Air movement through doorways—the influence of temperature and its control by forced airflow

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SYNOPSIS

The object of this research was to study the movement of air through a fully or partially opened doorway with and without the influence of temperature and to ascertain the amount of supply air required to prevent this movement. Door openings of 0·10 to 1·40 m wide and temperature differential of 0 to 12 degrees Celsius were studied. From these results critical areas in hospitals may be designed more effectively to given requirements.

1. INTRODUCTION

Critical areas in hospitals are designed, from the air movement control point of view, to prevent the exchange of contaminated air to areas where infection may be caused. Blowers and Crew (1960) studied operating-room ventilation and suggested that in order to prevent the ingress of contaminated air in the operating room through an open double doorway a designed air flow of 0·17 m^3/sec per m^2 of door area (33 cfm per ft^2) was required. For a normal sized double door 0·5 m^3/s (1000 cfm) of air was required.

Little consideration, however, has been given in hospital areas to the influence of temperature difference across doorways and the airflow required to overcome the resulting natural convection flow, although work by Ma (1965) and Baird and Whyte (1969) has indicated that the influence of a temperature differential would often be sufficient to completely disrupt the designed movement of air.

In the last ten years, as far as we are aware, there have been four major publications of theoretical and experimental work relating to convection through rectangular openings. There are a number of variables such as type of flow, area of opening, height of opening, temperature differential and condition of opening, the combination of which may be considered in any particular analysis. Table 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 or through openings of door size.

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Also there were no results for the effect of excess pressure within a room acting on the natural convection. The forced air used by Brown and Solvason was in fact a horizontal velocity parallel to the opening surface and acted as a type of air curtain. The work carried out by the last three authors mentioned in Table 1 was to assist the design of cold areas, such as cold meat stores, where the same type of problem of unwanted transfer of air across a doorway exists.

In view of the gaps of knowledge that exist, it was our intention to study the air flow through door openings from 0·10 m to 1·40 m wide, with and without the influence of temperature, and to determine the amount of supply air required to prevent this movement at temperature differences up to 12°C. These studies are reported in this paper.

2. THEORY

2.1 Nomenclature:

C coefficient of discharge
C_T coefficient of temperature
C_V coefficient of fictitious velocity
C_p specific heat of fluid
g acceleration due to gravity

Table 1. Comparison of variables as studied in previous research

Source	Convection	Area, m^2	Height, m	Range of ΔT°C
Brown & 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 & Lilienblum (1968)	Natural	4·5	2·5	12–41·5

H opening height
$P_1 P_2$ pressures in rooms 1 and 2
P_0 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
Q_L leakage transfer volume into an area which is under positive pressure
$T_1 T_2$ temperatures in rooms 1 and 2
t thickness of partition
V velocity
W width of opening
μ dynamic viscosity

2.2. The theory of the volumetric exchange of air due to natural convection through a rectangular opening in a vertical partition (after Brown and Solvason, 1962)

Consider a large sealed enclosure consisting of rooms 1 and 2 as shown in Fig. 1. The rooms are separated by a vertical partition with a rectangular opening of height H and width W. The temperatures 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

$$P_1 = P_0 + \rho_1 g Z \qquad \ldots (1)$$

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

$$P_2 = P_0 + \rho_2 g Z \qquad \ldots (2)$$

g being the acceleration due to gravity and ρ_1 and ρ_2 being the densities of air in rooms 1 and 2 respectively. The pressure difference in these two rooms at the same level is

$$P_2 - P_1 = (\rho_2 - \rho_1) g Z \qquad \ldots (3)$$

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

$$\bar{\rho} = \frac{\rho_1 + \rho_2}{2} \qquad \ldots (4)$$

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. frictionless).

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

$$V = \left[2 g h_a\right]^{1/2} = \left[2g \frac{\Delta\rho}{\bar{\rho}} Z\right]^{1/2} \qquad \ldots (5)$$

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

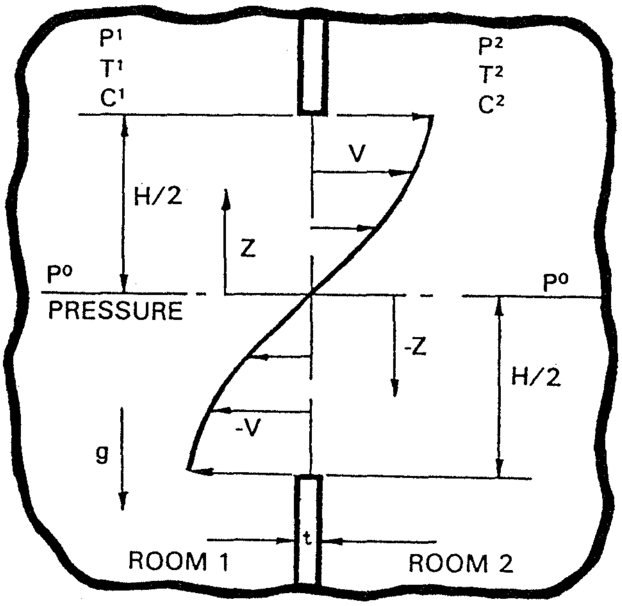

Fig. 1. Schematic representation of natural convection across an opening in a vertical partition.

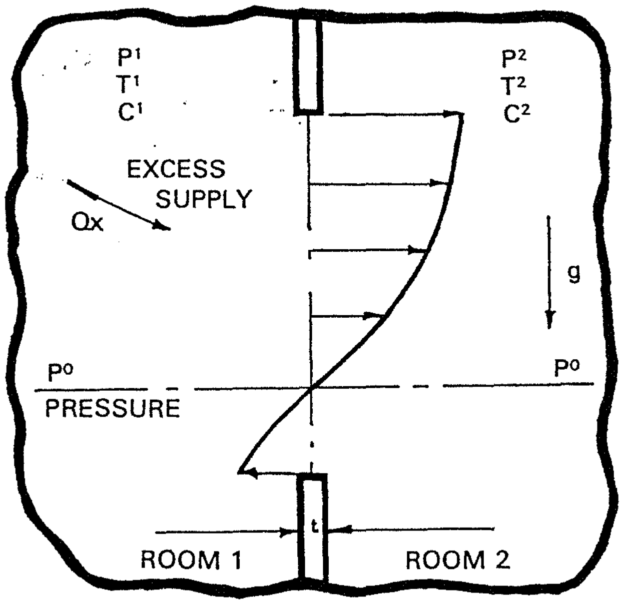

Fig. 2. Schematic representation of combined natural convection and forced-air flow across an opening in a vertical partition.

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

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

$$Q = C\frac{W}{3}\left[g\frac{\Delta\rho}{\bar{\rho}}\right]^{1/2} H^{3/2} \qquad \ldots (6)$$

2.3 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 (Shaw, 1971)

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, Fig. 2. In this case the enclosures are not sealed, air being supplied to one and extracted from the other.

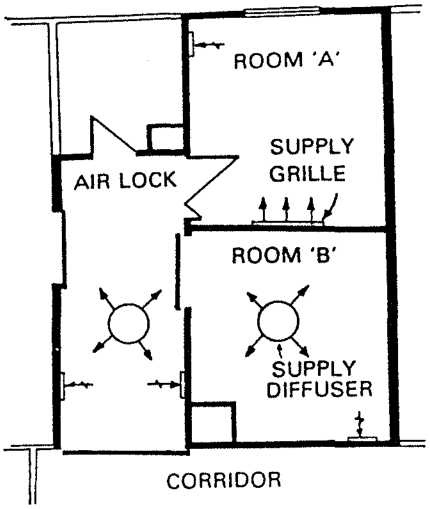

Fig. 3. Plan of test area.

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

$$P_1 = P_0 + \rho_1 gZ + P_X \qquad \ldots (7)$$

where P_X is the additional pressure within the room due to the excess supply ventilation and P_0 the absolute pressure at the level of the neutral zone in the opening. The pressure at the same level in room 2 will be

$$P_2 = P_0 + \rho_2 gZ \qquad \ldots (8)$$

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

$$P_2 - P_1 = (\rho_2 - \rho_1)gZ - P_X \qquad \ldots (9)$$

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

$$h_2 = \frac{P_X}{\rho g} = \frac{V_X^2}{2g}$$

Therefore from equation (9)

$$h_a = h_1 - h_2 \qquad \ldots (10)$$

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.

$$V = \left[2gh_a\right]^{1/2}$$
$$= \left[2g\left(\frac{\Delta\rho}{\bar{\rho}} Z - \frac{V_X^2}{2g}\right)\right]^{1/2}$$
$$= \left[2g\frac{\Delta\rho}{\bar{\rho}} Z - V_X^2\right]^{1/2} \qquad \ldots (11)$$

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

$$Q_L = C \,.\, W\frac{1}{2g(\Delta\rho/\bar{\rho})}\frac{2}{3}\left[2g\frac{\Delta\rho}{\bar{\rho}}\frac{H}{2} - V_X^2\right]^{3/2}$$

$$\therefore Q_L = C \,.\, \frac{W}{3}\frac{1}{g(\Delta\rho/\bar{\rho})}\left[g\frac{\Delta\rho}{\bar{\rho}} H - V_X^2\right]^{3/2} \qquad \ldots (12)$$

Further consideration as to the theory of heat and mass transfer by natural convection and combined natural convection and forced air flow through a rectangular opening in a vertical partition is detailed in a paper by one of the authors (Shaw, 1971).

3. EXPERIMENTAL PROCEDURE

3.1. Test Area

These tests were carried out at the BSRU Experimental Ward Unit, Hairmyres Hospital, East Kilbride, in the isolation rooms of the intensive care area which was located in the north-west corner of the ward unit. The two rooms are the same ones as studied by Baird and Whyte (1969). A plan of the two isolation rooms and their associated vestibule is given in Fig. 3.

These rooms open into a common air lock or vestibule. The air was supplied to room A by a high level grille. Room B and the vestibule were supplied by ceiling diffusers, these diffusers being shielded in order to prevent interference of the air movement pattern through the door by the supply airstream. 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 A to reduce any heat loss through the window.

3.2. Instrumentation

The variables observed in the air movement tests and the means by which they were measured were as follows:

1. Air volumes being supplied and extracted to each room by the air-conditioning system. These volumes were measured by averaging pressure tube flowmeters (Ma, 1967).
2. Air temperatures in the rooms and doorways. Copper-constantan thermocouples in conjunction with a multi-point recorder were used.
3. Air velocities in the doorways. These were measured by hot wire anemometers (Simmons, 1949) in conjunction with a 40 point scanner and recorder.
4. Airflow direction through the doorways. This was determined by cigarette smoke.

Fig. 4. 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.

3.3. Scope of tests and procedure

The four-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 in the plant room using Ma's balancing method to appropriate values and maintained at these throughout the test series. Supply and extract volumes to the rooms were varied from 0·05 to 0·30 m^3/s in steps of 0·05 m^3/s. 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.

Heating or cooling of the supply air to each room was controlled from the plant room. If additional heating was required, the radiator in that particular 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. Air temperature differences ranged from 0 to 12°C. As 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, the vestibule and the doorways. These grids consisted of a "Meccano" strip with ten hot wire 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 tested and these were set up by blanking off the door openings with wooden boards. The height of the doors was kept constant at 2·05 m while the door widths were varied from 0·10 m to 1·2 m in the case of room A and 0·10 m to 1·40 m in the case of room B.

Each test began by taking the temperature at the grids in both isolation rooms and the 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 determined by cigarette smoke 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 heads were placed sideways. Five sets of thermocouple and anemometer readings were recorded for that particular vertical section 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, five vertical grid position readings were obtained for the 1·20 m and 1·40 m openings, three positions for the 0·90 m, two for the 0·50 m and one position for the 0·10 m. 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.

3.4. 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 fitted the best curve (linear, quadratic and cubic) to the results, and gave a statistical percentage fit, along with the information required to work out the amount of air flowing in and out of the door, which is the volume beneath the surface of the curve. A typical printout for a 0·90 m door opening and a temperature difference of 1·07°C is shown in Fig. 4. This shows the "isovel" diagrams of the air movement in the doorway as drawn by the computer for the linear, quadratic and cubic trend surfaces. The reference contour for these diagrams was taken as 0 and represented zero velocity, i.e. neutral zone. Contour intervals were taken as 0·02 representing 0·02 m/s.

The velocities from the printouts can be determined as shown in the list below.

Printout Value	*Velocity Range (m/s)*
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

4. 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 difficulty was caused by the fact that the temperature in rooms varied both from floor to ceiling and wall to wall. In our case 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.

4.1. The problems due to stratification

The problem of stratification is demonstrated diagrammatically in Fig. 5 where the use of a radiator to achieve high temperature differentials between the two areas could cause convective currents which rise and force their way out of the top of the doorway. We have also postulated in our 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 Fig. 5 would possibly generate an air flow profile through the door as shown. If this was so it would not affect the practical application of the results but it could complicate the theory. High temperature differentials are, however, somewhat unusual in practice, as field tests in hospital isolation and treatment areas showed the differentials to be on average around 1·0°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 (Fig. 6). This action in turn stabilised the system, ensuring that the neutral zone would be at the mid-point of the doorway. This can be seen in Fig. 7 which shows the temperature gradients and velocity profiles, indicating the neutral zone for both small temperature differential (0·40°C) and a large temperature differential (4·67°C). Although all the results are not as good as those presented, inspection of our results, allowing for the screening of one doorway by air passing out of 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.

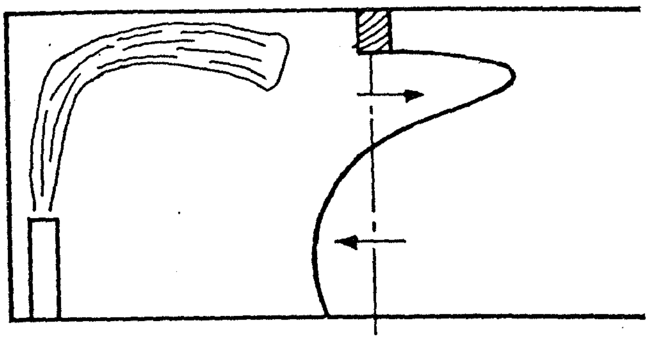

Fig. 5. Stratification due to the use of a radiator in one of the rooms.

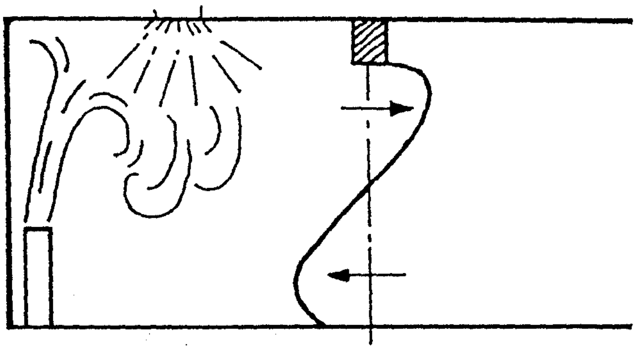

Fig. 6. Disruption of stratification due to interaction of incoming air through diffuser.

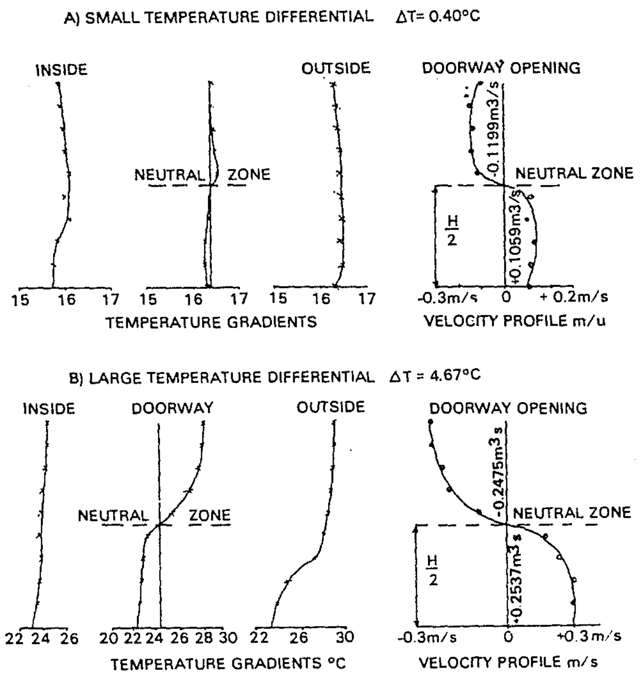

Fig. 7. Temperature gradients and velocity profiles in indicating position of neutral zone.

4.2. Measurement of temperature

Where to measure temperature was a problem to which at the initiation of the tests there was no satisfactory answer. We resolved therefore to measure what we considered sufficient temperatures to enable us to cover all eventualities. These were 10 readings

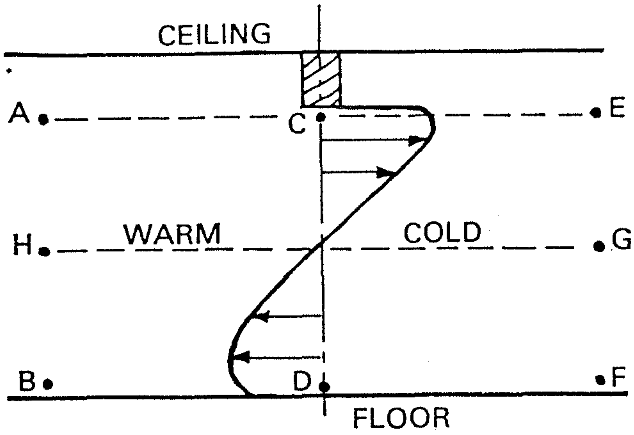

Fig. 8. Temperature measuring positions.

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 Fig. 8.

1. Temperature difference between top and bottom of the doorway (*C–D*).
2. Difference between top of the warm room and bottom of the 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 temperatures in each room *H–G*.
5. Difference between the mean of 10 readings 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, as measured on grid *C–D*.

The decision as to what was the best temperature measuring point was resolved by determining the temperature which could be used to most accurately predict 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 of:

a) the six temperature differences as stated above
b) door area
c) room number
d) volume of air supplied 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 temperatures which could predict 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. A similar analysis was performed by treating the door areas separately for each temperature, the following order being obtained: 6, 4, 5, 1, 3 and 2.

It was finally decided to use differential (4) which appears third in one analysis and second in the other. This is a simple temperature differential to measure in practice, requiring only two temperature readings, one at mid-door height in each room away from the influence of the door.

To indicate the temperature difference between the six differentials used, the following results from one of the tests are quoted:

temperature differential (1) 5·59, (2) 6·03, (3) 2·59, (4) 4·67, (5) 3·39 and (6) 4·60°C.

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 differentials are those of the difference between the top and bottom of the door, this being thought to be the best differential with respect to the theory, and that of the difference between the room temperatures taken at a level of half the door height. The latter temperature differential was chosen but any other could have been used. This would, however, require a change in the coefficient of discharge.

5. TEST RESULTS

5.1. Natural convection

In accordance with our conclusions in 4.2, the temperature differential used in the analysis of the results was that of the temperature difference between the middle of one room and the middle of the other. This was thought to be a convenient differential to measure in practice.

The experimental transfer volumes for the various door openings are presented in Fig. 9. 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 1·40 m wide door were considerably lower than predicted volumes. This divergence from theory may be explained in two ways; firstly by the interaction of the air flow through the two doors when fully opened, and secondly that the coefficient of discharge could vary with area of opening.

It can be seen from Fig. 3 that the two test doors were very close to one another, and that when both openings were at their maximum area the air flow out of one could influence the amount of air flow out 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 amounts 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 A was adjacent to the air flowing out of room B, i.e. if flowing out of the top of both doors or out of the bottom of both doors, then this fact was significant. Under these conditions the air flow out of room A

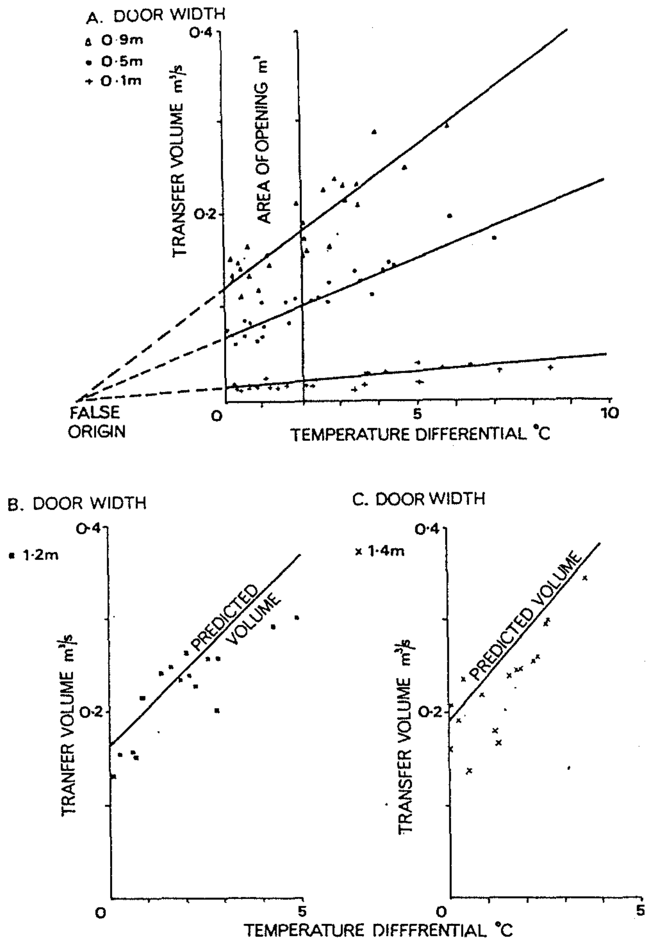

Fig. 9. Experimental natural convective transfer volumes (m^3s).

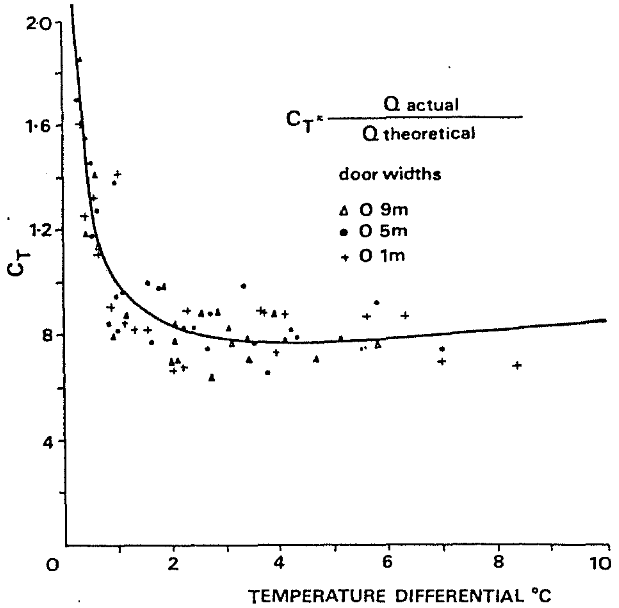

Fig. 10. Coefficient of temperature (C_T).

masked the air flow out of room B and subsequently reduced the volume of air transfer out of that room. By using the multiple regression results, the corrected values for the 1·40 m door (room B) were replotted.

The results for the 1·20 m wide and 1·40 m wide doors are shown in Figs. 9B and 9C and, as can be seen, they are 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 Fig. 10) and we are therefore inclined to consider that the discrepancy is still due to the 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 following theoretical equation. (See section 2.1.)

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

The coefficient values were found to be primarily a function of temperature differential, the door area not being significant. These coefficients are shown in the range of areas which we studied. It was therefore decided to refer to the coefficient as the coefficient of temperature. Fig. 10 is interesting in the fact that from about 1–2°C differential downwards, the value of the coefficient increase asymptotically with the coefficient axis. The reason for this trend may be explained as follows: The convective transfer volumes in or out of the doorway at zero temperature differential (by extrapolation) for each door area (Fig. 9) are listed in Table 2. By dividing these values by half the door area, a mean velocity in or out of the doorway may be arrived at.

Table 2. Transfer volumes at zero temperature differential

Door area	Transfer volume	Mean velocity
2·87 m²	0·1905 m³/s	0·1330 m/s
2·46 m²	0·1640 m³/s	0·1330 m/s
1·845 m²	0·1230 m³/s	0·1330 m/s
1·025 m²	0·0685 m³/s	0·1330 m/s
0·205 m²	0·0137 m³/s	0·1330 m/s

This results in a mean velocity of 0·1330 m/s (26 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 0·1016–0·1524 m/s (20–30 ft/min), this explains why the coefficient does not remain constant around 0·8. Above 4°C temperature differential the coefficient rises again, very slowly this time, reaching a value of unity about 20°C differential and continuing to rise. This is not in fact shown in Fig. 10.

The above results show that by use of the theoretically derived equation and an experimentally obtained coefficient of discharge it is possible to obtain the amount of air transfer through doorways due to natural convection.

5.2. Combined natural convection and forced air flow

The leakage transfer volumes for combined natural convection and forced air flow are presented in Fig. 11. Positive ventilation tests did not tend to be so consistent as the balanced tests and this may be due to the smaller number of samples which were taken. As stated previously the 1·20 m and 1·40 m wide door results are erroneous due to the extraneous influences, and are therefore not included in this paper.

When air is being forced through an opening across which there is a temperature difference, the amount of air escaping into the room (Q_L) against the flow is defined by theory as:

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

On analysis of the positive tests in conjunction with 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 (12), 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,

i.e. $C = \phi(Q_X/A, \Delta T)$

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

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. The values of these coefficients are plotted in Fig. 12. By use of these values in the theoretically derived equation it is possible to estimate the amount of air escaping into the room against a known flow of air.

6. DISCUSSIONS AND CONCLUSIONS

6.1. Natural convection

When two spaces are connected by an opening and there is no designed flow of air between the spaces, the amount of air transfer caused by temperature difference is:

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

By measurement of the actual air flow through doorways by means of hot wire anemometers we established that the coefficient of discharge, for temperature differentials of around 1–10°C is about 0·8.

By theory it is expected that no transfer of air will exist when there is no temperature differential. However, in the practical situation as in the ventilated isolation rooms we studied this is not so, the turbulence of the adjoining spaces making it impossible to prevent transfer.

This means, therefore, that because this air transfer is not predicted by theory, to adapt the theory to practice, the coefficient of discharge increases from 0·8 at approximately 1°C and tends towards infinity as the temperature difference drops to zero. For this reason temperatures between critical areas should be kept as low as possible. The significance of this is shown by the fact that we have found that a temperature difference of 4°C will transfer each way through a 0·90 m wide × 2.05 m high (2·9 ft × 6·70 ft) doorway 0·25 m³/s of air (530 cfm) and 0·38 m³/s (805 cfm) through a door 1·40 m wide (4·5 ft).

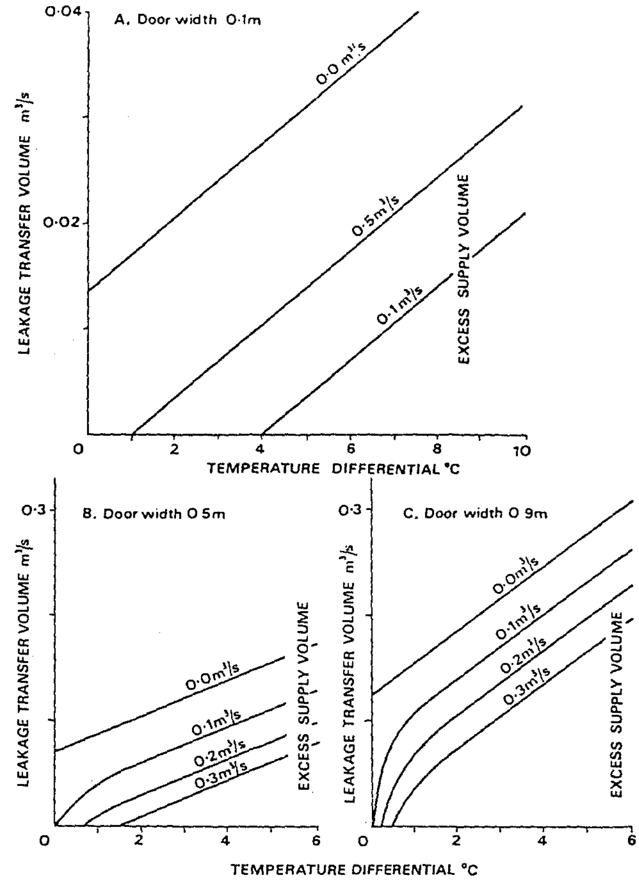

Fig. 11. Leakage transfer volume for combined natural convection and forced air flow.

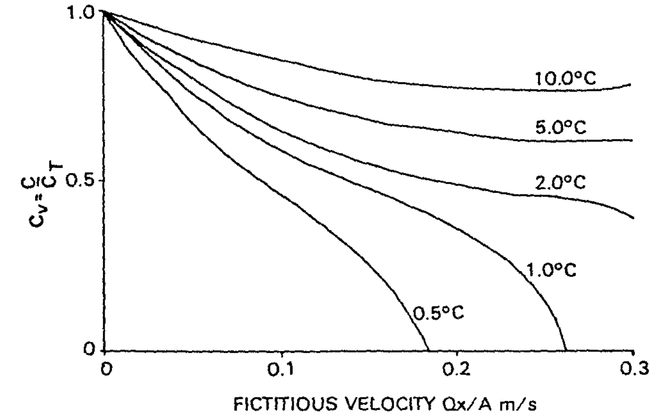

Fig. 12. Coefficient of fictitious velocity (C_V).

6.2. Combined natural convection and forced air flow

In order to prevent the unwanted transfer of air across doorways due to temperature differences, it is necessary (as in operating rooms) to supply excess air to the room which requires to be isolated. This excess air will then pass through the doorway and counteract the flow of the incoming contaminated air.

When air is being forced through an opening across which there is a temperature difference, the amount of air escaping into the room (Q_L) against the flow is defined by theory as:

$$Q_L = C_T C_V \frac{W}{3} \frac{1}{g(\Delta\rho/\bar{\rho})}\left[g\frac{\Delta\rho}{\rho}H - V_X^2\right]^{3/2}$$

By use of the coefficients obtained from this study,

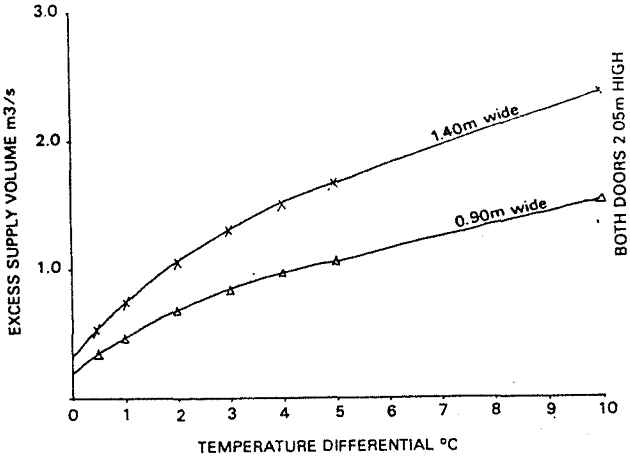

Fig. 13. Supply volume required to completely isolate an area (m^3/s).

the amount of air required to prevent unwanted airflow across an opening may be calculated.

It may then be established that the amount of air required to pressurise an area to prevent air passing through an open doorway would be extremely large indeed, if the temperature differences were not kept small. These supply volumes are shown in Fig. 13 which gives the volumes required for both a 0·90 m and 1·40 m wide door over a 0–10°C temperature differential. In the operating room situation with an open doorway 1·40 m wide and a difference in temperature of 1°C the amount of air required to ensure no adverse flow would be around 0·75 m^3/s (1600 cfm), a 2°C temperature differential would require 1·05 m^3/s (2200 cfm). This would mean that a flow of about 0·25 m^3/s for each square metre of doorway is required (50 cfm per square foot) for a temperature difference of 1°C.

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