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The fluid mechanics of natural ventilation

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Abstract

Ventilation of buildings is a topic close to our experience, but knowledge and understanding of the airflow within a building is usually scanty. Even in buildings with purpose built mechanical ventilation or air conditioned systems, designers use crude rules to specify the ventilation, and the result is often unsatisfactory. I will review current understanding and research on ventilation concentrating on the details of the air flow. I will establish some general principles, such as why layered stratification occurs in steady flows and why some flows are intrinsically unsteady.

Introduction

In this paper I will describe the fluid mechanics of building ventilation. The aim is to present the fundamentals of the flows that occur in buildings, to describe how they are set up and the effects they have both on the occupants' comfort and the energy consumption of the building.

Energy usage in buildings corresponds to a significant proportion of total national energy consumption, particularly in countries where the proportion of air-conditioned buildings is large. In the US, about 30% of the total energy consumption is used in non-domestic buildings, and of that fraction about 30% is used on heating and cooling.

Possibly a more important economic factor is the productivity of staff in buildings. There is considerable anecdotal evidence, but few hard data, that staff in many air-conditioned buildings are generally unhappy with their indoor environment. This is reflected in increased absenteeism and other possible reductions in productivity. Since labour costs are generally the largest costs for industry, improvements in air quality and satisfaction of building occupants with their environment offer large potential financial gains to industry.

Despite the fact that there is little hard evidence to support these claims about productivity, there is a growing belief among architects and clients that the totally sealed and conditioned building is not the optimum design. The capability of occupants to affect their environment by opening a window is believed to provide a significant improvement. This belief, along with pressures to reduce energy consumption, has led to an increased use of natural ventilation.

Recently a number of high-profile, naturally ventilated buildings have been constructed. An example is shown in figure 1, a new low-energy office building for BRE Ltd. This building uses solar chimneys, designed to collect solar radiation, to enhance the stack-driven (buoyancy-driven) ventilation. Figure 2 shows a naturally ventilated brewery in Malta. This building uses night-cooling to provide a constant temperature in the brewing hall. The external facade, shown in figure 2, is the outer skin of a double skinned building. During the day the vents, such as the ventilation towers, are closed, so that the brewing hall is isolated from the day-time high temperatures. At night they are opened so that air can cool the inner building. The brewing hall has a large exposed ceiling with high thermal mass, which cools at night and absorbs the excess heat during the day, providing

Figure 1: The low-energy office building for BRE Ltd., showing the solar chimneys that collect solar radiation and enhance the stack-driven ventilation. Courtesy of James Fisher, BRE Ltd.

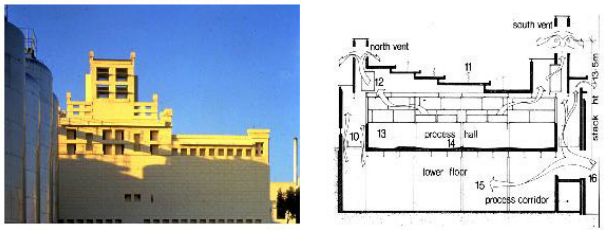

Figure 2: The Malta Brewery designed by Brian Ford and Alan Short. The cross section shows the double skin of the building and the process hall. Courtesy of Brian Ford.

an almost constant temperature within the brewing hall.

The Malta brewery uses principles that are still relatively uncommon in buildings. It has a control strategy coupled with natural ventilation that direct the night air, and uses the thermal mass of the building to alter the time response of the building from that of the thermal forcing. These features, which provide significant capabilities for designers, involve complex physics and fluid mechanics.

My interest in building ventilation was first sparked by George Batchelor who asked me to attend a meeting at the Engineering Department in Cambridge as the DAMTP representative on buildings. Subsequently I became intrigued with the gravity current that propagated into my house when I opened the door on a calm cold day. John Simpson, a close neighbour with a house a mirror image of mine, and I wrote our first paper on ventilation on this topic [30]. George had himself worked on a fluid mechanical problem when he was constructing his house 'Cobbers' in Cambridge. He was using double glazing, a relatively new technology in the UK at that time, and he investigated

the convection in a vertical slot with different temperature side walls [3]. George was always interested in answering practical questions. This paper is similarly aimed at answering the questions raised by optimising the ventilation systems in buildings.

Flow through an opening

Natural ventilation is the flow driven by naturally occurring pressure forces: the wind and buoyancy (stack) forces. Consequently, it is necessary to determine the flow through an opening corresponding to a pressure drop Δp across the opening. Idealized flow theory [4] for unidirectional flow through an opening of area A gives the flow rate Q as

$$Q = A\sqrt{\frac{2\Delta p}{\rho}}. \tag{1}$$

This relation assumes laminar uniform flow through the opening and no subsequent contraction after passing the opening. In practice the flow is not uniform as it is altered by the upstream geometry, turbulence occurs due to flow separation and there are contraction effects as the flow accelerates through the opening. The magnitudes of these effects depend on the shape of the opening and its location in the building, the detailed geometry of the opening itself, including the roughness of the surfaces and the nature of the window or other vent structures, and the Reynolds number of the flow.

These complicated effects can not be calculated explicitly and they are included by modifying (1) to include a non-dimensional discharge coefficient C_d, by writing

$$Q = C_d A\sqrt{\frac{2\Delta p}{\rho}}. \tag{2}$$

The value of C_d can be determined by measurement in some circumstances, such as in wind tunnel tests. Some measurements [12] support a value of $C_d = 0.6$, consistent with pipe flow measurements [31], while others [1], [14] argue that the use of a constant value is an oversimplification. [15] show that for flow through a horizontal opening C_d depends on the temperature contrast, since the thermal plume that rises from the opening can lead to additional contraction effects.

Since, generally, it is necessary to make empirical measurements it is convenient to define an effective area A^* by the relation

$$A^* = \sqrt{2}C_d A, \tag{3}$$

so that

$$Q = A^*\sqrt{\frac{\Delta p}{\rho}}. \tag{4}$$

The absorption of the factor of $\sqrt{2}$ is simply for convenience.

If the flow through an opening is not unidirectional, the situation is more complicated and even less well understood or measured. For a vertical opening, buoyancy forces produce a two-way exchange flow, with cool air flowing beneath warm air, and there is usually little mixing between them. This flow is hydraulically controlled, and the flow rates depend on the geometry and temperature difference [10]. If the opening is horizontal then warm and cool air intermingle and mix as they pass through the opening. As the angle of the opening changes from horizontal to vertical there is a transition between these two extremes. [11] showed that the critical angle when the flow behaves as though the opening is vertical is about 20 deg from the horizontal.

In these cases the flow is again characterized by a discharge coefficient. [29] showed that flow driven by a density difference $\Delta\rho$ is given by

$$Q = A^{*5/4}\sqrt{\frac{g\Delta\rho}{\rho}}, \tag{5}$$

where

$$A^* = \left(\frac{C_d}{2}\right)^{4/5} A. \tag{6}$$

The discharge coefficient depends on orientation and takes values of $C_d = 0.25$ and 0.05 for vertical and horizontal openings, respectively.

A building usually has multiple openings and their combined effects can be considered in terms of flow paths. If air flows through n openings in parallel

$$A^*_{\text{total}} = A^*_1 + A^*_2 + \ldots + A^*_n = \sum_{i=1}^{n} A^*_i, \tag{7}$$

while if the openings are in series

$$A^{*\,-2}_{\text{total}} = A^{*\,-2}_1 + A^{*\,-2}_2 + \ldots + A^{*\,-2}_n = \sum_{i=1}^{n} A^{*\,-2}_i. \tag{8}$$

Consider a building in which air enters through openings in the facade and leaves through openings in the roof. The individual sets of openings are in parallel, so that the effective facade openings A^*_{facade} are the sum of the effective areas of all the facade openings, and the same for the roof A^*_{roof}. These two sets of openings are in series so the effective area for the building is given by

$$A^{*\,-2}_{\text{building}} = A^{*\,-2}_{\text{facade}} + A^{*\,-2}_{\text{roof}} \tag{9}$$

If the roof openings total a significantly smaller area than the facade $A^*_{\text{roof}} << A^*_{\text{facade}}$, (9) shows that changes to the total facade area are unimportant in determining the flow through the building. This means that in such a configuration, ventilation control can be imposed centrally (by controlling the roof openings) while allowing individual control in particular spaces by changing the facade openings.

Stack-driven ventilation

The neutral pressure level

The basic mechanism of stack-driven ventilation is simply demonstrated. Consider as shown in figure 3, the pressure distribution inside and outside a space, in which the temperature within the space is higher than outside. If the openings are small enough the flow within the room is sufficiently slow that the interior pressure distribution is hydrostatic. The hydrostatic pressure distribution implies that, compared with outside, there is higher pressure inside the room at the ceiling and lower pressure at the floor, driving the warm air out at the top and cool air in at the floor. This figure shows a two-storey building, in this

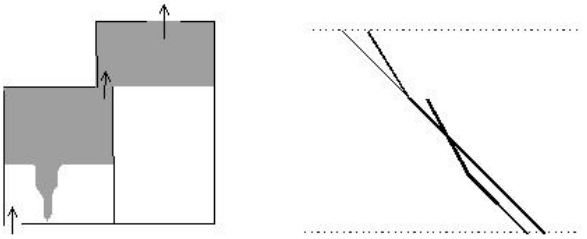

Figure 3: A ventilated two storey building and the associated pressure distribution. The thin line is the ambient distribution and the thicker lines are in the two storeys. There are two levels where the internal and external pressures are equal. From [16].

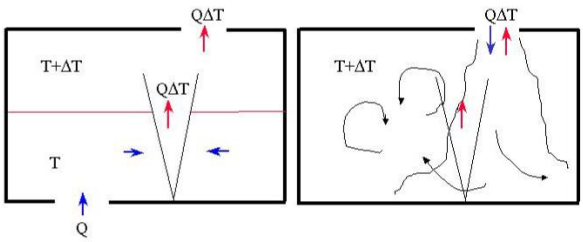

Figure 4: Displacement ventilation and mixing ventilation caused by a single heat source. Note that the difference is caused by the presence or absence of the lower opening.

case a ventilated room attached to an unventilated taller room or atrium. The hydrostatic pressure gradient, which is proportional to the temperature gradient for a perfect gas, within the space is less than the gradient outside. Thus, there is a height, the neutral pressure level, at which the two pressures are the same.

The pressure differences above and below the neutral pressure level drive inflow below this level and outflow above it. The position of the neutral pressure level is therefore of consequence to the location of ventilation openings. It is also of importance to the behaviour of smoke from a fire. Smoke will flow out of openings above the neutral pressure level and the fire will be fed by fresh air from openings below it.

In multi-zone buildings there may be more than one neutral pressure level, as discussed by [16]. In figure 3 there are two neutral pressure levels and their locations depend on the relative heights of the two spaces.

Ventilation patterns

There are two ventilation flow patterns known as mixing and displacement ventilation. Mixing ventilation occurs when there are no density effects or when air of a different temperature is introduced so that it mixes with the air within the space. Displacement ventilation occurs when air of a different temperature is introduced so that a stable stratification is established. This is achieved by introducing cool air at low levels or warm air at high levels. If cool air is introduced at high levels or warm air at low levels, mixing ventilation occurs.

These two ventilation patterns are shown schematically in figure 4. In a steady state an input heat flux into the space produces a ventilation flow that removes the flux. As can be seen from figure 4 the exit temperature is larger with displacement ventilation and so the ventilation flow rate Q is smaller. Since the ventilation requirement for heat removal is much larger than that required for air quality, displacement ventilation provides a more efficient ventilation strategy, especially in summer. This has the advantage that smaller openable areas are required, since the hydrostatic driving head (the vertical integral of the excess temperature) is the same in both cases.

The main limitation of displacement ventilation is that the lower part of the space is at the temperature of the incoming air. There are circumstances, particularly in winter, when this is not acceptable and some tempering of the incoming air by mixing it with air within the space is desirable. Also, in these cases, the net heat flux to be removed is usually lower than in summer, so the inherent inefficiency of mixing ventilation is not such a drawback.

Ventilation in a single space

Stack-driven mixing ventilation

Mixing ventilation occurs when the air entering a space is gravitationally unstable with respect to the air within the space. This is commonly used in air- conditioned systems where cool air is introduced by ceiling vents. In mechanical or natural systems mixing ventilation occurs when cool air enters at high level, and it also occurs when warm air enters at low level. This flow pattern may either be buoyancy driven or driven mechanically or by the wind.

In the case shown in figure 4 cool air enters and falls as a turbulent plume, which tends to mix the air in the space. However, a plume entering an enclosed space leads to a 'filling-box' stratification rather than a well mixed space [2]. In mixing ventilation the incoming air is not a pure plume, but has a significant volume flux (equal to the ventilation flow rate). This volume flux causes a net flow through the space which is responsible for the uniformity of the interior temperature field.

As shown by [5] the flow of cool air into a space through a high vent with small cross-sectional area with finite buoyancy and volume fluxes leads to two effects. The buoyancy flux B (as in the case of a pure plume) leads to the development of stratification within the space. This is essentially the filling box process [2] and occurs on the filling box time scale $\tau_B = \frac{S}{B^{1/3}H^{2/3}}$, where S is the floor area of the space [27]. The finite volume flux Q replaces the air within the room in a time $\tau_Q = \frac{SH}{Q}$. The ratio of these time scales $\frac{\tau_Q}{\tau_B} = \frac{B^{1/3}H^{5/3}}{Q}$ is the ratio of the volume flux in the plume at the floor compared with its initial volume flux. For flow through a square window of height d this ratio can be shown to be $(\frac{H}{d})^{5/3}$. Thus when the window is only a small fraction of the total height of the space the timescales are significantly different and stratification will develop. On the other hand, when the window is a significant fraction of the total height, the replacement time is comparable with the filling box time and the interior will remain at uniform temperature.

It is important to appreciate that the uniformity of the temperature is not a result of turbulent mixing, as has often been stated, including in my papers. Visually there is turbulent mixing (see figure 4) but this is not the primary mechanism. The warm air in the space is constantly replaced by the cool air entering the upper vent and the uniformity of temperature is a result of the large flow rate.

The flow in such a well-mixed space is easy to quantify. A heat source with buoyancy flux B in a space with openings characterised by area A^* and flow by (4) produces a ventilation flow Q and interior buoyancy g' given by

$$Q = A^{*5/6}B^{1/3} \text{ and } g' = A^{*-5/6}B^{2/3}. \qquad (10)$$

Stack-driven displacement ventilation

When the air entering the space is gravitationally stable, such as cool air entering at low level or warm air entering at high level, displacement ventilation occurs. The interior of the space becomes stably stratified and vertical motion within the space is suppressed by the stratification. Horizontal motions are driven by horizontal temperature gradients, but these tend to be small in practice. So, in contrast, to mixing ventilation the interior is relatively quiescent.

The stratification in displacement ventilation depends on the nature of the heat gains. For the case of a single steady heat source (as shown in figure 4 (a)) warm air accumulates near the top of the space forming a stable layer above the incoming cooler ambient air. The ventilation flow Q through both openings is the same, and since air can only cross the stable interface in the convective plume above the heat source, $Q = Q_p$, the volume flux in the plume at the interface. In a steady state the heat entering the upper layer $Q_p \Delta T_p$ in the plume equals that leaving the upper opening $Q\Delta T$. Hence, $\Delta T = \Delta T_p$ and so the upper layer is at a uniform temperature equal to the plume at the height of the interface. If the ventilation flow is increased (by increasing the openings say) the interface will rise (since Q_p increases with height) and so the upper layer will cool. If there are multiple heat sources of different strengths within the space more complex stratification, consisting of multiple layers forms [9], [28].

In the case of purely buoyancy-driven flow, produced by n equal steady heat sources (figure 4 (a)), a two layer stratification is formed. Matching the buoyancy flux and volume flux into the upper layer with that in the plumes at the interface height h gives

$$\frac{A^*}{nC^{3/2}H^2} = \frac{\left(\frac{h}{H}\right)^{5/2}}{\left(1-\frac{h}{H}\right)^{1/2}}, \tag{11}$$

where C is a constant proportional to the entrainment constant for a plume. As is obvious from dimensional considerations the height of the interface is independent of the buoyancy flux of the sources.

Combined effects of stack and wind

Suppose there are two openings, one at the floor and one at the ceiling with effective areas ${A_1}^*$ and ${A_2}^*$, and external pressures p_1 and p_2, respectively. Wind or some other external driving (a fan) could cause the external pressure difference. We assume that the pressure drop across each opening caused by the combination of the stack and wind is large enough so that there is unidirectional flow through each opening.

The flow rate Q is given by

$$p_1 - p_2 - g\int_0^H \rho dz = \rho_0 \frac{Q\,|Q|}{A^{*2}}, \tag{12}$$

where ρ is the density of the air in the space and A^* is the effective area of both openings $(A^{*-2} = A_1^{*-2} + A_2^{*-2})$. The Boussinesq approximation has been made and ρ_0 is a representative density.

For upward flow ($Q > 0$) to occur the left hand side of (12) must be positive, and this occurs, say, when the air in the room is warmer than outside and the wind assists the stack-driven flow ($p_1 > p_2$). Downward flow ($Q < 0$) can be driven by, say, an adverse pressure difference with $p_1 < p_2$. These flows are given by

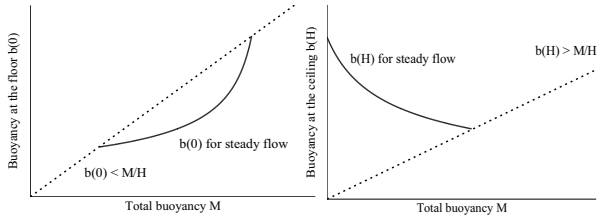

Figure 5: Locus of possible steady states of the stratification for (a) forward flow with $P+M > 0$ and (b) reverse flow with $P+M < 0$. From [17].
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$$Q = A^*\sqrt{P+M},\ Q = -A^*\sqrt{-P-M} \tag{13}$$

where $P = \frac{p_1-p_2}{\rho_0} - gH$ is the excess head (above that in a stationary fluid of density ρ_0), and $M = g\int_0^H \frac{(\rho_0-\rho)}{\rho_0}dz = g\int_0^H \frac{(T-T_0)}{T_0}dz$ is the total buoyancy per unit cross-sectional area in the room.

In a steady state, conservation of buoyancy gives

$$b(H) = b_1 + \frac{B}{A^*\sqrt{P+M}},\ b(0) = b_2 - \frac{B}{A^*\sqrt{-P-M}} \tag{14}$$

for upward and downward flow, respectively, where the buoyancy $b = g(T-T_0)/T_0$ and b_1 and b_2 are the values outside the lower and upper openings, respectively.

Figure 5 shows the locus of possible steady states for (a) upward flow and (b) downward flow. For upward flow, the maximum Q is attained for minimum $b(H)$ with a well-mixed stratification, while the minimum Q occurs when the buoyancy is collected in a thin layer near the ceiling. For downward flow, both maximum and minimum Q correspond to mixing ventilation.

The multiplicity of well-mixed flow states have been studied by [21],[17],[24],[25]. They show that there is a single stable wind-assisted mixing flow, but that multiple solutions exist for the wind-opposed case. In the latter case these are characterised by large flow rate and small buoyancy, which is in the wind-driven direction and is stable, and small flow rate and large buoyancy, which is in the stack-driven direction and is unstable.

Experimental studies of the combined effects of stack and wind have been carried out by [20],[22]. Generally when the wind-driven ventilation is in the same direction as the stack-driven flow, similar ventilation patterns are found to those in the absence of wind. In most cases, and especially in purpose designed displacement systems, stack effects produce stable stratification, and this inhibits vertical turbulent mixing. It is possible to mix this stratification under extreme conditions [18], but in practice this rarely occurs since openings are reduced in high wind conditions. The increased ventilation flow leads to reduced temperatures, either by increasing the height of the ambient zone in displacement ventilation, or more flushing in mixing modes.

When stack and wind effects act in opposition, the situation is more complicated. Figure 6 shows the flow rate in this case as a function of the Froude number $Fr = \frac{UH^{1/3}}{B^{1/3}}$, where U is the wind

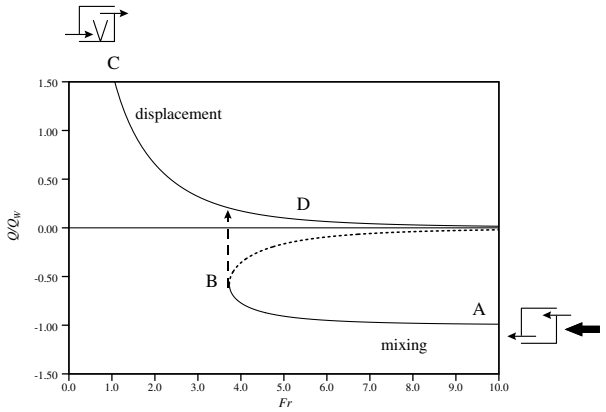

Figure 6: The ventilation flow as a function of Froude number when wind opposes buoyancy. Negative values of Q correspond to mixing ventilation, and positive values to displacement ventilation. From [22].

speed. The ventilation flow rate is non-dimensionalised by the flow rate Q_W driven by the wind, and $Q > 0$ implies ventilation in the stack-driven direction.

Consider the case of purely wind driven flow. Then $Fr \to \infty$ and $Q = -Q_W$ (point A) and the flow is in the mixing mode. As the buoyancy source strength increases, Fr decreases until the point B is reached. At this value of Fr, with further increase B or decrease in U, the flow changes direction and becomes displacement ventilation, which is maintained with further decrease in Fr. Alternatively, if the system is in the displacement mode ($Fr = 0$, point C), and now U increases, this mode may be maintained for values of Fr above the transition from mixing to displacement (point D). Thus the system has hysteresis resulting from the maintenance of stable stratification by the suppression of turbulent mixing.

The presence of interfaces

The earlier argument, which shows that a single source produces a two-layer stratification in displacement ventilation, may be generalised by considering the temperature equation. In the absence of heat sources, and assuming that heat conduction is negligible, the steady-state temperature field $T(\mathbf{x})$ satisfies

$$\mathbf{u} \cdot \nabla T = 0 \tag{15}$$

Horizontal variations in temperature produce buoyancy forces that drive air flows that rapidly reduce horizontal temperature gradients. So, except for isolated regions, such as in a plume, (15) reduces to

$$\overline{w \frac{\partial T}{\partial z}} = 0, \tag{16}$$

where w is the vertical velocity and the overbar represents a horizontal average taken in regions outside strong convective plumes. In regions where there is vertical flow, such as in the bulk of the space in displacement ventilation, (16) implies that the temperature does not vary with height. At a stable temperature interface where the vertical velocity is zero (since the stability precludes mean vertical motion) non-zero temperature gradients do not violate (16).

Thus in displacement ventilation where there is a mean vertical motion through the space, providing there are no internal heat sources, the steady-state stratification must consist of layers of uniform temperature separated by interfaces. On the other hand if there are sources of heat at different heights within the space other types of stratification are possible as discussed below.

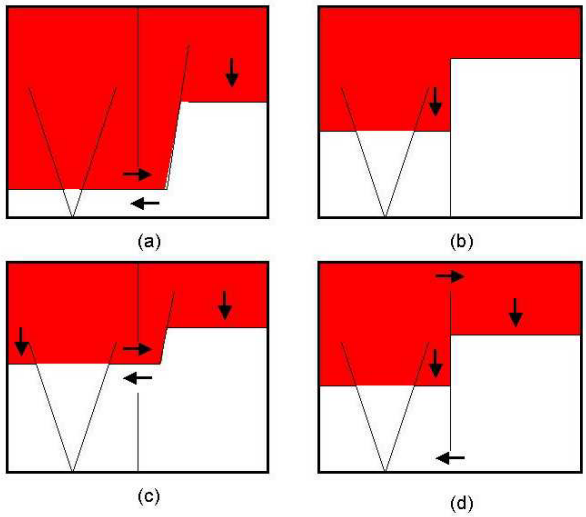

Figure 7: Ventilation in two connected chambers with a heat source in one. Cases (a), (b) and (c) are extensions of the filling box, while (d) is topologically different.

Multiply connected spaces

The discussion so far has been restricted to single spaces. Buildings consist of interconnected spaces, and their interactions raise new issues. Here I will restrict attention to two examples: two connected rooms and a multi-storey building.

Two connected rooms

Possibly this the simplest case, and I focus here on the flow produced by a single heat source in one of the rooms, the other room being unheated. I also restrict attention to the case where the two rooms are not connected to the outside, so that the flow is contained within the two rooms.

With a single opening between the rooms there are three cases of interest. When the opening is at the bottom (figure 7 (a)) the heat accumulates in the heated room and establishes a filling box stratification. When the stratification reaches the top of the opening, warm air flows into the unheated room, which then develops a filling-box stratification also. Cool air flows out of the bottom of the unheated room to feed the plume [33]. When the opening is at the top, warm air flows across the whole ceiling initially establishing a filling-box stratification in the upper parts of both rooms. When the descending front reaches the top of the opening (figure 7 (b)) the cool air in the unheated room is trapped below the interface, and that room takes no further part in the flow. A filling box develops in the heated room, with a continual exchange flow through the opening continuing to warm the top of the unheated room. If the opening is at mid-height (figure 7 (c)), the flow is a combination of figure 7 (a) and (b). When the warm air in the heated room reaches the top of the opening it flows into the unheated room as in figure 7 (a) and produces stratification at the top. An exchange flow continues through the opening until the interface descends to the bottom of the opening and traps the cool air there as in (figure 7 (b)).

The other possibility is to have two openings, at the top and

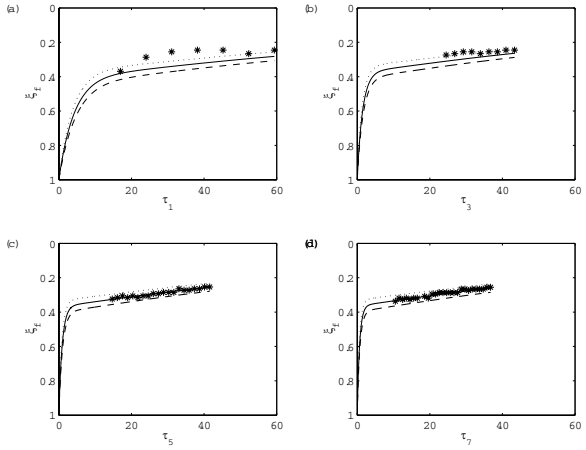

Figure 8: The interface height in the heated chamber. The time is non- dimensionalized by the filling-box time of the larger unheated chamber. From [26].

bottom (figure 7 (d)). The flow is topologically different in this case, with unidirectional flow through the openings. Warm air leaves the top of the heated room and cool air enters through the bottom opening. Displacement ventilation is established in both rooms: in the heated room it is in the conventional sense of upward flow, but in the unheated room the flow is downward from the ceiling. [26] have examined the effects of different sized rooms. When the rooms are comparable in size the two rooms behave similarly to a single space and produce a filling box stratification. When the unheated room is much larger, such as in the case of an office connected to an atrium, the flow within the heated room initially behaves as though it is connected to an infinite environment. Two-layer displacement ventilation is established, and the interface rapidly develops at some height. As warm air flows into the unheated room, the changing pressure distribution means that the exchange flow is reduced and the interface descends to compensate. This time evolution of the interface position is determined by the size of the unheated room. Figure 8 shows plots of the interface position against time for four aspect ratios. Note that the slow time evolution scales on the filling-box time for the larger unheated room.

Multi-storey buildings

Stack-driven ventilation in a multi-storey building has been studied by [19], [16]. Multi-storey buildings can have multiple neutral pressure levels (figure 3), and these can cause complex flow paths through the building. For an atrium building, heat storage in the atrium can be used to drive ventilation flows through other parts of the building. [16] show that enhanced ventilation depends critically on the ventilation openings of the atrium, and that poor design can lead to reduced ventilation. Multi-story buildings also provide opportunity to bring fresh air down from intakes at high levels on the facade. This has the advantage of avoiding pollution at street levels, and increases the potential for natural ventilation in urban areas. [19] show that the use of such intakes is feasible and provide guidance on design criteria based on friction losses.

Heat transfer from surfaces

Use of the thermal mass of a building is an important part of the armory of a designer using natural ventilation. The walls, floor and ceiling around a space can store or release heat, and their temperatures determine the radiant exchanges and the comfort temperature. In warm climates the thermal mass can be used to provide cooling during the day if it is exposed to cool air during the night. This strategy, called night-cooling, and others such as running chilled water through pipes in floors and ceilings is becoming a part of natural ventilation design, particularly in commercial buildings. The amount of cooling achieved depends on the mass of the structure that is cooled. This is determined by the air flow past the structure during the cooling phase, and the thermal properties of the structure itself.

Chilled ceilings or heated floors

Distributed cooling from a ceiling or heating of a floor provides a source of convective motion. The Rayleigh numbers Ra are typically very large. For a 3m high room, $Ra \sim 10^6$, at which the convection is turbulent, are achieved with temperature differences between the surface and the air of just 10^{-4} K. Thus turbulent convection will occur whenever there is any floor heating or ceiling cooling.

A buoyancy flux B distributed over the surface leads to well-mixed interior conditions in both mixing and displacement ventilation. This is obvious for mixing ventilation and can been seen in the limit of a large number of sources (representing the distributed heating) in (11). In this case the flow rate and temperature are given by [13]

$$Q = A^{*2/3} H^{1/3} B^{1/3}, g' = \frac{B}{Q}. \tag{17}$$

This leads to the question of the form of the stratification in displacement ventilation when there are both distributed and local sources of heating. This question relevant to the breakdown of the stratification when a ceiling is chilled, or when solar gains are distributed over the floor.

[32] examined the stratification formed within a closed space due to a point source of buoyancy with both a heated floor and a cooled ceiling. In the former case they found that the interior stratified when the localised buoyancy flux B_L exceeded the distributed buoyancy flux B_D. An interface formed at a height h given by

$$\frac{h}{H} = \frac{1}{1+R} \tag{18}$$

where $R = \frac{B_L}{B_D}$. The region above the interface was fed by the plume which entrained fluid from the convecting lower layer. This upward volume flux was balanced by downward entrainment across the interface driven by the penetrative convection. The interface stays at a constant height but the temperature of both regions increase with time as the space heats up.

This study has been extended to a ventilated chamber with openings at the top and bottom by [23]. Similar behaviour to the unventilated case was found, except that now a steady state is formed in which both the form and magnitude of the stratification are constant in time. The form of the stratification again depended on R, with a two layer stratification occurring for $R > 0.15$. The interface height is a fairly weak function of R, but depends on the size of the openings. The reason for this critical flux ratio appears to be related to the efficiency of entrainment into the convective zone. Typically the entrained flux is about 0.1 - 0.2 of B_D for penetrative convection, consistent with the maximum value that can balance the plume and destroy the stratification.

Heated sidewalls

The introduction of heat at different heights in a room allows for the development of more complex thermal stratification. This

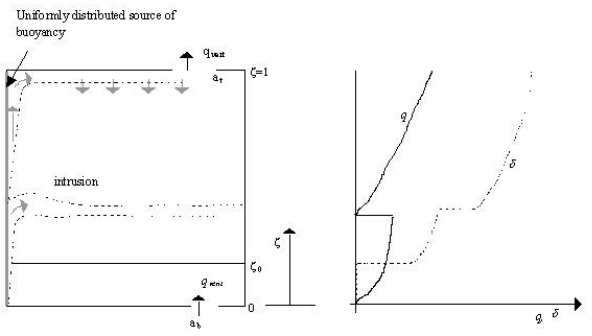

Figure 9: Schematic of a plume rising from a distributed source of buoyancy on one wall of a ventilated box. The plume rises until it is neutrally buoyant and then forms a horizontal intrusion. From [8].

heating may be fairly uniform, caused for example by night cooling of a wall with large thermal mass, or localised, such as resulting from a sun patch. The Grashof numbers of these flows are typically around 10^{12}, and the flow is turbulent. The case of a vertically distributed source was discussed by [29], who argued that the stratification will develop a series of layers separated by interfaces. This has recently been extended by [8] who modelled a heated sidewall as shown in figure 9. See also [6]. In this case plume theory can be applied and it is found both numerically and experimentally that distinct layers form when the vent areas take certain sizes.

The reason for the layered structure is that the volume flux through the space must be constant with height. Since the wall plume increases in volume flux with height by entrainment, it is necessary for detrainment to occur when the flux carried by the plume equals that through the openings. This detrainment is depicted in figure 9.

Conclusions

This paper summarises recent research on the fluid mechanics of natural ventilation. I hope that I have demonstrated that there are many fascinating fluid dynamics problems associated with a space connected to a large environment and subject to pressure forces across openings, and that the results are relevant to building ventilation, both natural and mechanically driven. The aim of this research is to gain fundamental understanding that can be used to underpin design strategies to provide energy efficient ventilation and good air quality in a range of climates.

There are many aspects of this subject that I have been unable to discuss in this brief review. I have ignored important time-dependent processes that occur with changing external (e.g. wind, solar radiation) and internal (e.g. changing heat loads, variable openings) conditions. Many of these are only now being addressed. There is relatively little work on the effects of non-adiabatic surfaces in combination with internal gains. Similarly, I have not discussed other cooling strategies such as evaporative cooling, which has significant potential in dry hot climates. I trust though that this paper has persuaded the reader that there are plenty of interesting problems to work on.

I would like to (almost) end this paper with a quote from GKB. In the Ascher Shapiro lecture entitled *Hooked on Research* he gave at MIT in 1995, George was bemoaning the quality of the writing of scientific papers that he dealt with as editor of the Journal of Fluid Mechanics. Having had some of my papers subjected to his scrutiny it is with some trepidation that I conclude this one. As George says in the lecture 'The contents of a paper are of value only to the extent that they can be understood by others'. However, I am consoled by Proust (also quoted by GKB) who writes in *Remembrance of Things Past*: 'Clear ideas, for each of us, are those which lie at the same level of confusion as our own.'

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