elsewhere (Ward, 1970, Whyte and Shaw, 1972). Tests are described in Section 7 regarding the air transfer through doors being opened and closed. As part of this study Nitrous Oxide gas tests were run to determine the air transfer through a shut door due to natural convection. These are the tests that are reported in this section.

Shown in Figure 5.1 are the dimensions of the doors. The swing door of isolation room 1 was nominally 0.90m wide by 2.05m high (a standard hospital door size). There was a small door leaf which could be unbolted to give a full width of 1.20m. This was not normally used except when beds were passing through but this door leaf contributed very slightly to the crack area. The treatment room had a sliding door, the dimensions of which were 1.40m wide by 2.05m high. During the tests the door was only opened 0.90m wide in order to keep the same open door area as the swing door. As this door was hung on runners there was quite a substantial air gap at the top of the door and as it also had to run without touching the wall there was also a substantial gap on one side of the door when shut.

5.2.2 Results

The air leakage through these cracks under the influence of different temperature differentials is shown in Figure 5.2.

Also shown in Figure 5.2 is the theoretical air flow as calculated for the range of temperatures 0 to 8^{o}C by the following equation.

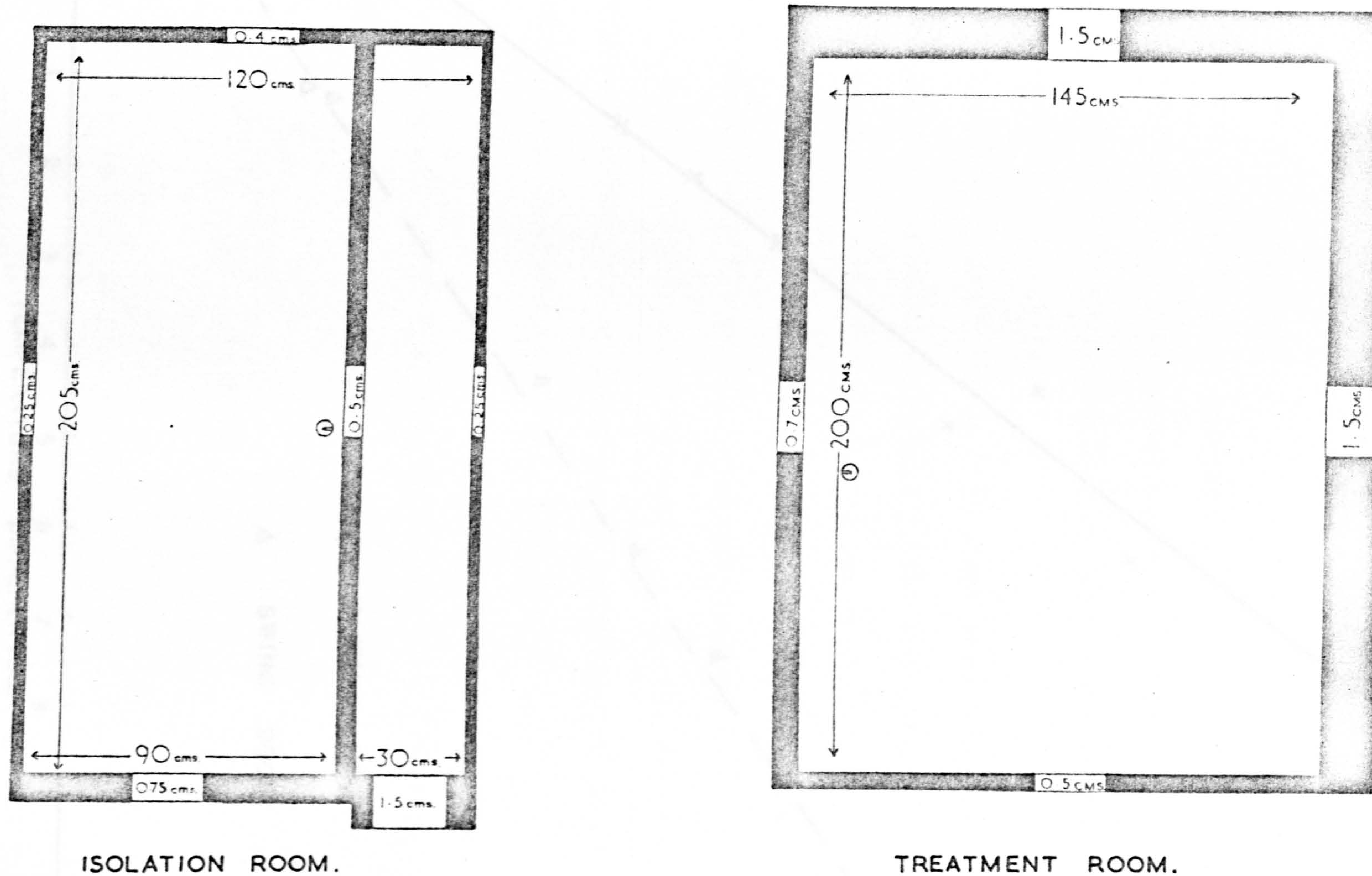


FIG. 5.1. OVERALL DIMENSIONS OF DOORS AND GAPS ROUND DOORS (not to scale).

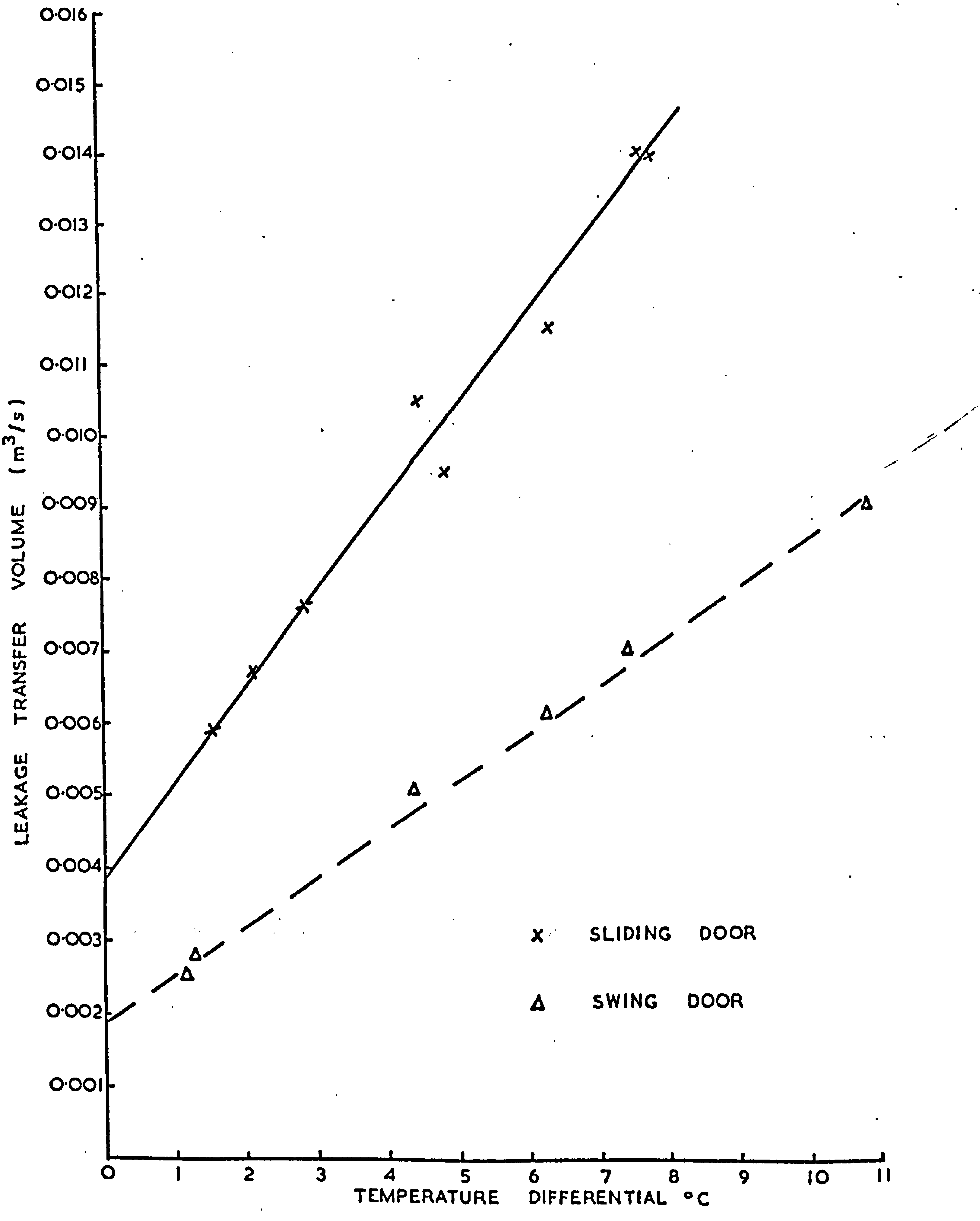


FIG.5.2. AIR LEAKAGE ACROSS CLOSED DOORS DUE TO NATURAL CONVECTION.

$$Q = C\frac{W}{3}\left[g\left(\frac{\Delta\rho}{\bar{\rho}}\right)\right]^{1/2} . H^{3/2} \qquad (5.1)$$

where

Q = airflow (m^3/s)

W = width of crack (m)

g = acceleration due to gravity (m/s^2)

$\Delta\rho$ = difference in density between rooms (kg/m^3)

$\bar{\rho}$ = average air density of the two rooms (kg/m^3)

H = height of crack (m)

The crack width (W) was taken as that width which when multiplied by the height door (H) would correspond to the crack area.

It may be seen in Figure 5.2 that the actual results compare well with the theoretical results especially at small temperature differentials.

Given in Table 5.1 are the pressures required to isolate a door opening against temperature differential. It should be noted that the pressure difference across a doorway is independent of width of opening although it is dependent on height.

These pressures are worked out in the following manner.

From equation 4.16 contained in Section 4 of this Thesis, a limit occurs when the portion in the bracket:-

$$2g.\frac{\Delta\rho}{\bar{\rho}}\cdot\frac{H}{2} - V_x^2 = 0$$

i.e. leakage through doorway is zero.

Now

$$2g\,\frac{\Delta\rho}{\bar{\rho}}\cdot\frac{H}{2} = V_x^2$$

$$\Delta\rho\frac{H}{2} = \frac{V_x^2\bar{\rho}}{2g} = P_V \text{ mm } H_2O$$

Knowing the temperature differential it is then possible to cal-

Table 5.1. Pressures required to isolate a door 2.05m high.

Temperature difference ΔT ^oC	Pressure required to isolate against temp. differential. mm H_2O
0.5	0.00205
1.0	0.00410
2.0	0.00820
3.0	0.01230
4.0	0.01640
5.0	0.02050

Table 5.2 Isolation of closed doors at a temperature differential of 4^oC.

Supply Volume m^3/s	0.05	0.10	0.15	0.20	0.25	0.30
Maximum crack area that may be isolated m^2	0.097	0.194	0.290	0.387	0.484	0.581

culate the velocity pressure through the doorway, $\Delta\rho\frac{H}{2}$. This is however the pressure required to isolate the doorway. Using the crack or door area and the following equation, the volume required to isolate is then determined.

$$Q = C \times A \times 4.43 \times \left(\frac{P_v}{\rho}\right)^{1/2} \qquad (5.2)$$

where A = Door crack area

P_v = Velocity pressure

From Section 6 in which studies are made of temperature differentials in hospitals it may be considered that 4°C would be the maximum temperature differential found between areas in air conditioned buildings. Using 4°C as the maximum condition it may be seen from Table 5.1 that the pressure required to isolate any door would be in the region of 0.0164mm w.g. Using equation 5.2, the following Table 5.2 can be constructed to show the area of crack opening which would create the correct pressure to isolate a door against a temperature differential of 4°C.

From the above table it may be seen that very low air volumes are required to isolate closed doors against differentials of pressure caused by temperature. A volume of 0.05m^3/s passing through the door cracks would be more than sufficient to isolate either of the two doors studied even though one of them had a very large crack area.

In normal situations with single doors passing greater amounts of air than 0.05m^3/s the door undercut should be calculated by subtracting the crack area round the two sides and the top of the door from the area given in Table 5.2 for the required volume. This will give the undercut area. For example a 0.90 x 2.05m door with crack widths of 0.4cm at the top and both sides would require an undercut of approximately 30cms to give isolation, when 0.15m^3/s of air is supplied to the room. With these larger volumes of air however the undercut area may be greater than the acceptable for privacy and may be restricted to a smaller size. A 10cm undercut would cover most cases.

This simple calculation would save the necessity of fitting a pressure relief damper or prevent the situation where an unnecessary back pressure would cause a reduction of air from the fan. When for example $0.3m^3/s$ of air was passing through the swing door studied, 200 times more static pressure was maintained than was necessary to effect complete isolation.

5.3 Multiple Door Areas.

5.3.1 Introduction

The most important example of a multiple doored room is that of the operating suite. It is British practice that around $0.57m^3/s$ (1200 cfm) of air is supplied to an operating room, this being the amount regarded, at present, as sufficient to protect one double door when fully open. It is further suggested that it is impracticable to supply enough air to protect all the doors in the suite. This is sensible in view of the very low chance of coincidence usage. The supply air finds its way out of the operating room through one or two low-level balanced flap outlets and under the doors. Given in Figure 5.3 is a typical airflow diagram as published in the Hospital Building Note No. 26 (D.H.S.S.).

As opposed to what has been suggested above for single door rooms it would seem best to keep the pressure drop across each door in the room as high as is practical, in order that when one door is opened the high pressure drop across the other door should force air out through the opened door. Blowers & Crew (1960) quoted a figure of 2.5mm w.g. (0.1 in. w.g.) as the pressure required to isolate operating room doors. This figure however is now considered to be the practical maximum to avoid build up of back pressure, whistling and door operating problems.

We know now that single doors with dimensions not less than the above (0.90 x 2.05m) could pass at the correct pressure drop, volumes of air up to $0.30m^3/s$ without necessarily resorting to a bypass damper, grilles or undercutting. If the set up as shown in Figure 5.3 is taken as typical, with flow out of the theatre through 2 double and 2 single

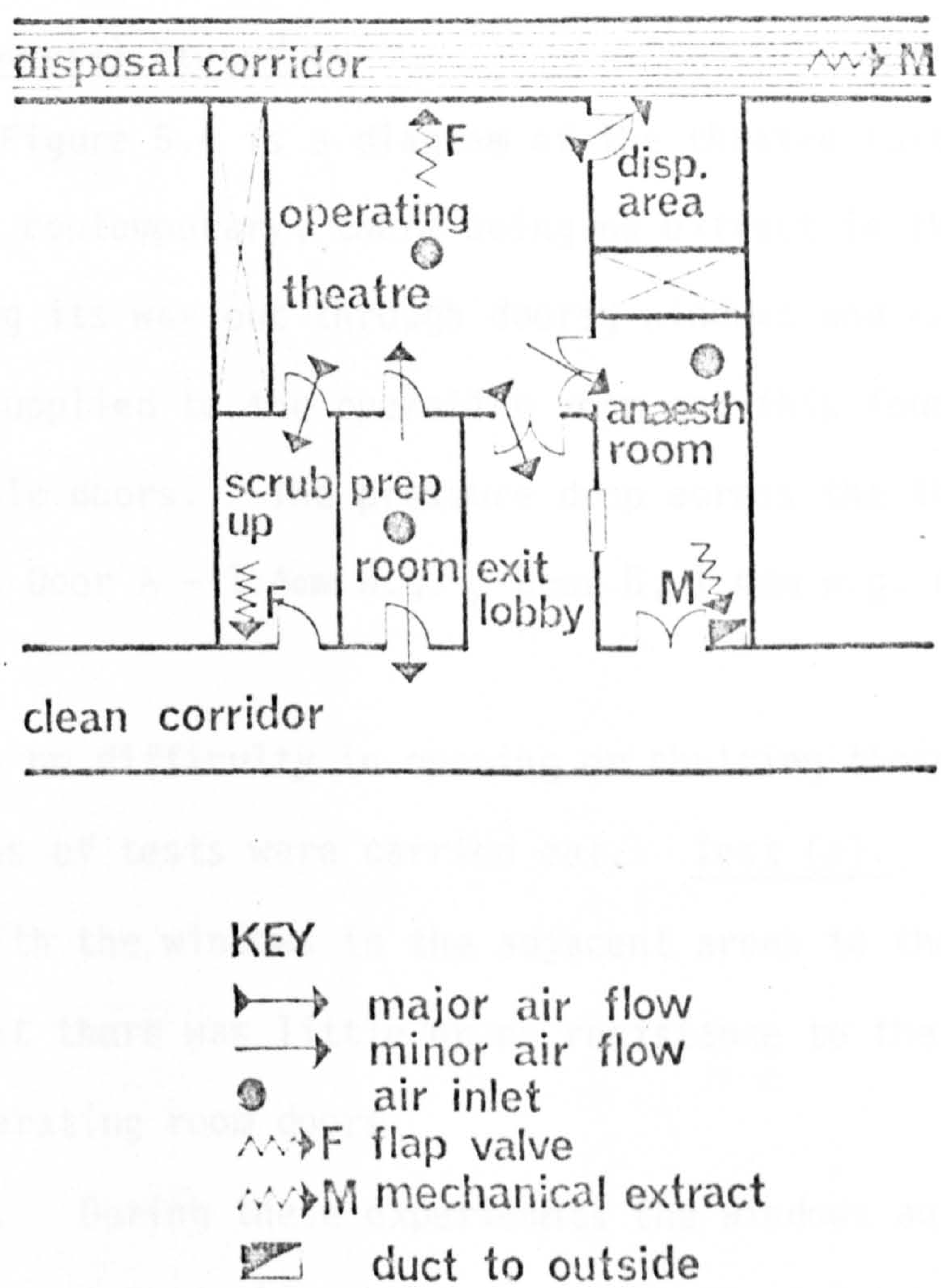


FIG. 5.3. AIR FLOW IN A TYPICAL THEATRE SUITE.

doors, it is probable that these doors will pass, without undue pressure, the air supplied to the theatre. There would be no need to fit pressure relief dampers in the operating room. In order to test these and other ideas the following situation was studied.

5.3.2 Experimental Procedure

Shown in Figure 5.4 is a diagram of the theatre suite studied. The design was not contemporary, there being no extract in the ancillary rooms, the air finding its way out through doors, windows and cracks.

Air was supplied to the operating room and this found its way through the three double doors. The pressure drop across the three operating room doors was Door A - 1.4mm w.g. , Door B, 1.4mm w.g. and Door C - 1.2mm w.g.

There was no difficulty in opening or shutting these doors.

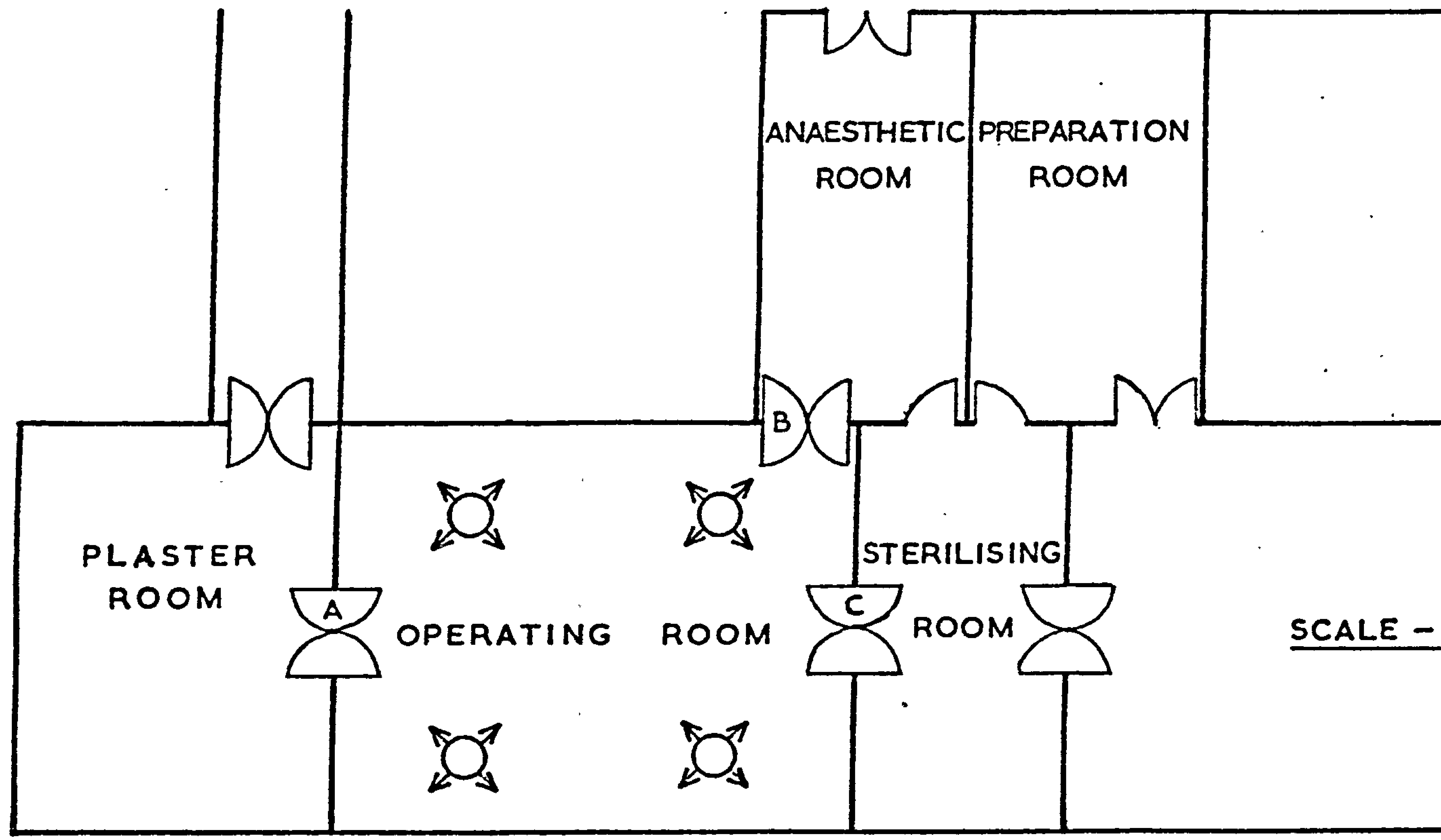
Two series of tests were carried out. Test (a). This set was carried out with the windows in the adjacent areas to the theatre open. This meant that there was little or no resistance to the air flowing out of the operating room doors.

Test (b). During these experiemnts the windows and doors were shut. This meant that when a door was open the air had to find its way out past cracks in the doors and windows. The hatch in the preparation room was opened however.

5.3.3 Results

(1) Shown in Table 5.3 are the pressure drops over two of the operating room doors when the other door was opened. It may be seen that there was little or no resistance to the flow out through the open door and the pressure drop over the other two doors dropped significantly although it did not reach zero.

It is shown in Table 5.4 that where there was no easy way for the air to get out, the air flow through the door will not increase significantly. It is only when the air may escape easily, in the tests through



SCALE – 1cm. to 5ft.

FIG. 5.4. PLAN OF OPERATING SUITE.

Table 5.3 Pressure drops over operating room door - little resistance.

Door & ΔP when all doors closed	ΔP when door opened		
	Door A	Door B	Door C
A 1.35	-	0.15	0.10
B 1.30	0.20	-	0.30
C 1.20	0.02	0.02	-

ΔP is difference in pressure, mm w.g.

Table 5.4 Pressure drops over operating doors - resistance

Door & ΔP when all doors closed	ΔP when door opened		
	Door A	Door B	Door C
A 1.20	-	1.20	0.35
B 1.20	0.35	-	0.125
C 1.40	0.70	1.25	-

ΔP is difference in pressure, mm w.g.

the open hatch, that the air flow will increase through that door.

5.4 Discussion and Conclusions.

It has been shown in Table 5.2 that when $0.10m^3/s$ to $0.20m^3/s$ of air is passing through a doorway, an opening of around 10cm would be effectively isolated as long as the temperature difference was not greater than 4^oC. Taking into consideration the crack area it was therefore decided that in the field tests, as shall be described in Section 6 that a door that was within 10cm of being fully shut, would for the purpose of the experiment be considered 'shut'. The door microswitches for these series of experiments were duly adjusted to give such conditions.

It is also hoped that the results obtained for pressure drops across doors in single and multiple door areas will be useful as design data in the analysis of air flow through buildings due to temperature differentials and mechanical ventilation.

General conclusions and design recommendations relating to this section are dealt with in detail in Section 8 both for single door and multiple door areas.

6. FIELD TESTS TO DETERMINE DOOR OPENING HABITS AND TYPICAL TEMPERATURE DIFFERENTIALS.

6.1 Introduction

In the previous section of this thesis results were presented which enabled the following to be determined.

1) The amount of air transferred through an open doorway both in the case of flow due to natural convection, and to the combined effect of natural convection and a forced supply of air passing through the doorway. (Section 4).

2) The isolation effectiveness of doors when they are shut or slightly open (Section 5).

These above tests are however dependent on either the temperature difference across the door or the number of times the door is used or they are dependent on both factors. It was therefore necessary to relate the results obtained in these sections to the actual hospital situation in order that their full relevance could be obtained. Towards this end, tests were carried out to establish the actual temperature differences and the door opening habits in existing hospital areas. These results are contained in this section.

Six hospitals were selected,and seven isolation rooms, three treatment rooms and one operating suite were studied. The door usage habits and temperature difference between these rooms and adjacent areas were studied over a few days both during the summer and winter periods. The areas studied in hospitals were chosen to give a wide and representative sample of various types of rooms. All hospitals were in Scotland, the majority being close to Glasgow. A full description of the various areas studied and the results obtained may be found elsewhere (Ward 1970, Whyte and Shaw, 1972). The salient points of these reports relating to the field tests are presented below.

6.2 TEMPERATURE DIFFERENTIALS

6.2.1 Experimental Procedure

Temperatures were measured by calibrated thermocouples. These were attached to both the wall of the room being studied and the wall of the area outside at a height equivalent to the centre of the door. They were placed in such a manner so as to be close to the doorway but away from the influence of the air flowing through the door. The temperature difference across the doorway was determined from these mid-point temperatures, the reason for using this temperature being that it was the most convenient in the circumstance.

A Honeywell 20 point recorder was used for recording the temperature values, the temperature being automatically recorded every 10 - minutes. In general, summer and winter readings were taken and, when convenient, outside and supply air temperature.

6.2.2 Results

Hourly readings of temperature difference were calculated from the readings taken both inside and outside the room studied. Given in Table 6.1 are the temperature differences found between various critical rooms and their adjacent areas. These differences are expressed as the median average taken over a 48-hr test and the maximum and minimum differences found over the same period. Temperature differences which are positive indicate that the room studied was at high temperature than the adjacent area; temperatures which are negative show the opposite.

From the results on door usage reported further on in this section it should be noted that disregarding the time the door was being used, (i) the isolation room doors were always shut, (ii) the treatment room doors were normally open and (iii) in the case of the only operating room considered the door to the clean corridor was always shut but there was no door between the theatre and the sterilizing room.

6.2.3 Discussion and Conclusions

A variety of temperature differentials, from $0^{o}C$ to $4^{o}C$, were found

TABLE 6.1

TEMPERATURE DIFFERENTIALS

Hospital	Area	Temp. diff. $\Delta T^{o}C$	Month
A	I.R. 1	1.81	August
	I.R. 2	2.26	August
	I.R. 1	-0.97	December
	I.R. 2	-0.56	December
B	I.R. (clean)	0.20	March
	I.R. (clean)	0.23	March
	I.R. (dirty)	-2.09	March
	I.R. (dirty)	-3.63	March
C	I.R.	0.41	August
E	T.R. 1	0.38	July
	T.R. 2	0.98	July
	T.R. 1	0.15	August
	T.R. 2	0.72	August
	T.R. 2	0.60	March
F	T.R.	-1.10	August
G	O.T. (clean)	-3.87	September
	O.T. (dirty)	1.15	September

I.R. - Isolation Room: T.R. - Treatment Room: O.T. - Operating Theatre

Negative readings indicate that room was cooler than adjacent area.

in the areas studied. Although insufficient areas were studied in great enough depth to be able to indicate with certainty the maximum differences which could be found in all areas and under all circumstances, the results obtained indicate that -

a) In well designed conditions i.e. internal rooms with common heater battery system, average temperature differentials of $0.5^{o}C$, with a maximum of around $1^{o}C$ should be expected.

Diversions from this may be expected by:-

b) Air supplies to adjacent areas being heated by separate heater batteries causing temperature difference i.e. zoning difficulties; consideration should therefore be given to the necessity of such systems or the possible type of better control of such systems e.g. by dual duct systems. The stipulation of temperature differences in Building Notes, e.g. treatment room temperatures are set at $5^{o}C$ greater than the rest of the ward, should be considered.

c) Solar gain can cause differences in temperature. Consideration should be given to this problem.

d) In non-ventilated situations ΔT maximum were found of $4.5^{o}C$ in isolation rooms and $5.2^{o}C$ in the case of the O.R. (between the sterilizer room and the theatre).

It is therefore concluded that in a ventilation system where care has been taken in minimising temperature differences a difference of $0.5^{o}C$ would be common with a maximum of $1^{o}C$. However one must be prepared for up to $4^{o}C$ in a poorly designed system.

6.3 Door Usage.

6.3.1 Experimental Procedure

Door monitoring was achieved by the making and breaking of an electrical circuit by either micro or reed switch attached to the door lintel. The switches were activated by the opening and closing of the door and the resulting change in electrical current was recorded by a "Record" pen recorder. In some hospitals, two or three were monitored simultaneously, this being achieved by using the "Record" recorder in conjunction with a resistive electrical network, the resistance varying with the number of doors open.

General observations have shown that people do not always close doors properly and they are often left slightly open. This posed the question - "In this situation, when is a door effectively open or closed with regards to air transfer?".

It has been demonstrated in Section 5 that when the air flow rate through the door is greater than $0.1 m^3/s$ an opening of 10cms would not reduce the isolation efficiency of the door. The door opening switches were therefore adjusted to allow for an opening of 10 cms - a door 10cms from its fully shut position being regarded for the circumstances of this test as "shut". This meant that the readings would be fully relevant to the conditions investigated i.e. how often would an isolation or treatment room be challenged by outside contaminated air and would this be of significance? The results were monitored continuously over a five or six day period during the summer and winter.

6.3.2 Results

The analysis was carried out in two parts, part (i) being results pertaining to isolation rooms and part (ii) being results pertaining to treatment rooms. This distinction is necessary as treatment rooms were used in a very different way from isolation rooms, the treatment rooms being used only for short periods of time during the day and normally left open the rest. The analysis of the treatment room doors was therefore carried out in a slightly different way.

6.3.3 Isolation Room Results

Tables of frequency of door opening over one hour periods were tabulated for a 5 - 6 day test period and mean number of openings established. Each hospital was analysed separately and the number of times a door was opened per hour was found. Given in Table 6.2 is a summary of these results. As the frequency of use of these rooms is dependent

Table 6.2

Door Usage in Isolation Rooms

Isolation Area	Room * No.	Test	Type of Patient	Median no. of openings per hour	Average time door open (Mode)
A	1	1	18-month old child with gastro-enteritis	2	7.4 secs.
	1	2	9-month old child under observation in incubator	1	
	2	1	3½-year child with gastro-enteritis	1	
	2	2	18-month child with salmonella infection	1	
B	1	1	Both patients attending for kidney dialysis	3	7.8 secs.
	2	2		4	
C		1	Both patients recovering from kidney transplant	6	10 secs.
		2		4	
D	1 (Single bed)		Adult suffering from bronchitis and gastro-enteritis	3	9 secs.
	2 (3-bedded)		Three sisters ages 2 - 4 with salmonella	4	
	3 (2-bedded)		Two boys 8 years and 10 years with gastro-enteritis	5	

Each test period extends over 5-6 days

*All single bedded except where indicated.

on the degree of nursing care, and therefore the type of patients these are also indicated in Table 6.2. Finally, the modal time of opening of the door is given. These values were calculated by averaging a series of readings but excluding the occasional long period of time when the door was opened and not shut for over a minute (about 10% of the openings were of long duration of between 5 - 10 mins). These long openings were excluded as it had been shown in Section 4 of this thesis that it was vital that isolation room doors should have self-closers to prevent the transfer of air through the open door. This would therefore mean that these long door openings would not occur in future installations. Isolation Rooms, A, B and C did not have self-closing mechanism, however Room D did.

6.3.4 Treatment Room Results.

As mentioned previously, treatment rooms were used in a different way to isolation rooms. Instead of being continually used over 24 hours, they were used intensively over one or possibly two sessions a day plus an occasional patient for a few minutes. These sessions amounted usually to not more than a couple of hours usage of the treatment rooms. For this reason the results of door usage were analysed only at these periods and not over 24 hours. These results are shown in Table 6.3. These tests were carried out in three treatment rooms in two hospitals.

From this Table it may be seen that the wound dressing and inspection sessions varied in time from 14 minutes to 2 hours 18 minutes. The door was kept open for a large part of this time and it could be as long as half the time of use. However during this period when the door was opened it was never observed that a patient was being treated. This time when the door was open was almost exclusively due to the time between patients, which could be less than a minute although could extend up to 20 minutes; presumably when a patient could not be found or the nurse was distracted from her duties. Because of this the results of the number of door openings per hour have been presented, not only as the

Table 6.3: Treatment room door usage.

Hospital A Test No.	Total Time used (mins)	% Time open	No. of times open/hour	No. times open/hr. during active use
1	28	39	15	21
2	16	32	19	25
3	49	51	4	8
4	53	40	11	18
5	46	15	8	9
6	14	2	9	9
7	24	30	10	13
8	40	35	12	18
Hospital B Test No.				
1	53	23	18	20
2	54	51	16	35
3	41	32	6	9
4	72	17	27	32
5	138	15	23	31
6	84	7	18	21
7	40	24	14	16

number of openings per hour during a treatment session but as the number when the room was actively being used, i.e. when a patient was present. These openings per hour varied from 8 to 35.

6.3.5 Conclusions

It was found in general that the isolation room doors were opened on an average of between 1 and 6 times per hour. The maximum number of times a door was opened during one hour was found to be 10 whereas the minimum was 0.

The determination of the length of time a door was open is of a dubious nature, as it is dependent on so many variables. For example, the time taken to open a door, enter, and close the door was found to be approximately 6 seconds while in the hospitals tested the time ranged from 7 to 10 seconds with a mean of 8.5seconds. The variation in the results may be accounted for by the fact that some doors were fitted with automatic closers which tended to increase the time taken for the door to close. However, these results compare well with those obtained by Dr. Lidwell (personal communication) who arrived at a mean value of 6.4 seconds. It was also noted in the isolation room tests that approximately 10% of the openings were of a longer duration, between 5 and 10 minutes. This is of sufficient length to allow such factors as temperature and pressure differences to influence the air transfer across the door to a significant level and for this reason self closing mechanisms should be fitted to isolation room doors.

From the analysis of the treatment room results, it was found that the temperature differences were of the same order as those found in isolation rooms, but that the door usage patterns were completely different. As mentioned before, treatment rooms were only used during certain parts of the day. This usage amounted to one or two sessions a day plus perhaps the occasional single patient, giving a total usage of approximately two hours. From the analysis of the door openings when patients were in the rooms, it was found that the number of door actions per hour varied from 8 to 35.

The relevance of these studies of door usage and temperature differentials in hospital areas will be fully discussed,and the results applied,in other sections of this thesis.

Note:- The experimental work in this section was carried out by an undergraduate at the University of Strathclyde (Ward, 1970) as part of a final year project, the tests being performed under the direction of Mr. W. Whyte and the author.

7. AIR MOVEMENT THROUGH DOORWAYS DURING USAGE.

7.1 The Decay Equation, with reference to door opening habits in a ventilated room.

7.1.1 Efficiency

The equations derived are in terms of the rate at which the isolation efficiency decays or grows in a ventilated room under the influence of door opening habits.

Consider a room, as shown in Fig. 7.1 having a ventilation supply volume of Q_s. With the door closed, the efficiency in the room should theoretically be 100% and is calculated as follows:-

$$\eta = \left(1 - \frac{Q}{Q + Q_s}\right) \times 100 \ \%$$

where Q is the inflow volume to the room due to natural convection, i.e. temperature differential between the rooms. Since Q = 0 in this case the efficiency would be 100%. This condition is to be taken as the initial state.

Two alternatives are now to be considered. These are -

1) the door to the room is opened with no ventilation supply to to the room. With these conditions the efficiency of the room will eventually reach zero, and

2) with the room efficiency at zero and the door still open, the ventilation plant is started up. With this new set of conditions the room efficiency will eventually reach equilibrium state, i.e.

$$\eta = \left(1 - \frac{Q}{Q + Q_s}\right) \times 100 \ \%$$

Consider the first set of conditions. During time δt, a small volume of air δq will flow out due to natural convection. A similar volume of "contaminated" air will flow in. The word "contaminated" in this text implies either bacteriological contamination or enthalpy transfer (positive or negative). The efficiency within the room is

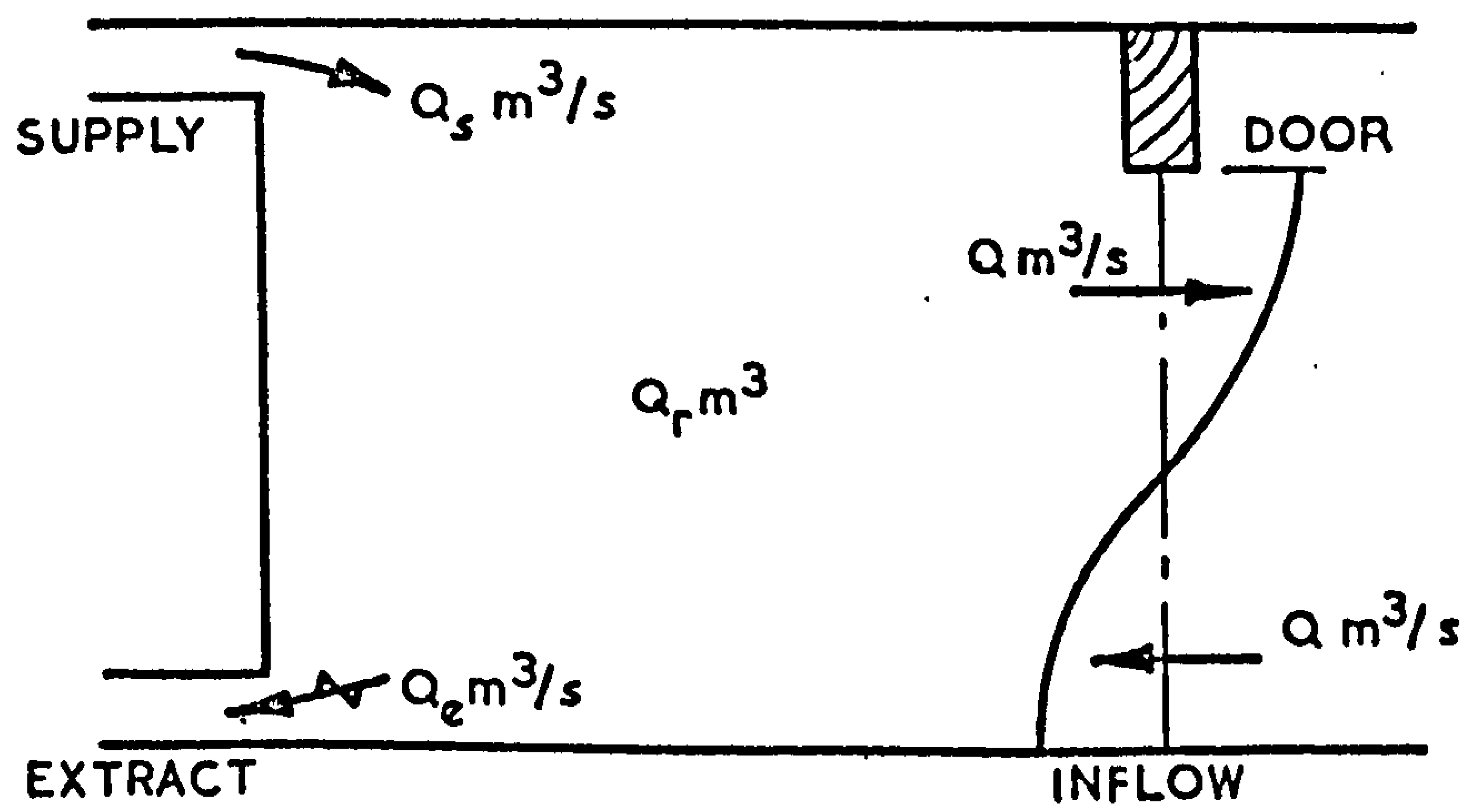


FIG. 7.1. VOLUME BALANCE IN A VENTILATED ROOM.

therefore reduced by an amount $\frac{\delta q}{Q_r}\eta$ where Q_r is the volume of the room.

This reduction in efficiency can be expressed by $\delta\eta$ defined as follows:

$$\delta\eta = -\frac{\delta q}{Q_r}\eta \tag{7.1}$$

The negative sign is used because the efficiency is reducing.

It follows that the rate of change of efficiency is given by $\delta\eta/\delta t$, defined as follows:

$$\frac{\delta\eta}{\delta t} = -\frac{\eta}{Q_r}\cdot\frac{\delta q}{\delta t} \tag{7.2}$$

$\delta q/\delta t$ is the rate of influx of contaminated air and may be considered constant at Q say. Thus the rate of change of the efficiency with respect to time may be written as

$$\frac{d\eta}{dt} = -\frac{\eta Q}{Q_r} \tag{7.3}$$

The physical problem has now been phrased as a simple differential equation, and a solution to this will be of practical value in determining the answers to real problems.

By integration the solution to eqn. (7.3) is

$$\int\frac{d\eta}{\eta} = -\frac{Q}{Q_r}\int dt$$

$$\log_e\eta = -\frac{Qt}{Q_r} + \log_e A$$

where $\log_e A$ is a constant of integration.

Hence,

$$\log\eta - \log_e A = -\frac{Qt_1}{Q_r}$$

and

$$A\exp\left(-\frac{Qt_1}{Q_r}\right) = \eta$$

The value of the constant A is established by considering the boundary condition $\eta = \eta_0$ (the initial efficiency in the room) at t = 0 (the instant that the door is opened).

The solution to equ.(7.3) is therefore

$$\eta = \eta_0 \exp\left(-\frac{Qt_1}{Qr}\right) \qquad (7.4)$$

The graph of equ.(7.4) is an exponential curve, as shown in Fig.7.2.

Consider now the second set of conditions with the efficiency in the room at zero, and the air conditioning plant started up. The efficiency will increase exponentially towards the equilibrium state of the system. This curve is represented by the equation

$$\eta = \left[\left(1 - \frac{Q}{Q+Q_s}\right)\left(1 - \exp\left(-\frac{Qt_1}{Qr}\right)\right)\right] \times 100 \qquad (7.5)$$

and is shown in Fig. 7.2.

Combining equations (7.4) and (7.5) we obtain

$$\eta = \left[\left(1 - \frac{Q}{Q+Q_s}\right)\left(1 - \exp\left(-\frac{Qt_1}{Q_r}\right)\right) \times 100 + \exp\left(-\frac{Qt_1}{Qr}\right)\right] \qquad (7.6)$$

which represents the combined effects of the two sets of conditions, i.e. natural convection volume transfer and supply ventilation to the room. This process is represented by the third curve on Fig. 7.2. Equation (7.6) therefore represents the decay in efficiency in a room under these conditions.

The growth of efficiency under the same set of conditions, i.e. the door now closed with no transfer of contaminated air, may be represented by the curve on Fig. 7.3 and may be expressed mathematically by the following

$$\eta = \left\{\eta_c + \left[\left(1 - \exp\left(-1 + \frac{Q_c - Q_s t_2}{Q_c}\right)\right)\left(1 - \eta_c\right)\right]\right\} \times 100 \qquad (7.7)$$

where η_c is the efficiency when the door is closed, Qc is the volume of contaminant in the room at that time and may be calculated from $\eta_c = \frac{Q_c}{Q_r}$ or Qc = $\eta_c \times Q_r$ and t_2 is the time in appropriate units commencing from

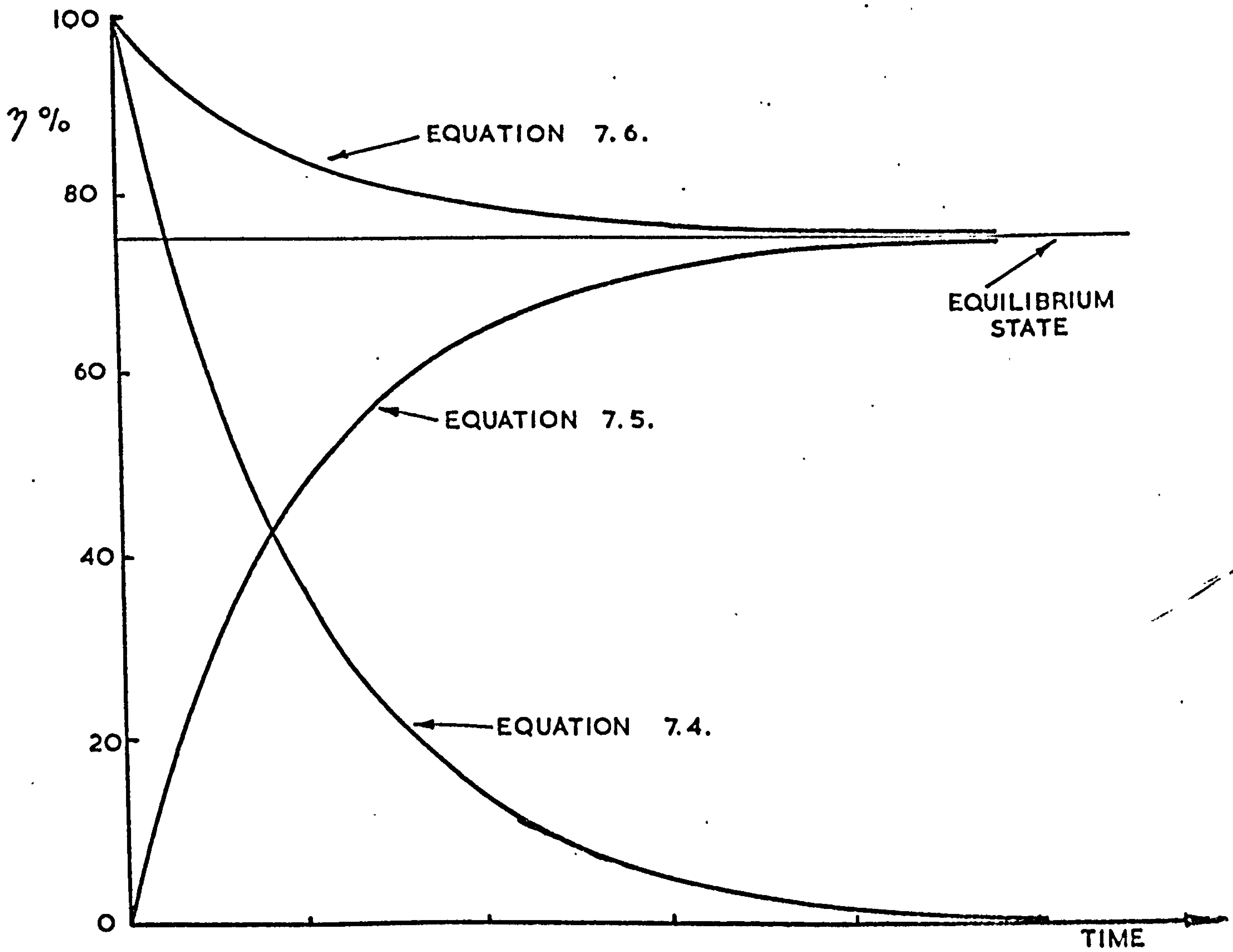


FIG. 7.2. EXPONENTIAL DECAY AND GROWTH CURVES.

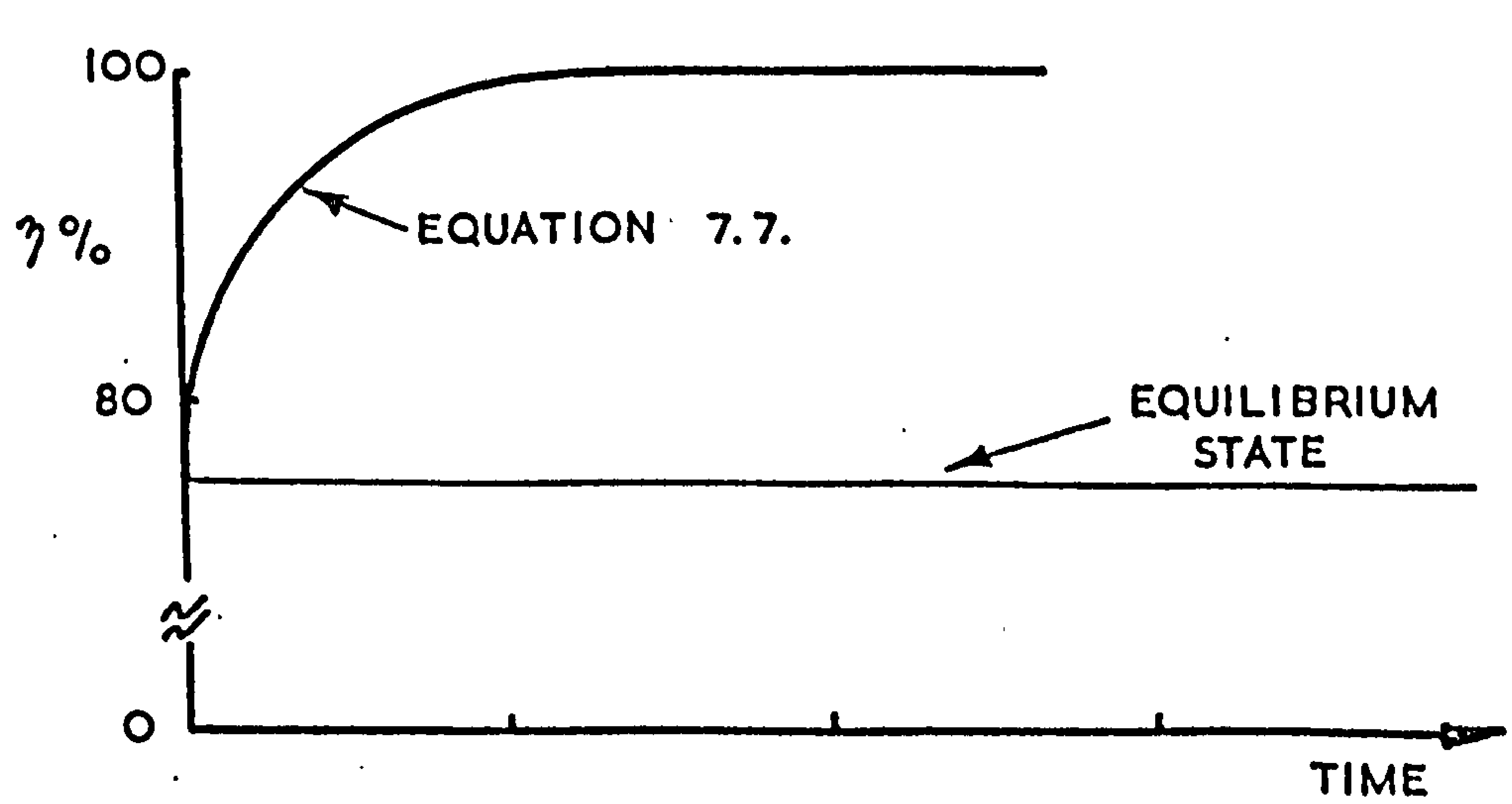


FIG. 7.3. EXPONENTIAL GROWTH CURVE.

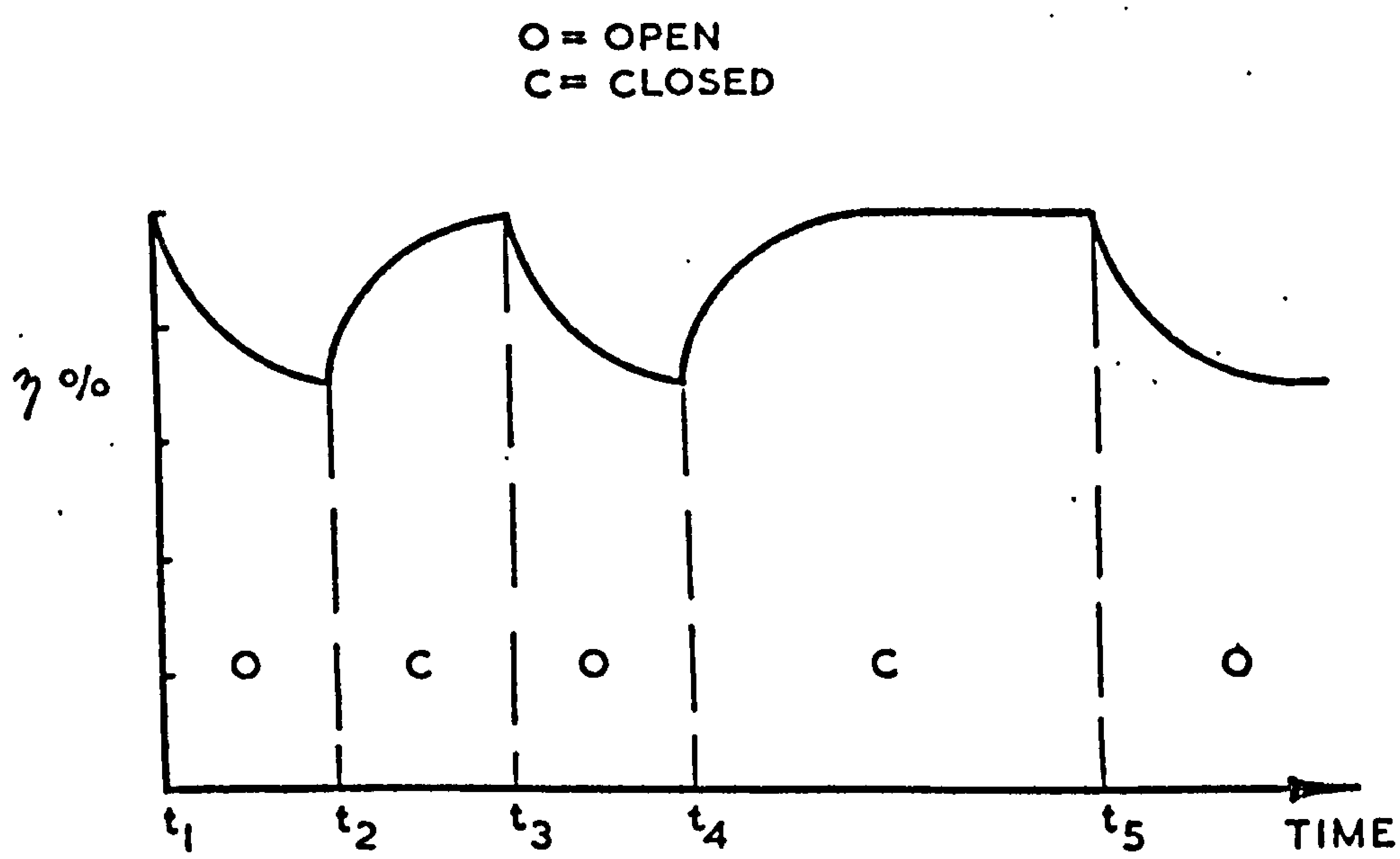


FIG. 7.4. EXPONENTIAL SAW-TOOTH WAVE FUNCTION.

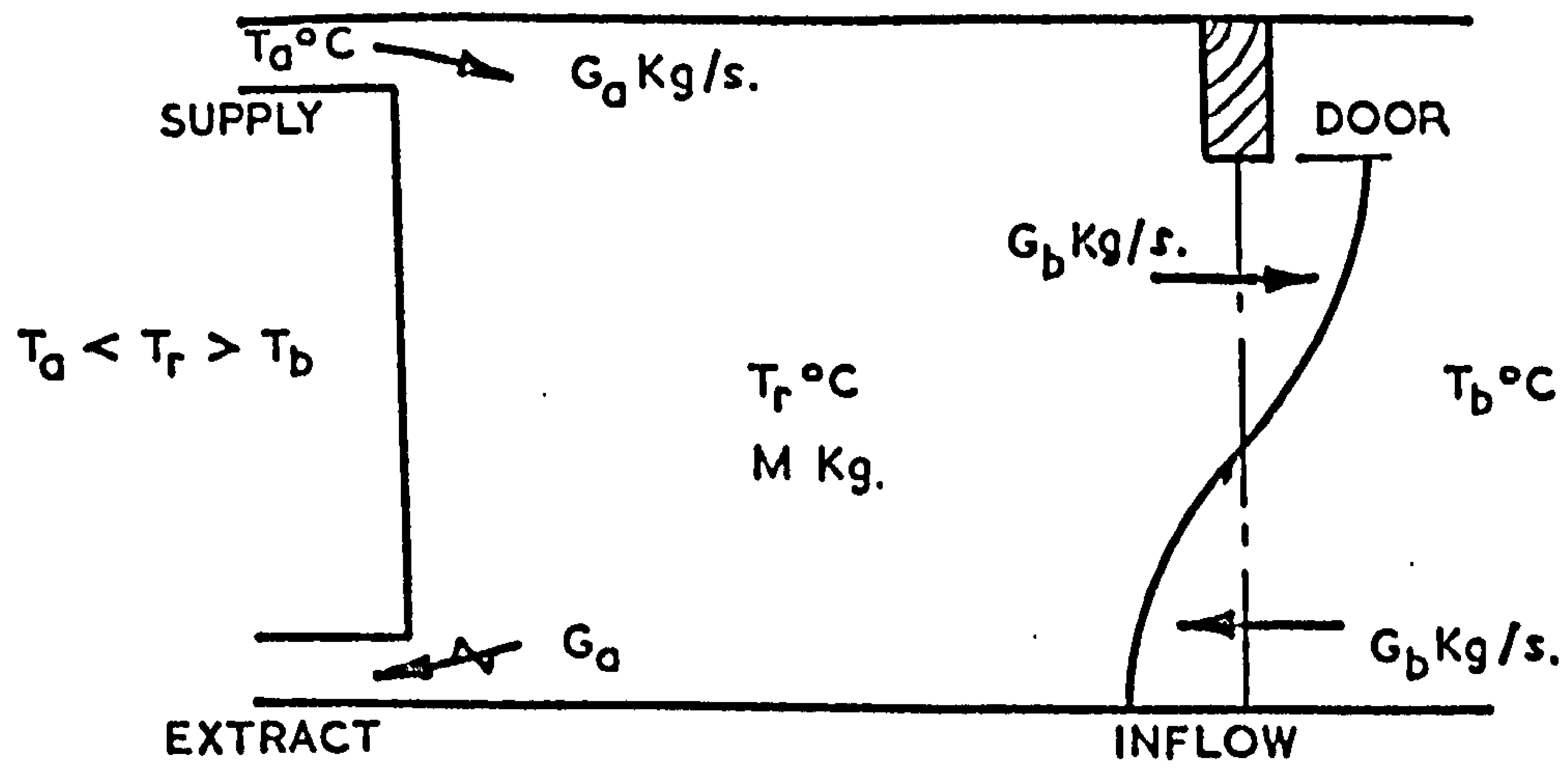


FIG. 7.5. MASS BALANCE IN A VENTILATED ROOM.

the instant the door is closed.

From these decay and growth equations it is therefore possible to plot the change of efficiency within a room due to door opening habits. The curve obtained from such a plot would be of the form of an exponential saw-tooth wave function, the shape depending on the door opening habits. It is also possible to obtain the efficiency within a room at any instant knowing a limited amount of information.

a) volume of the room

b) temperature differential

c) door area

d) supply ventilation volume to room

e) door opening habits, i.e. time when door is opened or closed.

7.1.2 Enthalpy

The decay equations have so far been expressed in terms of change in efficiency within a room. Let us now consider a change in terms of enthalpy. This problem is approached in a similar manner to that of efficiency and the basic theory and equations may be found in a paper by Jones (1963).

Consider firstly, the simple case of a room the air within which has a certain heat content, but to which no heat gain occurs and, assume further, that the air supplied to the room has no heat content.

If a mass of air δg_a is supplied to the room, then a similar mass δg_a must be displaced from it Also if the mass of air within the room is M, then the fraction displaced is $\delta g_a/M$. Clearly the heat content within the room must have reduced by an amount given by the equation:

$$\delta s = -\frac{\delta g_a S}{M} \text{ in time } \delta t \qquad (7.8)$$

Consider next the complication resulting from the fact that the air supplied to the room has a heat content, h_a. The heat content to the room by virtue of the mass flow of supply air is then given by

$$\delta s = h_a G_a \, \delta t \quad \text{in time } \delta t \tag{7.9}$$

Consider also the case of a temperature differential across the open door to the room. Due to variation in air density there will be convective flow in and out of the doorway. This will also influence the heat balance within the room and two similar equations to those above may be included, i.e.

$$\delta s = -\delta g_b \, S/M \tag{7.10}$$

$$\delta s = h_b G_b \, \delta t \tag{7.11}$$

Following upon this, take account of the presence of a heat gain to the conditioned space from both internal and external sources. This causes an increase of heat content within the room of:

$$\delta s = H(t) \left[G_a + G_b\right] \delta t \quad \text{in time } \delta t \tag{7.12}$$

Adding together these components which affect the heat content of the air in the conditioned space, one obtains an overall expression:

$$\delta s = h_a G_a \delta t + h_b G_b \delta t + H(t)\left[G_a + G_b\right]\delta t - \delta g_b \frac{S}{M} - \delta g_a \frac{S}{M}$$

Hence $$\frac{ds}{dt} = h_a G_a + h_b G_b + H(t)\left[G_a + G_b\right] - \frac{S\,dg_b}{M\,dt} - \frac{S\,dg_a}{M\,dt} \tag{7.13}$$

but $$\frac{dg_a}{dt} = G_a \text{ and } \frac{dg_b}{dt} = G_b$$

Hence $$\frac{ds}{dt} + \frac{S}{M} G_a + \frac{S}{M} G_b = h_a G_a + h_b G_b + H(t)\left[G_a + G_b\right]$$

Multiply both sides by exp $(G_a + G_b)\, t/M$ and one can say that

$$\frac{d}{dt}\left[S \exp\left[(G_a + G_b)\frac{t}{M}\right]\right] = \exp\left[(G_a + G_b)\frac{t}{M}\right]\left[h_a G_a + h_b G_b + H(t)\left(G_a + G_b\right)\right]$$

$$A + S \exp\left[(G_a + G_b)\frac{t}{M}\right] = \int \exp\left[(G_a + G_b)\frac{t}{M}\right]\left[h_a G_a + h_b G_b + H(t)(G_a + G_b)\right]$$

where A is a constant of integration.

H(t) is here as a constant. Upon this assumption then the solution is simplified to

$$A + S\exp\left[(G_a + G_b)\frac{t}{M}\right] = \frac{M}{(G_a+G_b)}\exp\left[(G_a+G_b)\frac{t}{M}\right]\left[h_aG_a + h_bG_b + H(t)\ (G_a+G_b)\right]$$

To assess the value of the constant A, take as boundary conditions S = So when t = 0

Then $A = \frac{M}{(G_a+G_b)}\left[h_aG_a + h_bG_b + H(t)(G_a+G_b)\right] - S_o$

Hence

$$S = \frac{M}{(G_a+G_b)}\left[h_aG_a + h_bG_b + H(t)/(G_a+G_b)\right]\left[1 - \exp\left(-(G_a+G_b)\frac{t}{M}\right)\right] +$$

$$+ S_o\exp\left(-(G_a+G_b)\frac{t}{M}\right) \qquad (7.14)$$

Using equation (7.14) it is therefore possible to calculate the dry bulb temperature and moisture content (relative humidity) within a room at any instant. This procedure involves using the sensible heat gains to calculate the dry bulb and the latent heat gains to calculate the moisture content of the air.

7.2 Experimental Procedure.

7.2.1 Test Area. Sliding door tests were carried out in the treatment area which was located in the centre core of the building. A detailed plan of the treatment area is given in Figure 7.6. The treatment room opened onto an air-lock or vestibule to which there was access from both corridors and a dirty utility room. Swing door tests were carried out in isolation room 1 as described in Section 4.2.

The air in the treatment room was supplied by a ceiling diffuser and in the isolation room by high level grilles, the extracts being by low level grilles. Grilles and diffusers were sealed when not in use as was a by-pass damper from the treatment room into the vestibule. It should be noted that the treatment area did not have any outside walls.

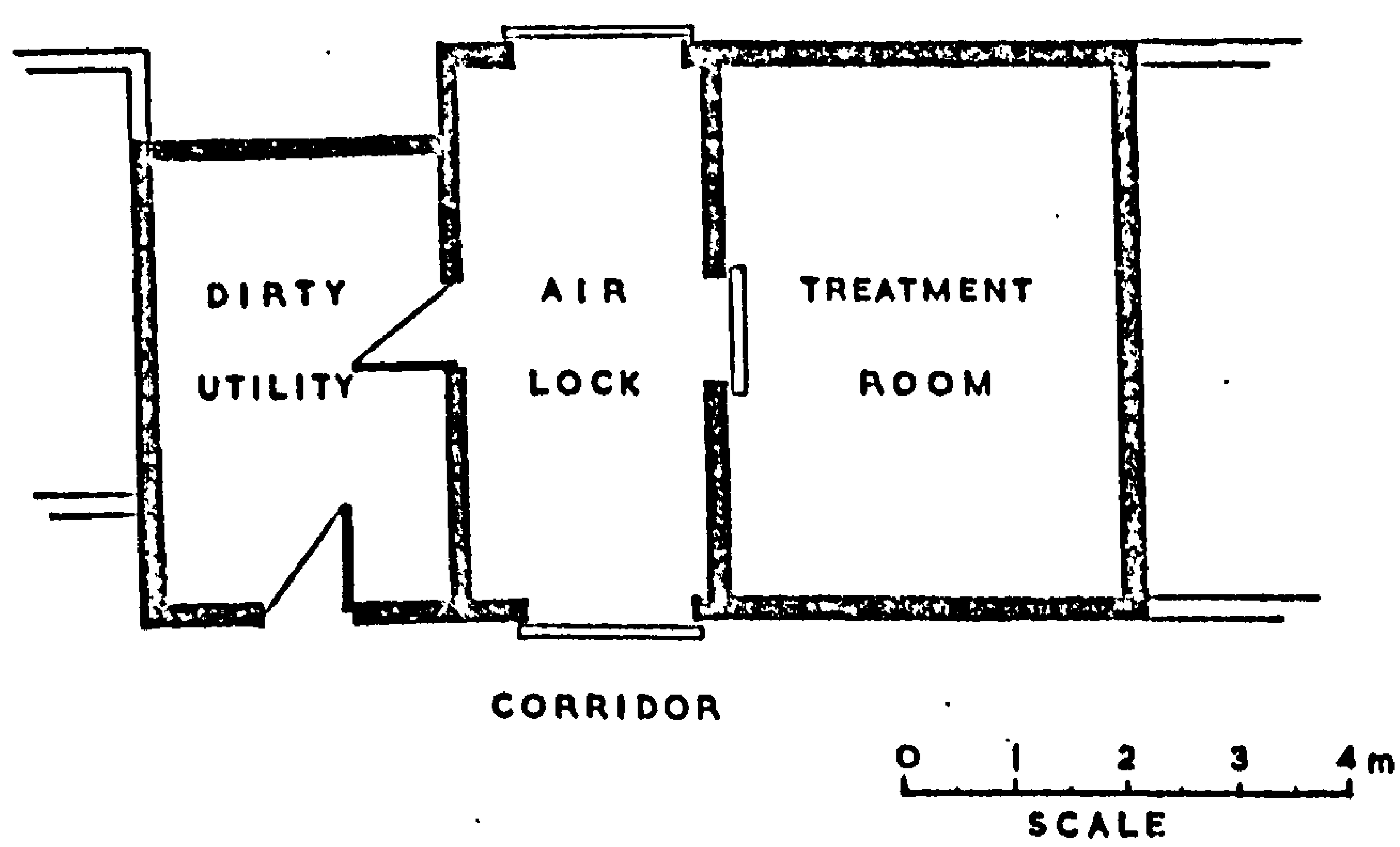


FIG. 7. 6. TREATMENT ROOM AREA.

7.2.2 Instrumentation.

Air supply volume and temperatures, were measured in a similar manner to that of Section 4.2. However, instead of calculating the air transfer volume through the doors as measured by hot wire anemometers, tracer gas was used and the volume calculated by decay equation theory.

7.2.3. Scope of Tests and Procedure.

A limited series of balanced and positive tests were carried out to determine the effect of door opening habits on the air transfer volume. Tests were also carried out to determine the amount of air transferred through an open door when a person walked through it.

Balanced tests, with no supply or extract, were carried out, in the treatment room. The reason for no supply or extract was for experimental accuracy, but in order to simulate the turbulence of air as expected in ventilated rooms, a small fan recirculated about $0.15m^3/s$ of air in the room. The experiments to determine the air transferred solely by a person walking through an open door were conducted under the same experimental conditions. The positive tests, which were carried out in a slightly different way, had a supply of $0.15m^3/s$ and no extract to the treatment room.

All tests were carried out over a similar range of temperature differences (0 to 8^oC). Both balanced and positive systems were tested using the sliding door of the treatment room but only balanced tests were carried out using the swing door of the isolation room.

During the door opening tests each door was opened for one standard opening (as near as possible to 6 seconds) every 30 seconds, and this was repeated 40 to 60 times to obtain an accurate value as to the amount of air transferred per single opening of the door. Only one door breadth was used, i.e. 0.90m, and a person walked through the opening each time to simulate 'traffic'.

If additional heating was required a radiator in the treatment room (or isolation room) was switched on from one to two hours before the beginning of a test to enable the radiator to come up to maximum temperature and the air temperatures in the rooms to stabilise. Temperature grids were taken in the room and vestibule out of the influence of the door.

a) Balanced tests: Once the room temperature had stabilised each test began by releasing a limited amount of nitrous oxide into the room once the room temperature had stabilised. The release of nitrous oxide was terminated when the gas analyser read between 80 and 100 ppm. Grid temperatures in the room and vestibule were then recorded and left cycling throughout the test. Because of large gaps around the treatment room door when closed there was considerable exchange of air from the vestibule. Leakage decay tests were therefore carried out to determine the exchange volume through these gaps when the door was closed. These tests are reported in Section 5.

Once the gas analyser had been read, the door was opened and closed for the stipulated period and number of times, a person passing through the opening each time. When the reading of the gas analyser had diminished a sufficient amount, the test was terminated (approximately 30 minutes).

The total transfer volume was the amount of air which both passed out of the door when opened and closed, and the amount that leaked out of the room through the crack round the door when closed. This volume was calculated over the full time of the test by the equation:

$$\eta = \eta_0 \exp\left(-\frac{Qt}{Q_r}\right)$$

hence solving for Q. The transfer volumes were calculated for every 5 minutes of decay and an average value obtained.

The 'test' or total transfer volume, as stated above, was calculated for a 30 minute period and comprised of the transfer volume due to leakage and the door opening. The leakage transfer volume was also calculated for a 30 minute period (i.e. 24 secs. for every 30 sec. due to a door opening period of 6 seconds). From the above results the door opening transfer volume could be determined in terms of volume per opening (m^3/standard door opening of 6 seconds), i.e. (Door + Leakage volume) - (leakage volume) = Door transfer volume.

but $\frac{\text{Door Transfer volume}}{\text{Number of Standard Openings}} = $ Transfer volume per S.O.

b) <u>Positive tests</u>: The positive tests were carried out in a similar manner, the only difference being that the nitrous oxide was released at a constant rate in the vestibule and the growth of the gas in the room was measured. Since very little gas entered the room, the sample in the vestibule was diluted with nitrogen (1:8) while the room sample was neat. These tests were also carried out for approximately 30 minutes. No leakage tests had to be carried out because there was a large enough pressure within the room to prevent ingress of air while the door was closed.

The transfer volume due to the door opening was caluclated from the equation:

$$\eta = \left[1 - \frac{Q_{so}}{Q_{so} + Q_s}\right] \times 100$$

where η was the equilibrium efficiency

Q_{so} = volume of air transferred through the doorway per standard opening (m^3/s).

and Q_s = supply volume (m^3/s)

solving for Q_{so}

The efficiency used in this equation was the equilibrium efficiency, i.e. the efficiency within the room decayed from 100% at the start of the test to the equilibrium state in approximately 15 mins. The transfer volume per standard door opening (Q) was then calculated.

c) <u>Air Transfer solely by a person</u>: The procedure followed for these tests was similar to that of the balanced ones in that there was no extract or input of air, and a small fan circulated the air in the room. The door (0.90m x 2.05m) was left open but no temperature readings were taken.

In order to evaluate the transfer of air through the door by convection currents and turbulence, decay tests were run. Walking through the doors then started, and speed being either slow (approximately that of a person passing through a door which has to be opened) or fast (a brisk walk). This decay rate was also monitored.

These two decay experiments were alternated for each speed and the two alternate air flows through the door over the test period. This is as described in the balanced tests (7.2.3a). The difference in these two values divided by the number of times a person passed through the door gave the air transferred by a person walking through an open doorway.

7.2.4 Isolation Efficiency of the rooms under use:

The following formulae may be used to determine the overall, or average, room isolation efficiency for different conditions, i.e. sliding or swing doors and balanced or positive ventilation.

$$\eta \text{ overall} = \left[1 - \frac{(Q_{so} \times N) + (Q_L \times t_c)}{(Q_{so} \times N) + (Q_L \times t_c) + Q_s}\right] \times 100 \qquad (7.15)$$

where Q_{so} = air transfer volume per standard door opening at temperature differential ΔT ^oC.

N = number of standard door openings per hour.

Q_L = transfer volume due to leakage in m^3/hr.

t_c = percentage time during which door was closed in an hour.

Q_s = ventilation supply rate m^3/hr.

This formula does not take into account the build-up of contamination in the room by successive openings if the time between openings is not sufficient to allow the ventilation system to remove the contamination completely. However, this overall value expresses the isolation efficiency in a convenient form suitable to engineering practice.

If the efficiency was based on the build-up of contamination with successive openings, the decay and growth equations (7.6 and 7.7) would have to be applied to each opening and clearly if, say, 10 openings per hour were recorded, 10 operations involving both the decay and growth equations would be extremely laborious which in itself would render it unsuitable for general use unless by computer. With a large number of openings, the former method would in fact give as accurate enough indication of the isolation effeciency, due to the decay of efficiency level towards the equilibrium state.

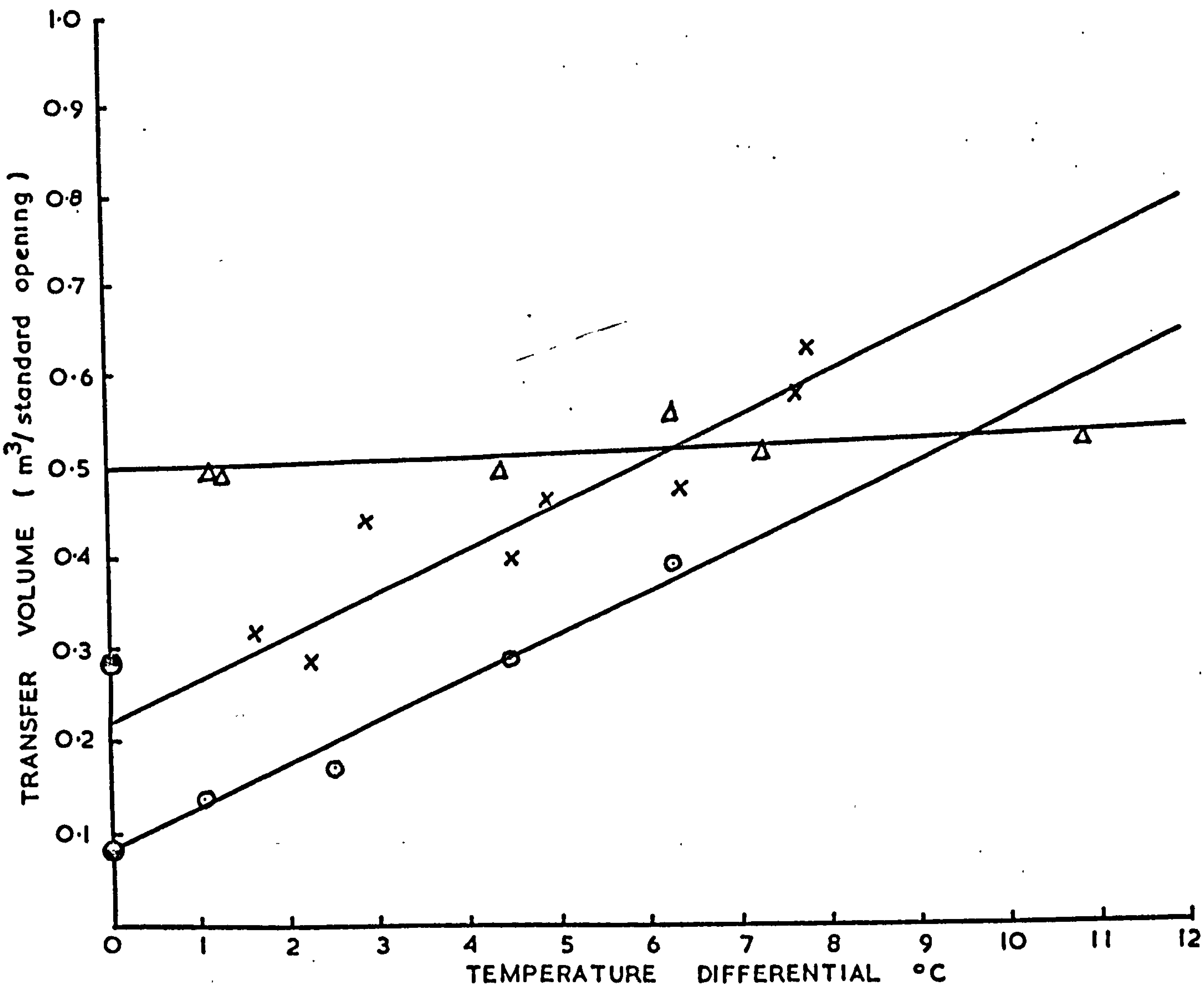
7.3 Results

7.3.1 Air transfer volume through doors being opened and closed:

Given in Figure 7.7 are the test results for a sliding door with balanced or positive ventilation. The swing door was nominally 0.90m x 2.05m in dimension and the sliding door was the same height but was adjusted so that it did not open more than 0.90m. The volume transfer is expressed as the quantity of air flowing through a doorway when a person opens a door, steps through it and shuts the door, taking 6 seconds to do so. These values do not include the normal leakage which occurs continually through cracks in the door owing to temperature difference or air movement.

It would appear from these graphs that with a standard size swing door, the amount of air transferred per standard opening is about $0.5m^3$ and this amount is not influenced by temperature. With the sliding door about half of this amount of air is moved with no temperature difference ($0.22m^3$) but the air movement is influenced markedly by temperature difference. With a temperature difference of 6^oC the amount transferred is the same as a swing door.

Shown in Figure 7.8 is the leakage expected from both a swing and sliding door 0.90m x 2.05m in size due to the movement of air through the cracks round the door when it is shut. This graph is taken from Section 5, but the results were obtained at the same time as the results reported in this section. From this graph, 7.8, the following should be noted that (i) the leakage of the sliding door was twice that of the



KEY.

- ⊙ SLIDING DOOR , POSITIVE (0·15 m /s).
- × SLIDING DOOR , BALANCED.
- Δ SWING DOOR , BALANCED.
- AMOUNT TRANSFERRED BY FAST WALK.
- AMOUNT TRANSFERRED BY SLOW WALK.

FIG. 7.7. QUANTITY OF AIR TRANSFERRED ACROSS A DOOR PER SINGLE STANDARD OPENING.

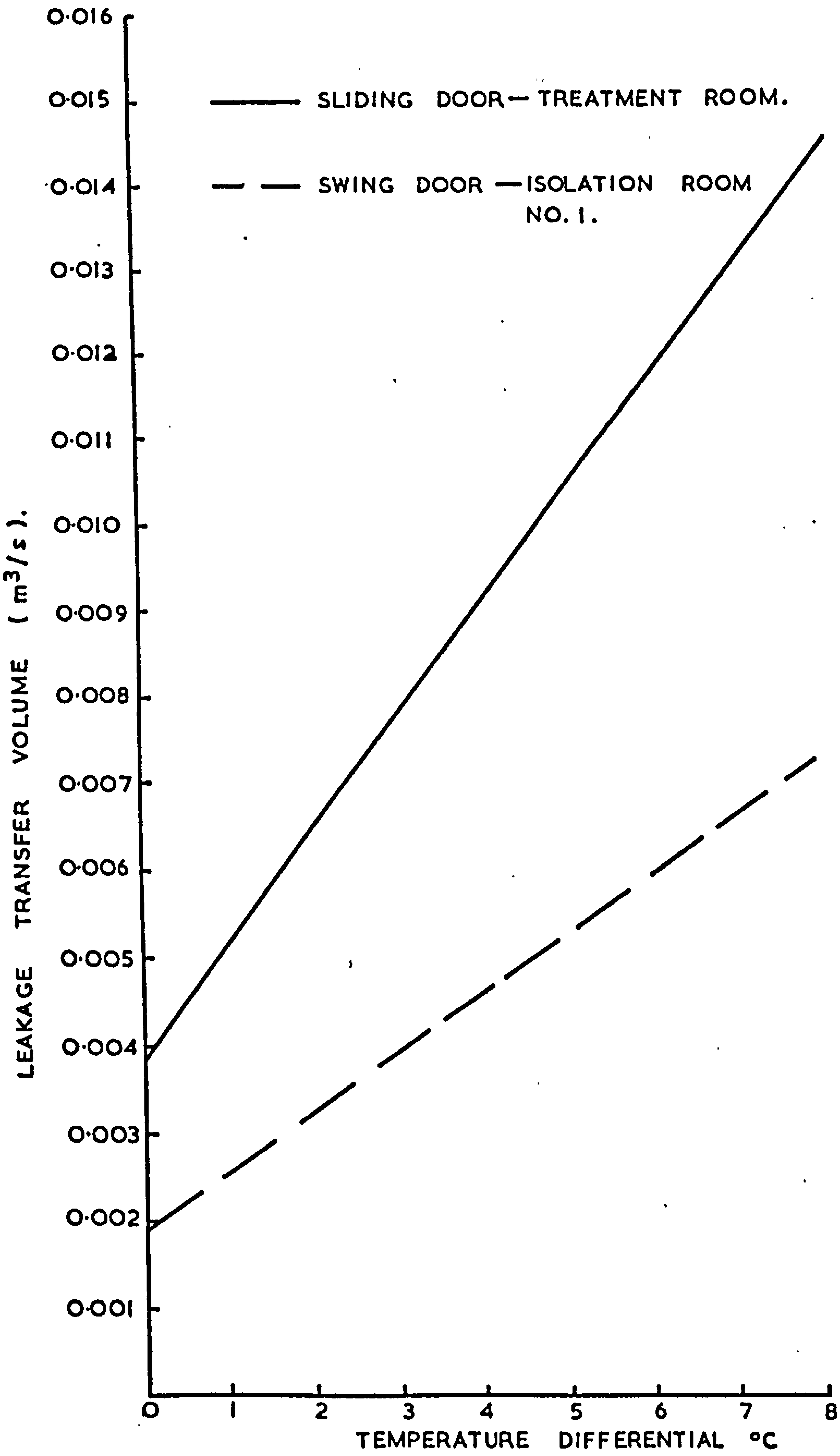


FIG. 7.8. NATURAL LEAKAGE ACROSS A CLOSED DOOR IN A BALANCED OR UNVENTILATED SITUATION.

swing door, the crack area in the slide door being twice that of the swing door (ii) that a quadrupling of the ΔT from $1.5^{0}C$ to $6^{0}C$ would entail a doubling of the amount of air exchange.

A comparison of the two graphs (7.7 and 7.8) shows that under the following standard conditions:-

i) temperature difference of $2^{0}C$ over a period of an hour.

ii) in an isolation room which has a usage of 6 door openings per hour with

iii) a balanced system,

the average leakage due to openings would be $0.000528m^3/s$ for a slide door and $0.000837m^3/s$ for a swing door. This is in comparison to a crack leakage rate over that period of $0.0064m^3/s$ (slide door) and $0.0032m^3/s$ (swing door).

What is also important to note is the comparison of the quantity of air being transferred due to the door being opened, and that of a door fully opened under 'equilibrium' conditions. It would appear that 0.3 of the quantity of air is transferred by opening a sliding door than if the door was fully open for a temperature differential of $0^{0}C$ to $5^{0}C$. Swing doors would give a figure of between 0.65 to 0.35 for a temperature differential of $0^{0}C$ to $5^{0}C$ respectively. It is perhaps surprising that less air is transferred during the time a door is opened than would have been transferred in the time had it been open,but it must be remembered that the air required time to accelerate and establish equilibrium conditions. It is also true that in the case of the swing door the pumping action is two way in that the air will be pushed into the room when the door is opened and a certain amount removed when the door is shut.

7.3.2 Quantities of air transferred by a person walking through an open door (0.90m x 2.05m):

From the gas decay method it was established that the amount of air transferred through an open door 0.90m wide and 2.05m in height was $0.286m^3$ when walking fast and $0.0875m^3$ when walking slow (a threefold difference).

The relationship of these results to those obtained when a person walks through a door but also opens and closes it is shown in Figure 7.7. It is possible that these results are slightly low as there could have been entrainment of air when the person after passing through the door, turned and passed back through the door.

7.3.3 <u>Isolation efficiencies</u>: The regression equations relating the air transfer for a sliding door with balanced or positive ventilation are given below.

<u>Sliding Door</u>

Transfer volume due to crack leakage, Q_L

$$Q_L = 0.003872 + 0.001285\ \Delta T$$

Air transfer volume per standard door opening

$$Q_{s.o.} = 0.221633 + 0.046691\ \Delta T - 0.925693 Q_s$$

<u>Swing Door</u>

$$Q_L = 0.001953 + 0.000641\ \Delta T$$

$$Q_{s.o.} = 0.449030 + 0.003087\ \Delta T$$

By substituting the values of $Q_{s.o.}$, Q_L and Q_s into equation 7.15 the overall isolation efficiency of rooms against unwanted airflow may be calculated. Based on this procedure graphs were constructed to show the efficiency of typical situations which represent those to be found in isolation and treatment rooms. These graphs are given in Figures 7.9 to 7.12 and are based on a door 0.90m wide x 2.05m high.

Figures 7.9 and 7.10 compare the isolation efficiency of balanced and positive systems of ventilation with an air supply of $0.50 m^3/s$ and $0.15 m^3/s$, over a temperature differential of 0 to 5^oC. Figure 7.9 simulates the conditions that could be met in an isolation room with 6 openings per hour. Figure 7.10 considers conditions as found in treatment rooms of 30 openings per hour.

Figures 7.11 and 7.12 compare the isolation efficiency of rooms with either a swing or sliding door and served with an air supply and extract of $0.15 m^3/s$ or $0.50 m^3/s$ i.e. a balanced system. Figure 7.11

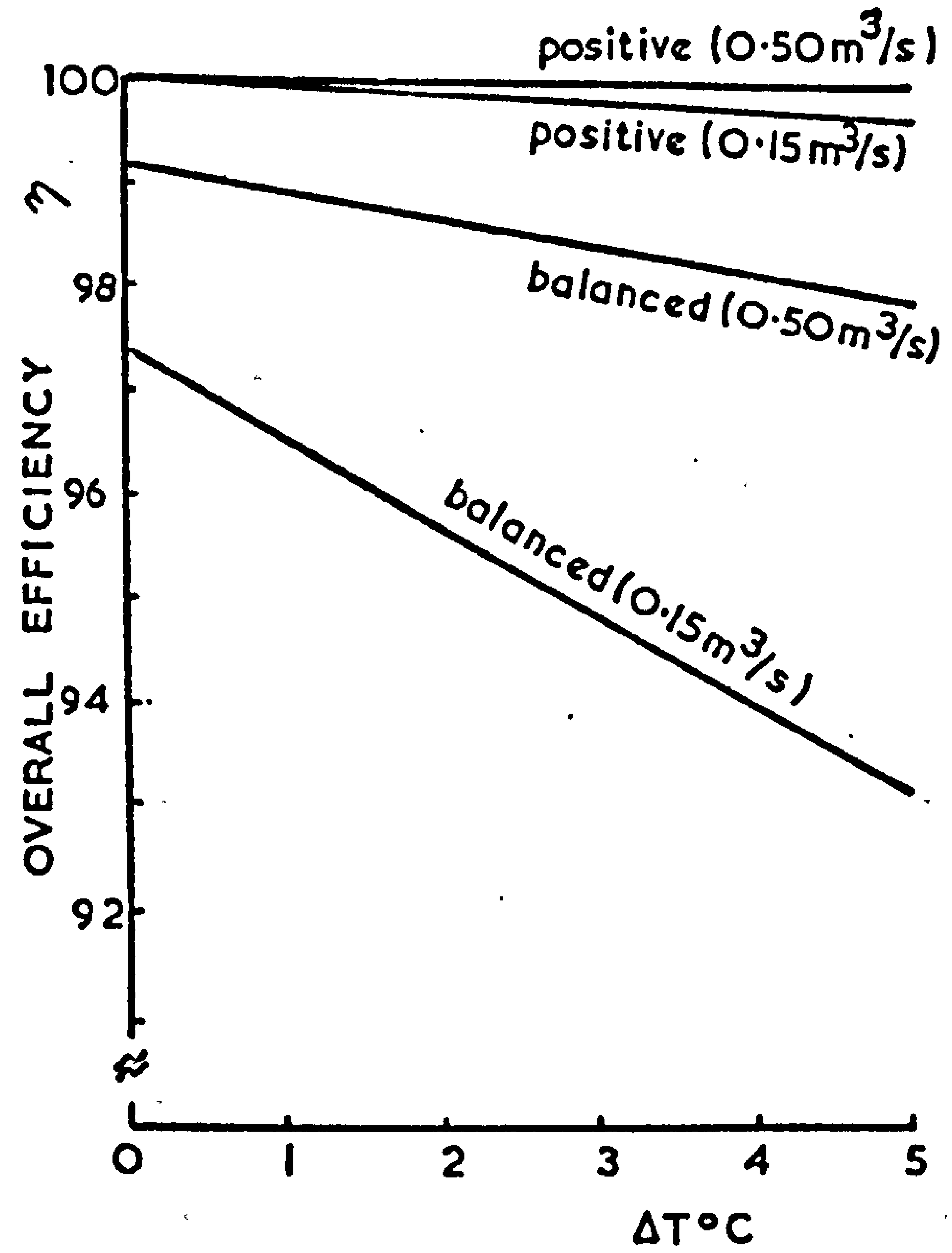


FIG. 7.9. COMPARISON OF BALANCED AND POSITIVE SYSTEMS WITH SLIDING DOORS. (6 OPENINGS/HR.)

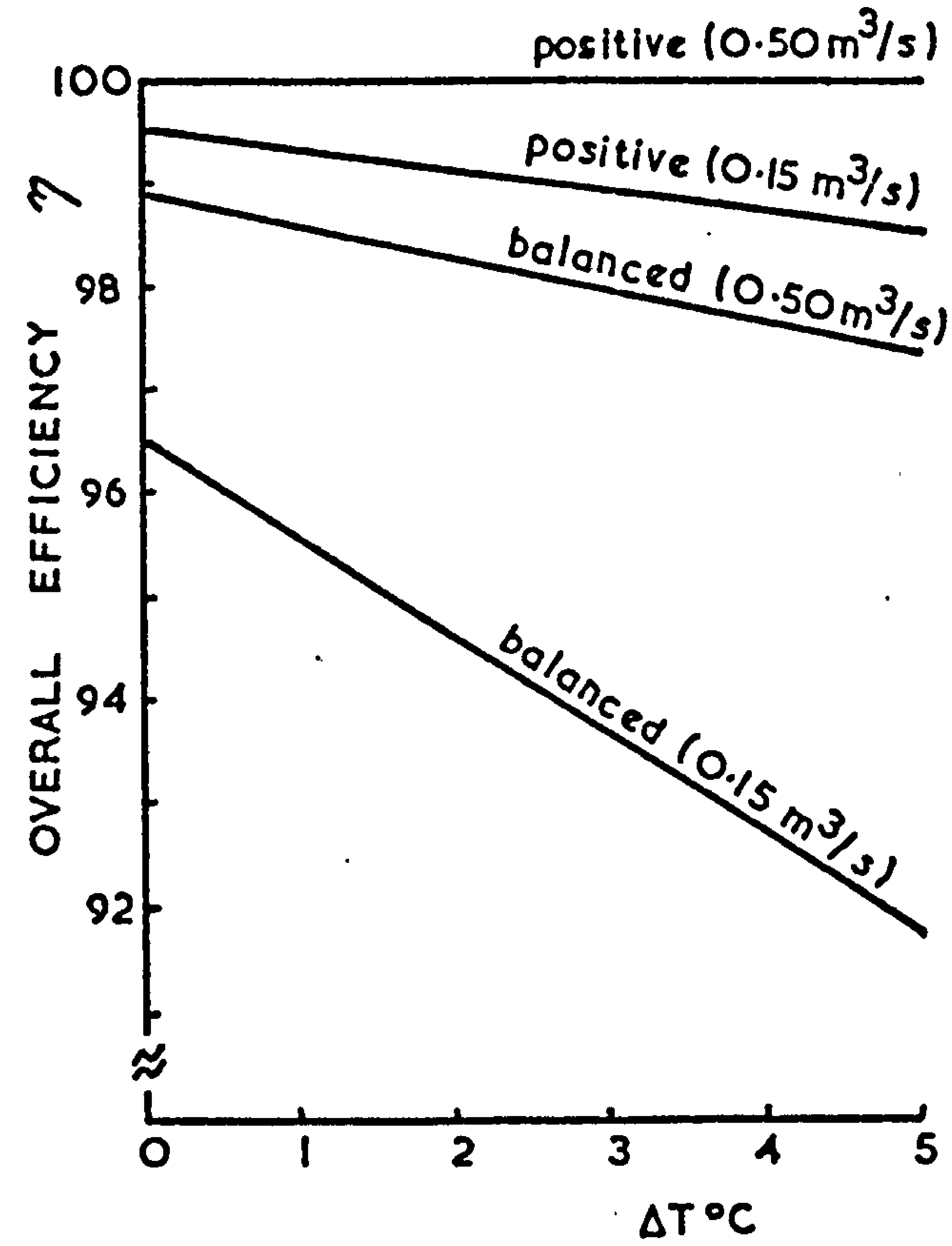


FIG. 7.10. COMPARISON OF BALANCED AND POSITIVE SYSTEMS WITH SLIDING DOORS. (30 OPENINGS/HR.)

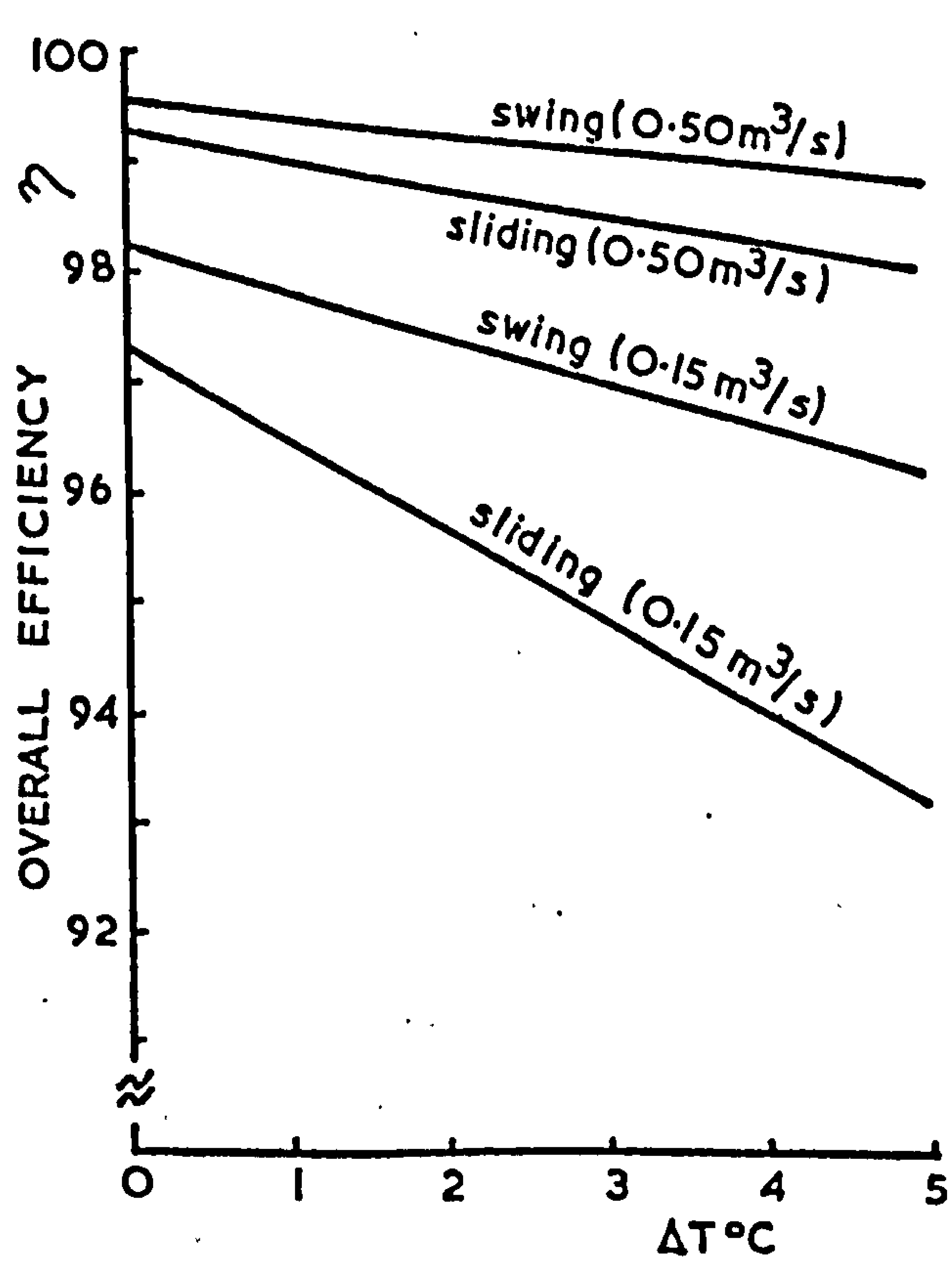


FIG. 7.11. COMPARISON OF SWING AND SLIDING DOORS, BALANCED SYSTEM. (6 OPENINGS / HR.)

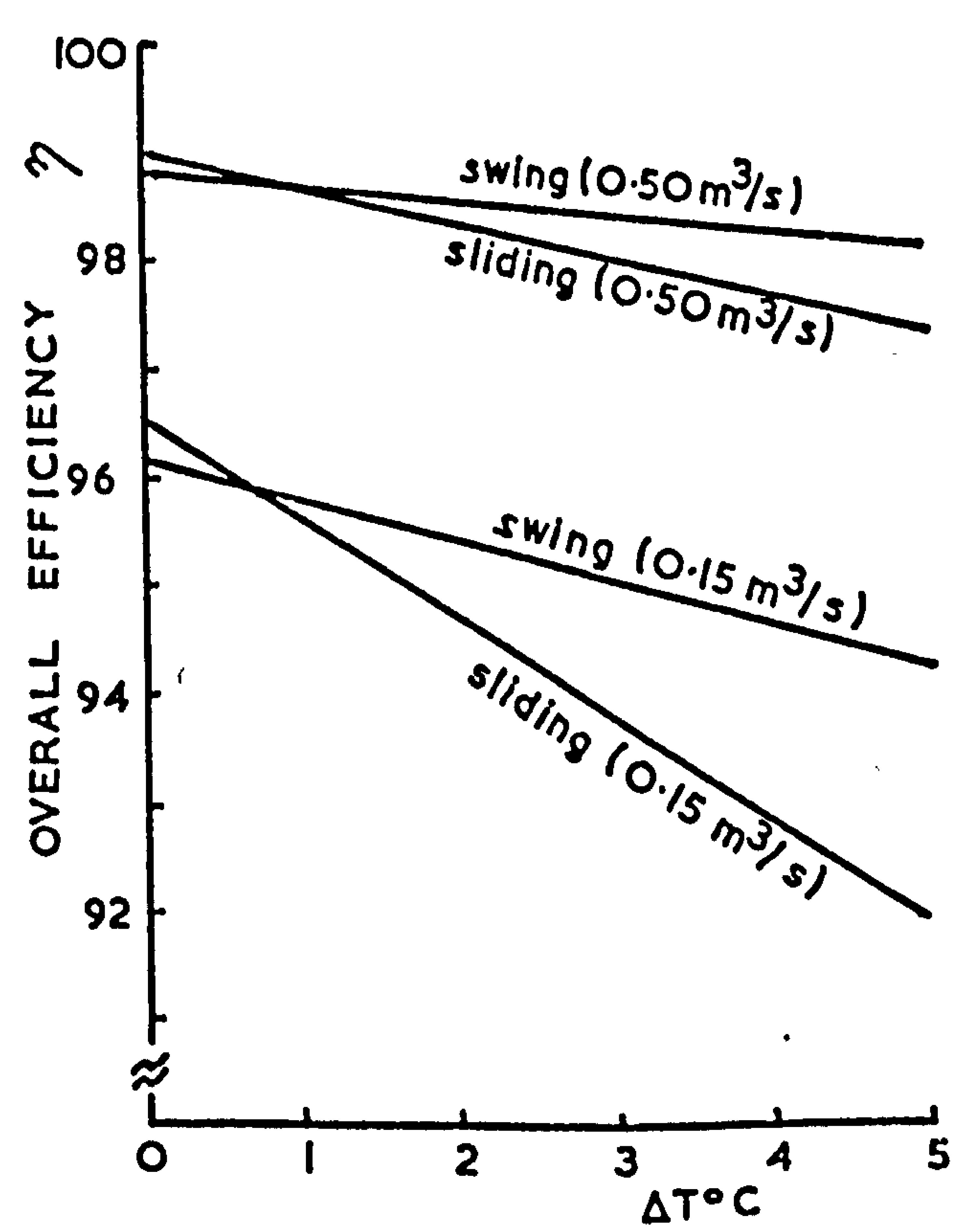


FIG. 7.12. COMPARISON OF SWING AND SLIDING DOORS, BALANCED SYSTEM. (30 OPENINGS / HR.)

simulates isolation room conditions, i.e. 6 openings per hour and 7.12 treatment room conditions, i.e. 30 openings per hour.

7.4 Discussion and Conslusions

Door usage and infiltration through cracks make a significant contribution in the undersirable transfer of air into critical areas. However this is comparatively small in comparison to a door being left open and therefore reinforces the contention that self closing mechanisms must be fitted on all doors connecting critical areas.

When there was no temperature difference between the areas connected by a door, a swing door will transfer twice the volume of air as a slide door ($0.5m^3$ per standard opening compared to $0.22m^3$). Temperature difference between adjacent zones has little or no effect on the amount of air transferred by a swing door. This is probably caused by the door pumping action overriding the pressures causing convection flow. With a slide door a 5^oC to 6^oC increase in temperature across a door will double the quantity of air being transferred per opening at 0^oC and cause the sliding door to transfer as much air as the swing door.

Pressurisation of an area will reduce the transfer of air from one area to another. Tests were carried out with a sliding door only and it is not expected that pressurisation will have as great an effect on swing doors owing to the pumping action. The graph indicates that for a 1^oC temperature differences over a sliding door $0.30m^3/s$ would be necessary to reduce transfer to a negligible amount.

When the door usage is around 6 per hour (the average as found in the study of isolation rooms) the natural leakage through cracks in doors has greater effect than door usage. With a ΔT of 2^oC the leakage through cracks round the door was $0.0032m^3/s$ for swing doors and $0.0064m^3/s$ for slide doors as compared to $0.000837m^3/s$ and $0.000528m^3/s$ respectively. That is to say there was 12 times more leakage round a slide door than was transferred by a person opening a door, walking through, then shutting it.

Less air is transferred during the time a door is opened than would have been transferred in the time had it been open. About 0.3 of the quantity of air is transferred by opening a sliding door than if the door was fully opened for a ΔT of 0^oC to 5.0^oC. Swing doors would give a figure of between 0.6 to 0.3 for a ΔT of 0.75^oC to 5.0^oC respectively.

When a person walks fast through a door 0.90m x 2.05m, he transferred $0.286m^3$ of air. A person walking slowly through that door will transfer $0.0875m^3$ of air; about 3 times less.

8. CONCLUSIONS AND DESIGN RECOMMENDATIONS

8.1 Doorways when constantly open.

With the results obtained, it is now possible to predict the volumetric exchange of air through an opening either under balanced or positive conditons and also to determine the supply volume of air required to completely isolate an area, preventing the movement of contaminated air to critical areas.

These supply volumes are presented in Figure 8.1 which gives the volumes required for both 0.90m and 1.40m wide doors over a 10^{o}C temperature differential range. For a 0.90m wide door with 1^{o}C temperature differential, a normal value in practice, as shall be seen in Section 6, it is necessary to supply an excess of 0.50m^3/s (1060 cfm). With a 1.40m wide door under the same conditions the volume required is in order of 0.75m^3/s (1590 cfm) if rooms are to be completely isolated from other areas. Even with zero temperature differential, volumes of 0.20m^3/s (424 cfm) and 0.30m^3/s (636 cfm) are requied for 0.90m and 1.40m wide doors respectively.

With such large exchange volumes it is obvious that it is necessary to keep temperature differentials to a minimum if rooms are to be completely isolated from other areas. One exception to this rule is obviously the cold storage room.

8.2 Doorways when they are shut or slightly open.

8.2.1 Single Door Areas

When a normal door is shut, little air is effectively required to isolate it. For a single door 0.05m^3/s of air would be sufficient to carry out this purpose.

It would appear that in the case of single door rooms a pressure differential of 0.02mm w.g. would be quite sufficient to pressurise the room against movement due to thermal differences of up to 5^{o}C.

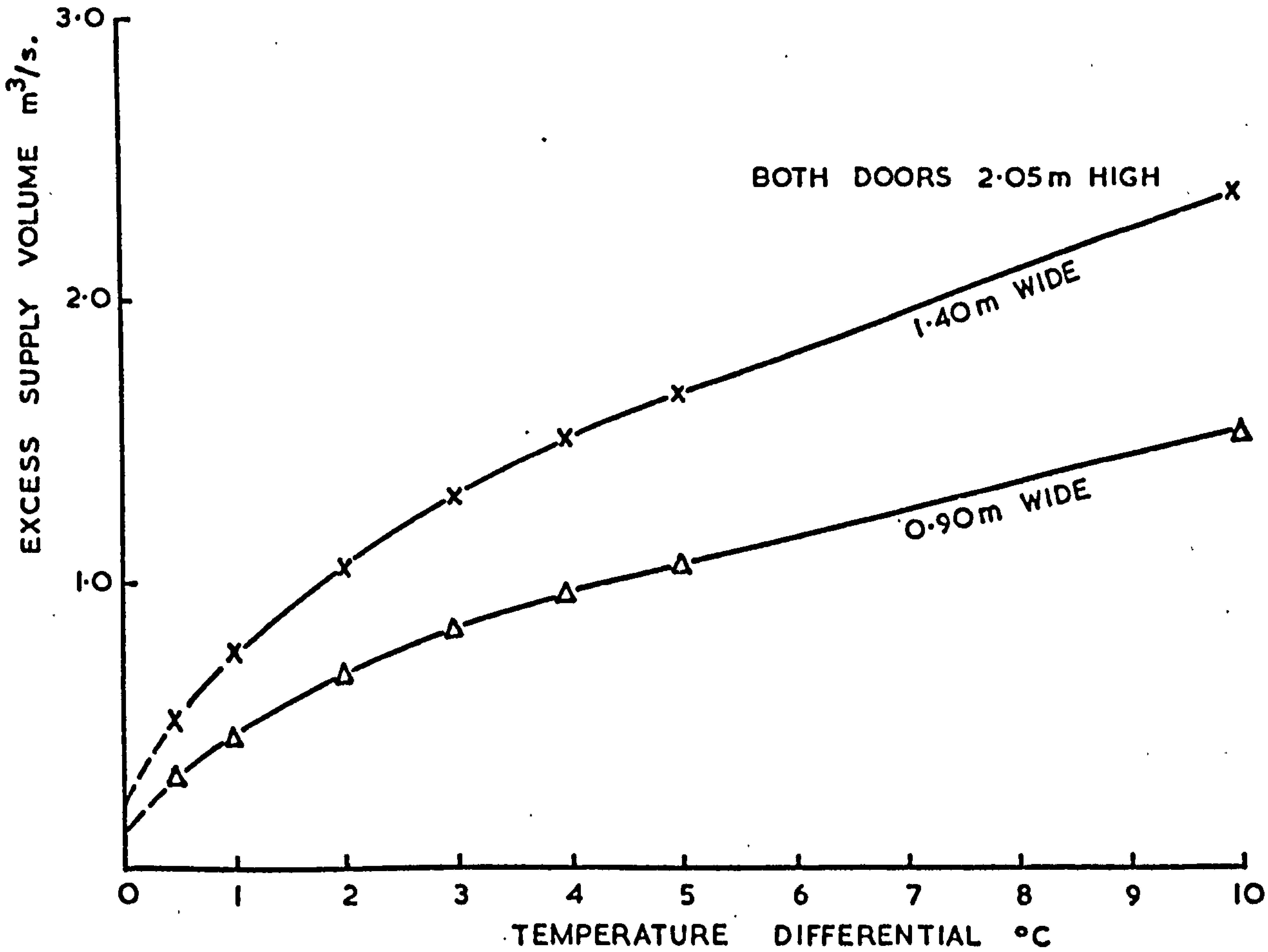


FIG.8.1. SUPPLY VOLUME REQUIRED TO COMPLETELY ISOLATE AN AREA (m^3/s).

When a single swing door of normal dimensions (0.90 x 2.05m) was shut the following approximate pressures were found with these amounts of air passing through.

m^3/s	0.30	0.25	0.20	0.15	0.10	0.05
mm w.g.	3.25	2.50	2.00	1.00	0.50	0.10

It can be seen from the above table that it would be possible to pass up to $0.25m^3/s$ of air without causing back pressure greater than 2.50mm w.g. A sliding door as described above would pass this quantity.

The figures quoted above will give an unnecessary back pressure on the fan and hence, a reduced flow, or will require a higher design static pressure from the fans. It is therefore suggested that with single doors passing amounts of air greater than $0.05m^3/s$ the door undercut should be calculated by subtracting the crack area round the two sides and top from the area given in Table 5.2 for the designed airflow. This will give the undercut area and do away with the necessity of providing a bypass damper. A 10cm undercut would cover most cases.

When a door situation does not conform to the above suggestions, e.g. a greater amount of flow of air is required or the dimensions differ equation 5.2 should be used i.e.

$$Q = C \times A \times 4.43 \times \left(\frac{P_v}{\rho}\right)^{1/2}$$

Crack widths round a normal door may be taken as 0.4cm and undercuts or door louvres thereby calculated.

8.2.2 Multiple Door Areas.

In the case of areas with more than one door, large pressure differences over doorways should be used in order that when one door is opened as much air as possible should pass out of it. In these cases a pressure differential no greater than 2.50mm w.g. (0.10 in. w.g.) should be used. A normal set of doors in an operating room should not require any undercutting if a high pressure drop is to be achieved. Flap dampers may therefore be dispensed with inside an operating room, hence removing a source of mechanical failure and an object of annoyance (due to banging).

Although the operating suite studied with regard to pressure drop across multiple doors was not typical of a modern designed suite it gives an interesting insight into the design of an air movement control scheme.

It was most obvious from the results obtained that it is naive to consider that when a door is opened, all the air supplied to the room will surge out through that door. The results have shown that this will not be so unless provision is made for the extra air to pass out. The experiments have in fact shown that in some cases no significant amount of extra air goes through the door when a door is opened into another room whose only other opening is another shut door; it simply moves the resistance to airflow on to the other door and little additional flow takes place.

For the above reasons more thought must be given to where the air will go when a door is opened. If Figure 5.3 is considered as a well thought out system, it may be seen that if the door to the anaesthetic room was opened, little or no extra air would pass through the door to protect the operating room from contamination from the anaesthetic room. In order to help this situation a flap damper should be fitted in a similar position to that in the scrub-up room.

It is therefore advocated that no flap dampers are necessary in operating rooms but may be useful if aiding the removal of air from ancillary rooms when the operating room door is opened into that area.

It would be necessary to ensure that both the clean and dirty corridors would be capable of extracting the same amount of air as supplied to the room in order that full protection could be given. As this would be difficult to achieve, what is the possibility of air movement schemes, as planned in modern operating suites, actually working to anything like their design?

8.3 Doorways during usage.

8.3.1 Isolation Rooms

Ventilation: It has been shown in Section 6 that the average number of door openings per hour in an isolation room is between 0 and 6, with a maximum of 10. Due to this, the infiltration rate through door cracks in rooms positively ventilated is more important than door usage. As well as this, due to economics, isolation rooms would not have much more than $0.15 m^3/s$ of air supplied to them. This however should be sufficient to give good isolation in a well thought out scheme. Such a scheme would necessitate the use of either a positive or negative system of air movement, the use of a balanced system of ventilation being appreciably inferior at this low air supply rate. This is shown in Figure 7.9 where with a 0 - $2^{\circ}C$ temperature differential the overall isolation efficiency of a scheme using a supply of $0.15 m^3/s$ would be between 97.27% and 95.64% for a balanced scheme and 99.91% and 99.80% for a positive scheme.

These figures imply that the door is always kept closed but on occasion of the door being left open the positive scheme would afford more protection than the balanced system. (See Section 4).

Doors: If a balanced system of air movement in isolation rooms was chosen, then a swing door (from the point of view of the small contribution of leakage round the crack areas of the door) would have been best. This is shown in Figure 7.11. However, in a positive scheme it will also prove to be the best as these doors can be easily fitted with self-closers which are an absolute necessity for good air control. The normally larger crack area of sliding doors would not affect the isolation efficiency in a positive or negative system as the air should pass out through the cracks and undercuts of the doors and hence isolate it (See Section 5).

8.3.2 Treatment Rooms

Ventilation: Section 6 of this report will show that in order to ensure a low concentration of bacteria in the air during wound dressings and to ensure that the carry-over of bacteria from one patient to another is reduced to a minimum, a high volume must be used.

Figure 7.10 compares the isolation efficiency of a balanced and positive system of ventilation under air supply rates of $0.15m^3/s$ and $0.50m^3/s$. Although it may be seen that a balanced system of ventilation with an air supply of $0.15m^3/s$ would give an efficiency of 96.45% with no ΔT and only 94.55% at a ΔT of 2^oC. However when the supply rate is increased to $0.50m^3/s$ the isolation efficiency increases to 98.91% and 98.29% respectively.

Although this isolation efficiency is not as high as for positive systems it would enable nurses to change and inspect wounds without recourse to selecting either a positive or negative system of ventilation. It has been found in general that the simpler the system the more effectively it will work, from the point of view that people do not appreciate or have no interest in complex systems and do not attempt to change them.

A reasonable criticism of such a system is that it is often difficult to balance an air-conditioning system to give the same supply and exhaust volume. However, criticism of this must be balanced with the fact that we have found positive/negative systems to be radically malfunctioning under use, due to the breakdown of control dampers, etc.

The use of a small ventilated vestibule or environmental lock could function as a buffer to differences of supply and extract volume, temperature difference, and could be used as a gowning area. Consideration should be given to an in-room air recirculating system, which could be a fan with filters and silencers sited in the room. This would be switched on when the room was in use. The usefulness of an in-room recirculating system could be quite marked in a large building where it may be easier to run a large number of treatment rooms off the same plant, the large plant supplying enough air for conditioning the room and the recirculating in-room system being switched on when the room was in use.

This would also mean that any mistakes in the balancing would be smaller as a percentage of the total air supply.

It is also possible that the high air supply rates required(0.40 to $0.50 m^3/s$) would give rise to engineering and comfort problems in what is normally a fairly small room.

Doors: Owing to the fact that the system advocated will be a balanced one it would appear at first sight that a swing door, with its normally better seal, would be better. However due to the large number of openings that may be expected when the Treatment room is being used there is little to choose between the two. It can be seen from Figure 7.12 that the overall isolation efficiency for ΔT of 0 - $2^{0}C$ would be between 98.81% to 98.57% for a swing door and 98.91% to 98.29% for a sliding door for air exchange rates of $0.50 m^3/s$.

If in fact the infiltration rates could be reduced by a better fitting door the sliding door would be much superior. A self closing mechanism would not be necessary, as, from observation, Treatment Room doors are always closed during use, presumably largely for patient privacy.

In conclusion the following recommendations may be put forward.

1) Isolation rooms should be ventilated with either a positive or negative system depending on requirements or, as second choice, a dual purpose positive/negative system. An air flow of around $0.15 m^3/s$ should be used. The excess air should be allowed to find its way through cracks round the door or, if necessary, under-cuts. A swing door with self-closing mechanism would be best.

2) Treatment rooms should have a balanced system of air exchange around 0.40 - 0.50m^3/s. Owing to this large air requirement and the low demand for treatment room facilities it would be best that the plant could be shut off when not in use. Another possibility is the use of air to condition the room and a large in-room recirculatory filtration system which could be switched on when required.

A vestibule or "environmental" lock with a small amount of balanced ventilation would be very useful to prevent unnecessary exchange of air. A slide door would be best and thought should be given to making it a good fit.

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