C117/71 HEAT AND MASS TRANSFER BY NATURAL CONVECTION AND COMBINED NATURAL CONVECTION AND FORCED AIR FLOW THROUGH LARGE RECTANGULAR OPENINGS IN A VERTICAL PARTITION

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Original theory for combined natural convection and forced air flow across a rectangular opening in a vertical partition has been postulated and generalized to include both heat and mass transfer. Experiments for natural convection, and combined natural convection and forced air flow, were carried out with openings 2·05 m high and from 0·10 to 0·90 m wide with air as the convecting fluid. Temperature differentials were in the order of 0–12 degC and the supply and extract volumes in the range 0·0–0·30 m³/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.

INTRODUCTION

UNTIL 1960, studies of natural convection were primarily concerned with problems of heat transfer involving vertical and horizontal plates and bodies of varying shape. Schmidt (1 †), in a review in 1961, mentioned a type of natural convection which, until 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 (2) had treated the case of multiple openings in a wall and had 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 generalized for all fluids.

No direct measurements had been made to substantiate or extend the theory and this may partly be explained by measurement difficulties and by the fact that opening sizes of practical importance were too large to be investigated in a laboratory. However, in the last 10 years four major sources have published theoretical and experimental work relating to this type of convection. 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 117.1 compares these pertinent variables as studied by each source.

It may be seen from Table 117.1 that no previous research has been carried out with small temperature differentials. Also, there were no results for the effect, within a room, of excess pressure 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 paper does, however, consider these variables and as a result considerably widens the knowledge of convection through openings in vertical partitions.

Notation

C	Coefficient of discharge.
C_T	Coefficient of temperature.
C_V	Coefficient of fictitious velocity.
c_p	Specific heat of fluid.
c_1, c_2	Concentration, e.g. moisture content.
D	Diffusion coefficient of mass transfer.

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† *References are given in Appendix 117.1.*

Table 117.1. Comparison of variables as studied in previous research

Source	Convection	Area, m^2	Height, m	Range of ΔT, degC
Brown and Solvason 1962 (3)	Natural Natural plus forced air flow	0·005 81–0·091 90	0·0762–0·3048	8–47
Graf 1964 (4)	Natural Natural plus forced air flow	—	—	Theory
Tamm 1966 (5)	Natural	—	—	Theory
Fritzsche and Lilienblum 1968 (6)	Natural	4·5	2·5	12–41·5
Shaw 1971	Natural Natural plus forced air flow	0·205–1·845	2·05	0–12

D_h Hydraulic diameter of doorway $[= 2WH/(W+H)]$.
g Acceleration due to gravity.
H Opening height.
h Heat transfer coefficient.
h_m Mass transfer coefficient.
k Thermal conductivity of fluid.
$\dot{m}$ Mass transfer rate through opening.
$\dot{q}$ Heat transfer rate through opening.
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, Q_x 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.
V_b Velocity defined in equation (117.19).
W Width of opening.
μ Dynamic viscosity.
ν Kinematic viscosity.
ρ Fluid density.

Dimensionless groups

Fr_Δ Densimetric Froude number $\{= V/[gH(\Delta\rho/\bar{\rho})]^{1/2}\}$.
Gr Grashof number based on density differences $[= g\,\Delta\rho H^3/\bar{\rho}\nu^2]$.
Nu Nusselt number $[= hH/k]$.
Pr Prandtl number $[= c_p u/k]$.
Re Reynolds number $[= \bar{\rho} V_b D_h/\mu]$.
Sc Schmidt number $[= \nu/D]$.
Sh Sherwood number $[= h_m H/D]$.
Sw Dimensionless group $\left| = \frac{Re^3}{Gr}\frac{H^3}{D_h^3} = \frac{\mu V_b^3}{\nu^2 g\,\Delta\rho}\right|$.

THEORY

Theory of the volumetric exchange of air due to natural convection through a rectangular opening in a vertical partition (3)

Consider a large sealed enclosure consisting of rooms 1 and 2 as shown in Fig. 117.1. 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 a level Z below the centre-line will be

$$P_1 = P_0+\rho_1 gZ \quad . \quad . \quad (117.1)$$

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

$$P_2 = P_0+\rho_2 gZ \quad . \quad . \quad (117.2)$$

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

$$P_2-P_1 = (\rho_2-\rho_1)gZ \quad . \quad . \quad (117.3)$$

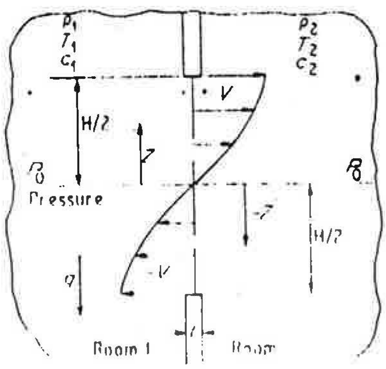

Fig. 117.1. Schematic representation of natural convection across an opening in a vertical partition

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

$$h_a = \frac{p_2 - p_1}{\bar{\rho}} Z = \frac{\Delta p}{\bar{\rho}} Z$$

in which $\bar{\rho}$, the mean density, is written

$$\bar{\rho} = \frac{\rho_1 + \rho_2}{2} \quad . \quad . \quad . \quad (117.4)$$

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.

$$V = (2gh_a)^{1/2} = \left[2g\left(\frac{\Delta p}{\bar{\rho}}\right)Z\right]^{1/2} \quad (117.5)$$

where V is the air velocity.

Now $Q = CAV$, where Q is the rate of volumetric discharge, C the coefficient* (unknown as yet and to be determined from tests), and A the area of the opening.

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

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

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

$$\boxed{Q = C\frac{W}{3}\left[g\left(\frac{\Delta p}{\bar{\rho}}\right)\right]^{1/2} H^{3/2}} \quad . \quad (117.6)$$

With the flow Q is now associated the heat transfer rate

$$\dot{q} = Q\bar{\rho}c_p(T_1 - T_2) \quad . \quad . \quad . \quad (117.7)$$

and the mass transfer rate, i.e. moisture content,

$$\dot{m} = Q\bar{\rho}(c_1 - c_2) \quad . \quad . \quad . \quad (117.8)$$

where c_p is the specific heat.

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

$$h = \dot{q}/[WH(T_1 - T_2)] \quad . \quad . \quad (117.7a)$$

and

$$h_m = \dot{m}/[WH\bar{\rho}(c_1 - c_2)] \quad . \quad . \quad (117.8a)$$

equations (117.7) and (117.8) lead to the following equations in terms of dimensionless variables:

for heat transfer,

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

$$= \frac{C}{3}Gr^{1/2}Pr \quad . \quad . \quad . \quad . \quad . \quad (117.9)$$

for mass transfer,

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

$$= \frac{C}{3}Gr^{1/2}Sc \quad . \quad . \quad . \quad . \quad (117.10)$$

where the symbols are as defined in the Notation.

Equations (117.9) and (117.10) cannot be exact for all conditions owing to neglect of viscosity in equation (117.5) and neglect of thermal conductivity and diffusivity in equations (117.7) and (117.8). The effect of these properties is considered in detail by Brown and Solvason (3). 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 (117.9) and (117.10).

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 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 (Fig. 117.2). In this case the enclosures are not sealed, air being supplied to one and extracted from the other. In room 1 the pressure P, at a level Z below the centre-line, will be

$$P_1 = P_0 + \rho_1 gZ + P_x \quad . \quad . \quad (117.11)$$

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

$$P_2 = P_0 + \rho_2 gZ \quad . \quad . \quad (117.12)$$

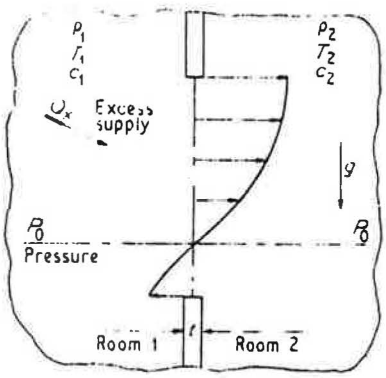

Fig. 117.2. Schematic representation of combined natural convection and forced air flow across an opening in a vertical partition

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

$$P_2 - P_1 = (\rho_2 - \rho_1)gZ - P_x \quad . \quad (117.13)$$

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 is

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

and the supply air pressure is

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

Therefore, from equation (117.13)

$$h_a = h_1 - h_2 \quad . \quad . \quad . \quad (117.14)$$

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.

$$\begin{aligned} V &= (2gh_a)^{1/2} \\ &= \left[2g\left(\frac{\Delta\rho}{\bar{\rho}} Z - \frac{V_x^2}{2g}\right)\right]^{1/2} \\ &= \left[2g\left(\frac{\Delta\rho}{\bar{\rho}}\right) Z - V_x^2\right]^{1/2} \quad . \quad (117.15) \end{aligned}$$

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

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

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

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

therefore,

$$Q_L = C.W \frac{1}{3} \cdot \frac{1}{g(\Delta\rho/\bar{\rho})}\left[g\left(\frac{\Delta\rho}{\bar{\rho}}\right) H - V_x^2\right]^{3/2} \quad . \quad (117.16)$$

It has already been shown that with natural convection on its own, the Nusselt number may be expressed as a function of the Grashof and Prandtl numbers, i.e. $Nu = \phi(Gr, Pr)$. 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 the Nusselt number could be expressed in terms of both natural and 'forced' convection, 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 we now associate the heat transfer rate

$$\dot{q} = Q_L \bar{\rho} c_p (T_1 - T_2) \quad . \quad . \quad (117.17)$$

and the mass transfer rate

$$\dot{m} = Q_L \bar{\rho}(c_1 - c_2) \quad . \quad . \quad . \quad (117.18)$$

where c_p is the specific heat.

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

$$h = \dot{q}/[WH(T_1 - T_2)] \quad . \quad (117.17a)$$

and

$$h_m = \dot{m}/[WH\bar{\rho}(c_1 - c_2)] \quad . \quad (117.18a)$$

equations (117.17) and (117.18) lead to the following equations in terms of dimensionless variables:

for heat transfer,

$$\begin{aligned} Nu &= \frac{hH}{k} = \frac{C}{3} \cdot \frac{c_p \mu}{k}\left(\frac{\mu V_h^3}{\rho^2 g\,\Delta\rho}\right) \\ &= \frac{C}{3} Pr . Sw \quad . \quad . \quad . \quad . \quad (117.19) \end{aligned}$$

where V_h is the equivalent velocity within the square brackets of equation (117.16) and *Sw* is a dimensionless group.

From equation (117.19) it is found that the group *Sw* is in fact equal to the value

$$Sw = \frac{Re^3}{Gr} \cdot \frac{H^3}{D_h^3} \quad . \quad . \quad . \quad (117.20)$$

where D_h is the hydraulic diameter of the doorway. Thus the dimensionless group *Sw* is a function of the Reynolds 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 nonetheless 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

$$Nu = \frac{C}{3} Pr \frac{Re^3}{Gr} \cdot \frac{H^3}{D_h^3} \quad . \quad . \quad (117.21)$$

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

$$Sh = \frac{h_m H}{D} = \frac{C}{3} Sc . Sw = \frac{C}{3} Sc \frac{Re^3}{Gr} \cdot \frac{H^3}{D_h^3} \quad (117.22)$$

EXPERIMENTAL

Test area

The test area was situated at the Experimental Ward Unit at Hairmyres Hospital, East Kilbride, the tests being carried out in the isolation rooms of the intensive care area (Fig. 117.3). The rooms opened into a common air lock/vestibule. Radiators were positioned in each of the three rooms to supply additional heating to that of the supply air, and a sheet of expanded polystyrene was placed over the window in room 'A' to reduce any heat loss through the window.

Instrumentation

The mechanical supply and extract volumes to each room were measured and balanced with averaging pressure flowmeters in accordance with Ma's method of balancing (7). Air temperatures in the room and doorways were measured using copper–constantan thermocouples. Hot wire anemometers were used to measure the air velocities in the doorway. Air flow direction through the doorways was initially determined using cotton wool swabs soaked in titanium tetrachloride, but it was found that the smoke propagated rusting and cigarette smoke was subsequently used to determine the air direction.

Scope of tests and procedure

Tests were conducted for single rectangular openings of the following nominal dimensions: 0·90, 0·50, and 0·10 m wide, all areas being 2·05 m high. These different door areas were set up by blanking off the door openings with wooden boards. 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. Air temperature differences ranged from 0 to 12 degC. Owing to the massiveness of the test apparatus periods of up to 3 hours were required to reach equilibrium conditions, especially when large temperature differentials were being set up.

When the air temperatures within each area had stabilised, a grid consisting of a vertical Meccano strip was suspended from the top of each opening. These grids had 10 anemometers and 10 thermocouples fixed at equal intervals down their length and were suspended in such a manner that the air velocities and temperatures at any vertical section could be measured. Air flow direction at each point on the grid was then determined and the anemometer heads adjusted accordingly to face the oncoming air flow. If the direction was not definite, i.e. in the neutral zone, the anemometer head was placed sideways. The anemometer and thermocouple readings were then recorded for that particular vertical section, five sets of readings being 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 opening and the air temperatures at the openings, three vertical grid position readings were obtained for the 0·90 m opening, two positions for the 0·50 m, and one position for the 0·10 m. Once this procedure had been completed for a specific opening 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 opening areas.

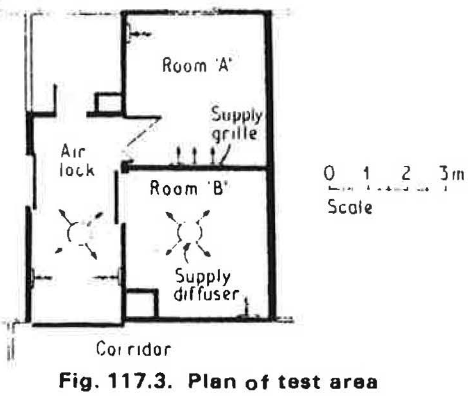

Fig. 117.3. Plan of test area

TEST RESULTS

The velocity readings that had been recorded during the tests were analysed with a trend surface analysis program which fitted the best curve (linear, quadratic, and cubic) to the results, thus calculating the volume of air flowing in and out through the opening. The program also printed out isovel diagrams of the air movement in the doorway.

Natural convection

The temperature differential which was used in the analysis of the results was that of the temperature difference between top and bottom of the opening. 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 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 (117.6). The coefficient values were found to be primarily a function of temperature differential, the door area not being significant (Fig. 117.4). It was therefore decided to refer to the coefficient as the coefficient of temperature. An interesting feature of Fig. 117.4 is that from about 4 degC downwards the value of the coefficient increases asymptotically with the coefficient axis. The reason for this trend may be explained as follows. The convective transfer volumes at zero temperature differential (by extrapolation) for each door area are listed in Table 117.2. By dividing these values by half the door area a mean velocity may be obtained. This results in a mean velocity of 0·1362 m/s (27 ft/min) for any door area. As the free air velocity, or turbulence, within the ventilated room is generally quoted as being in

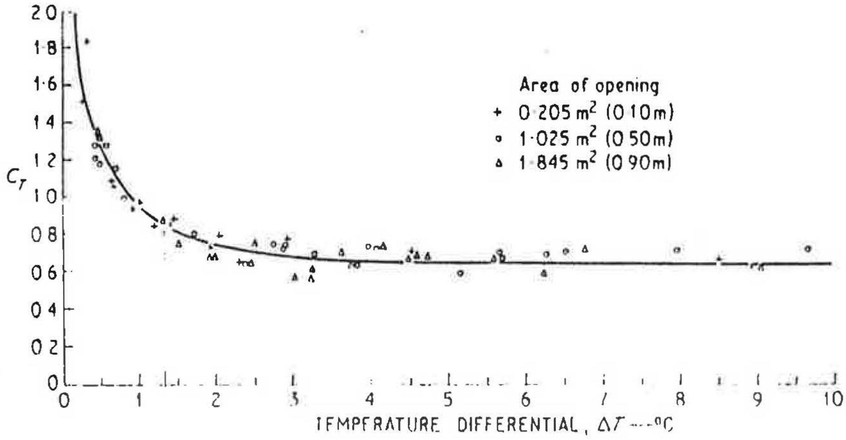

Fig. 117.4. Coefficient of temperature (C_T)

Table 117.2. Transfer volumes at zero temperature differential

Door area, m^2	Transfer volume, m^3/s	Mean velocity, m/s
1·845	0·1250	0·1360
1·025	0·0700	0·1362
0·205	0·0140	0·1362

the range 0·1016–0·1524 m/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 degC temperature differential the coefficient rises again, very slowly this time, reaching a value of unity at about 50 degC differential and continuing to rise. This is not, in fact, shown in Fig. 117.4. Although the reason for this trend is not at present apparent, it compares favourably with limited results of Fritzsche and Lillienblum (6) working in the region of 20–30 degC differential.

Experimental results for heat transfer are given in Fig. 117.5, 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 (3) are also shown. With an opening 2·05 m high, the upper theoretical curve, and a temperature differential of 10 degC, the Grashof number equals $1{\cdot}3\times10^{10}$. As can 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 degC, $Gr = 6{\cdot}76\times10^{10}$. Comparing these values with

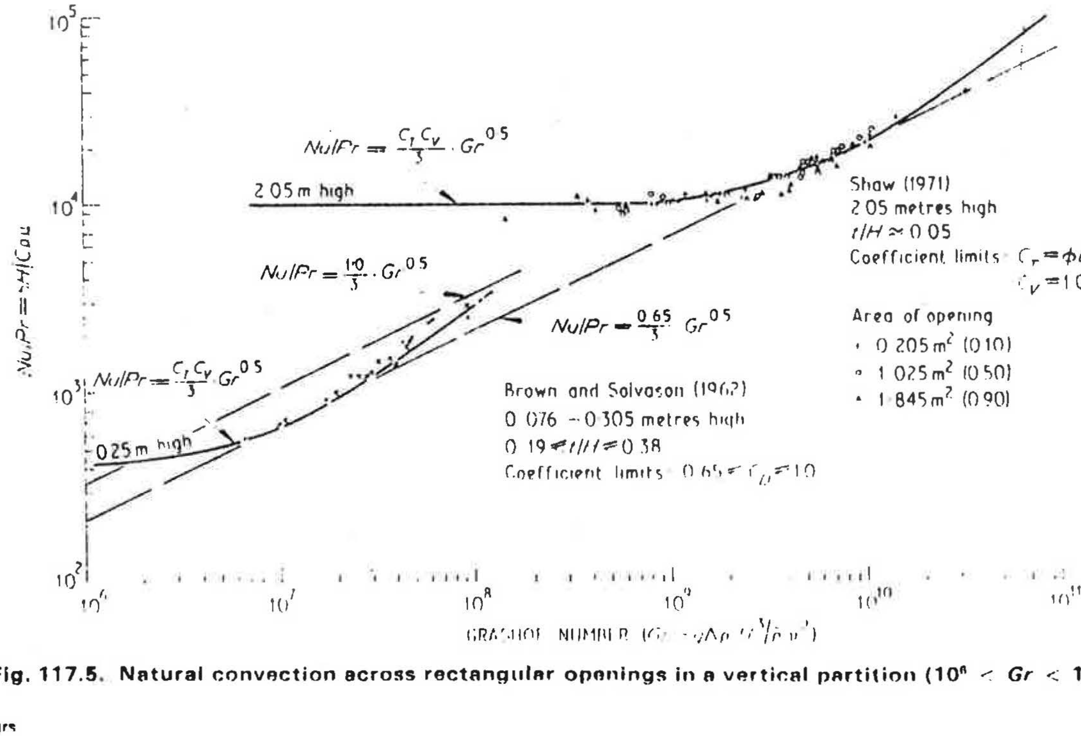

Fig. 117.5. Natural convection across rectangular openings in a vertical partition ($10^6 < Gr < 10^{11}$)

the results of Brown and Solvason it may be seen that a theoretical curve for an opening 0·25 m 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 degC, as this was the region in which Brown and Solvason were working. With this height of opening and a differential of 10 degC, $Gr = 2{\cdot}35\times10^7$ while at 40 degC, $Gr = 1{\cdot}23\times10^8$. This is in fact the breakaway 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 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 temperatures 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.

Brown and Solvason (1962):

$$Q = C_D \frac{W}{3} \cdot H^{3/2}(g)^{1/2}(\Delta\rho/\bar{\rho})^{1/2}$$

Gral (1964):

$$Q = \frac{W}{3} \cdot H^{3/2}(g)^{1/2}(\Delta\rho/\bar{\rho})^{1/2}$$

Tamm (1966):

$$Q = \frac{W}{3} H^{3/2}(g)^{1/2}(\Delta\rho/\rho_s)^{1/2}$$

Fritzsche and Lilienblum (1968):

$$Q = C_T \frac{W}{3} H^{3/2}(g)^{1/2}(\Delta\rho/\rho)^{1/2}$$

Shaw (1971):

$$Q = C_T . C_V \frac{W}{3} H^{3/2}(g)^{1/2}(\Delta\rho/\bar{\rho})^{1/2}$$

The existence of an excess pressurization 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.

Combined natural convection and forced air flow

On 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 (117.16) 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, 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 started at unity for a system with no excess volume, decreasing as the amount of excess volume increased (Fig. 117.6).

Experimental results for heat transfer are shown in Fig. 117.7, 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 pressure reduces the heat transfer rate across the opening. Once again the type of flow is considered to be in the turbulent regime.

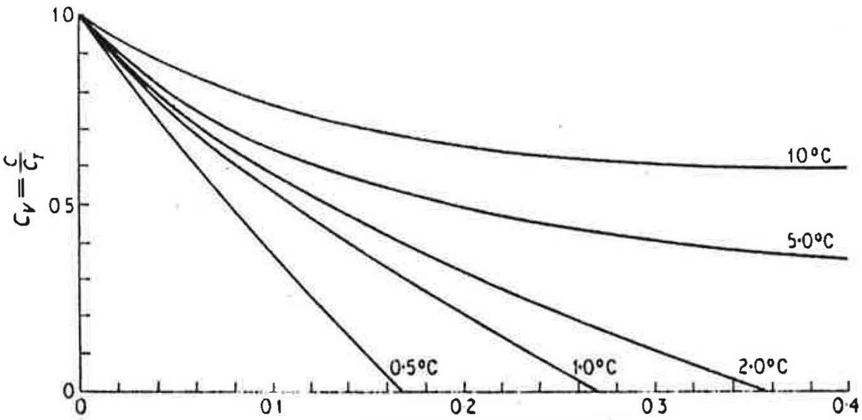

Fig. 117.6. Coefficient of fictitious velocity (C_V)

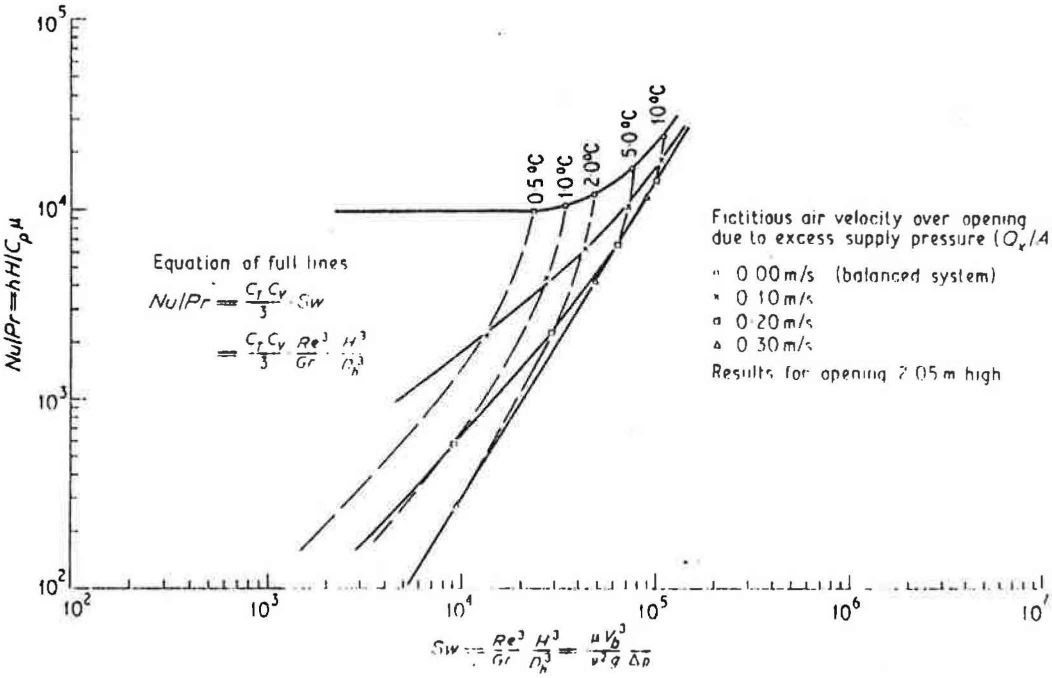

Fig. 117.7. Combined natural convection and forced air flow across rectangular openings in a vertical partition ($10^3 < Sw < 10^4$)

DISCUSSION AND CONCLUSIONS

The test results for air in natural convection, and combined natural convection and forced air flow, across rectangular openings in vertical partitions are in good agreement with theory.

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.

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 numbers. This was indeed found to be the case, the relationship being in the ratio, $Sw = (Re^3/Gr)(H^3/D_h^3)$, where Sw is a dimensionless group and the other significant terms are the height and hydraulic diameter of the opening.

An interesting analogy to the subject of this paper 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 (8), 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.

$$Fr_\Delta . Re = \left(\frac{g\, \Delta\rho H^3}{\bar{\rho}\nu^2}\right)^{1/2} \quad . \quad (117.23)$$

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 (117.23) is in fact the Grashof number ($Gr^{1/2}$) of equation (117.9), 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.

$$Sw = \frac{Re^3}{Gr} \cdot \frac{H^3}{D_h^3} = \left(\frac{\mu V_b^3}{\nu^2 g\, \Delta\rho}\right) = Re . Fr_\Delta^2 \frac{H}{D_h} \quad (117.24)$$

There exist several fields of study to which the results of this paper may be applied. First there is the problem in public buildings, shops, supermarkets, and restaurants of convective air currents causing unpleasant draughts and loss of heat at doorways ~~where~~ where is normally a heavy concentration of pedestrian traffic. The general movement of air within buildings is also of importance whether it be naturally ventilated or air conditioned, and under special conditions, such as a fire within the building, it is essential to know the movement of smoke up vertical shafts—especially in tall buildings. The problem concerned with cold storage rooms, with temperature differentials in the range 30–40 degC, is that of heat and mass transfer through the access doorway resulting in greater running costs. 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. With the theory and results of this paper it is now possible to determine accurately the volumetric exchange of air, hence heat and mass transfer, for all these situations.

The importance of the airborne route of infection in critical areas within hospitals has been shown by numerous workers for several decades. In particular, various papers on the subject of convective transfer through doorways of bacteria such as *Staph. aureus*, an important group of bacteria which causes wound infection, have been published in the last 10 years. It is now possible to predict the volumetric exchange through doorways under certain conditions and, hence, the isolation efficiency of the system. It is also possible to determine the amount of excess ventilation required to completely isolate the critical area from the rest of the hospital.

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APPENDIX 117.1

REFERENCES

(1) SCHMIDT, E. 'Heat transfer by natural convection', *Int. Heat Transfer Conf.*, University of Colorado, 1961.

(2) EMSWILER, J. E. 'The neutral zone in ventilation', *Trans. Am. Soc. Heat. Vent. Engrs* 1926 **32**, 59.

(3) BROWN, W. G. and SOLVASON, K. R. 'Natural convection through rectangular openings in partitions—vertical partitions', *Int. J. Heat Mass Transfer* 1962 **5**, 859.

(4) GRAF, A. 'Consideration of the air exchange between two rooms', *Schweiz. Bl. Heiz. Lüft.* 1964 **31** (1), 22.

(5) TAMM, W. 'Cold losses through openings in cold rooms', *Kaltetechnik-Klimatisierung* 1966 **18** (42), 142.

(6) FRITZSCHE, C. and LILIENBLUM, W. 'New measurements for the determination of cold losses at the doors of cold rooms', *Kaltetechnik-Klimatisierung* 1968 **20** (9), 279.

(7) MA, W. Y. L. 'The averaging pressure tubes flowmeter for the measurement of the rate of airflow in ventilation ducts and for balancing of air-flow circuits in ventilating systems', *J. Instn Heat. Vent. Engrs* 1967 (No. 34, February), 327.

(8) BARR, D. I. H. 'Densimetric exchange flow in rectangular channels', *Houille blanche* 1963 (No. 7), 739.