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CONVECTIVE TIME-SCALES AND OTHER PARAMETERS GOVERNING AIR MOVEMENT WITHIN BUILDING SPACES

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Introduction

Warm-air heating systems, probably incorporating a ventilation capability, have many potential advantages for the 'low energy' house (1). These benefits include a good energy efficiency (70-75% over the heating season), and ease of control with rapid response to load changes. In traditional warm-air systems the operation of the circulating fan is controlled by an on-off thermostat, and therefore the warm-air is injected into the room intermittently (2,3). This intermittent air injection can cause discomfort due to the re-establishment of cold window downdraughts and infiltration draughts during the 'off' period of the control cycle (4). An alternative modulating control system may therefore be utilised, in which a variable-speed fan is operated continuously, except at low load when cyclic operation again occurs. Pimbert (2) has argued, on the basis of laboratory tests using a 'comfy-test' meter, that modulating control results in an improved thermal environment at the most frequently experienced loads. Room airflow under these conditions are governed by a balance between the 'forced' warm-air and the negatively-buoyant window downdraught. Thus, a 'mixed convection' regime prevails within the space,so that the motion will depend on the heating system/room configuration and on the relative strengths of the buoyancy-driven and forced air streams. The present contribution evaluates the time-scales of the convection processes that dominate the motion in warm-air heated rooms. These parameters may be used to gain a better physical understanding of the resulting airflow patterns.

The time-dependent character of the operation of warm-air heating systems has received little attention in the published literature. However, a study of cyclic or intermittent air injection into rooms was undertaken by Sidaway, Hammond & Probert (3,5) at Cranfield in the early 1980's. Experiments were carried out in a full-scale environmental test room in which the air supply could be terminated rapidly and recycled directly to the fan, providing the required intermittent injection. This facility was used to identify the physical characteristics in intermittent air jets in both their 'starting' and 'stopping' modes, and to obtain data which may be used to predict the development of these jets. Smoke flow visualisation studies in small-scale models were also undertaken in order to examine the qualitative behaviour of the complete flow field. The early work on the injection of an isothermal plane 'wall-jet' along the ceiling of the test room was described by Sidaway et al (3). Further studies, in which more complex jet configurations and thermal conditions were investigated, are reported in Sidaway's thesis (5).

Recently the author and his co-workers (1,6,7) have been engaged in a study aimed at obtaining improved convective heat transfer data for building energy simulation programs. A common weakness in modern approaches to building thermal modelling is that emphasis is placed on simulating the transient performance of the building fabric, while air flow and convective heat exchange in and around the structure are modelled using only rough approximations. In order to alleviate this situation, a hierarchy of interacting and interdependent calculation methods have been developed at Cranfield, (1,6-8). These were initially directed towards computing internal convective heat exchange specifically for the case of warm-air heated or mechanically-ventilated building spaces, including domestic rooms or commercial offices. The calculation methods ranged from 'lower-level' or 'short-cut' approaches, such an analytical solutions and data correlations, to the development of a 'high-level' or 'field' flow model that solves the governing 'elliptic' equations for the complex, jet-induced room airflow; the ESCEAT computer code of Alamdari, Hammond and Mohammed (7). Both the higher- and lower-level methods were then used to develop and verify an 'intermediate-level' or 'zonal' computer code known as the ROOM-CHT program (1). The philosophy behind this approach is set out in the review of Alamdari, Hammond & Melo (6), where they argue that intermediate-level models appear to offer the best prospect for meeting the heat transfer data requirements of building energy simulation programs. These include the need to obtain a balance between accuracy, economy and user-friendliness.

Not withstanding the above comments regarding the usefulness of intermediate-level models of surface-averaged heat transfer rates, Alamdari, Hammond & Mohammad (7) argued that high-level flow models are a necessary requirement for the accurate prediction of, for example, the occupation zone thermal comfort conditions. This is because the flow and thermal fields within ventilated enclosures are complex, and cannot be computed using the sort of simplifying assumptions that form the basis of zonal models. In the present work, the results of the case study of warm-air heating by Alamdari et al (1,7) will be used to illustrate the convective time-scales that dominate airflow patterns in ventilated spaces. Emphasis will be placed on the modern practice of controlling the warm-air supply via a modulating control system. Alamdari et al (7) computed the airflow patterns for this case using the ESCEAT computer code. These results will therefore be employed to illustrate the discussion of the various convective time-scales. The influence of the flow development time in the case of the more traditional intermittent warm-air systems will also be discussed, but only to a limited extent.

Simulated Enclosure

In order to evaluate the convective time-scales prevailing in warm-air heated rooms, the enclosure illustrated in Figure 1 will be considered. The airflow pattern in this room has been simulated using a high-level or field computer code by Alamdari et al (7). It represents a corner, ground-floor domestic living room having dimensions 4.30m length, 2.45m width and 2.45m height. Modern practice in the UK would normally utilise warm-air injected through a 'low side-wall register' (9), with supply conditions regulated by a modulating control system (2).

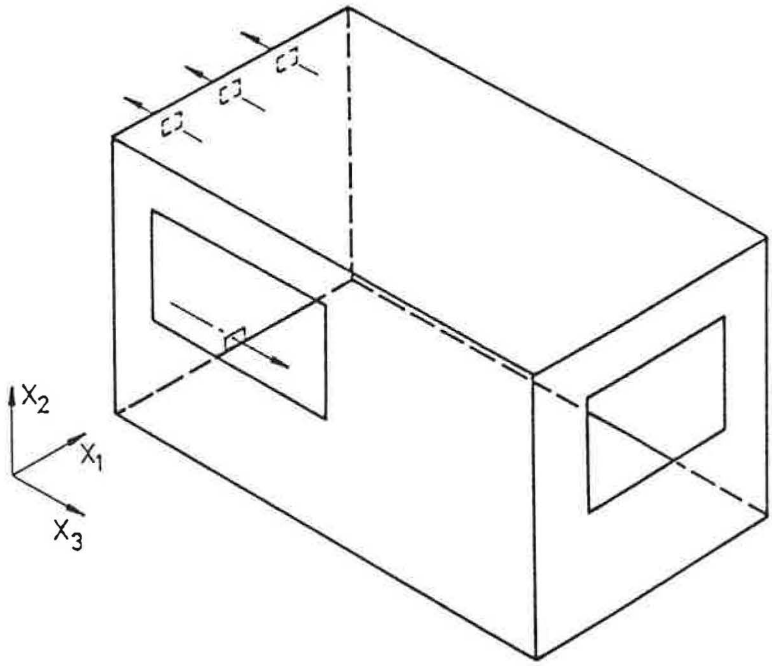

Fig.1 Schematic diagram of warm-air heated room with 'low side-wall register' (1)

A notional (reference) occupation zone air temperature of 23 ± 0.5°C was adopted for the simulated enclosure, while the surface temperature of the internal walls and ceiling were similarly assumed to remain constant at 21°C over the heating season. The two external walls, incorporating single-glazed windows (1.45m x 1.00m in the far-wall and 1.80m x 1.00m in the side-wall), and the floor were given inside surface temperatures estimated on the basis of best current British practice U-values (10). These temperatures are given in Table 1 as a function of three representative heat loads. The full load condition coresponds to a supply warm-air ventilation rate of about 3½ air-changes per hour (ACH).

Table 1. Demand-dependent supply air conditions (modulating control) and surface temperatures

DEMAND	HEAT LOAD	OUTSIDE AIR TEMPERATURE (°C)	INTERNAL SURFACE TEMPERATURES (°C)		SUPPLY AIR CONDITIONS	
			Exterior Walls and Floor	Windows	Velocity (m/s)	Temperature (°C)
High	Full	-1	16	6	1.50	65
Intermediate	65%	+7	18	11	1.21	55
Low	30%	+15	20	16	0.93	39

The size of the warm-air supply register was determined by the requirements for full load operation, with face velocities and temperatures within the limits recommended in the British design manual for gas fired systems (9). This suggested a rectangular grille (200mm x 120mm, 70% free area) located at the bottom centre of the interior wall, as shown in Figure 1. Such an arrangement gives rise to the formation of a three-dimensional 'wall-jet' (11), which initially spreads out from the terminal device across the floor. Three room air extract grilles are positioned at high-level above the supply register, and these are also illustrated in Figure 1. The supply air conditions would be regulated by the modulating control system, which adjusts a variable-speed fan. This normally operates continuously to give a constant face velocity, whose value depends on the heat load. In contrast, the supply air temperature modulates very slightly about its load-dependent mean value, although this was neglected for the purposes of the present computations. The supply conditions that correspond to the three representative heat loads are again given in Table 1, where the temperatures are those suggested by modern practice (1).

Intermittent warm-air heating systems under normal operating conditions would employ control cycles like those given in Table 2. The corresponding supply air velocity and temperature would be held constant at all heat loads with typical values of $1.95ms^{-1}$ and $65^{\circ}C$ respectively (1) for a similar configuration to that above.

Table 2. Intermittent heating control cycles.

DEMAND	CONTROL CYCLES
High	20 min ON/6 min OFF (76.9% ON)
Intermediate	6 min ON/6 min OFF (50.0% ON)
Low	6 min ON/20 min OFF (23.1% ON)

Convective Time-scales

Mean Motion in 'Steady' Flow

The air movement within a warm-air heated room is a jet-induced turbulent flow which typically gives rise to a recirculation pattern (in the fluid dynamic sense). Heated air from the supply aperture or inlet grille initially forms a thin shear layer, known as a three-dimensional wall-jet (11), that spreads out from this low side-wall register and over the floor in the present configuration. When the warm-air system operates under modulating control, the turbulent airflow pattern generated by the forced convective heating becomes statistically stationary (or 'steady') in time. The mean, or time-averaged, motion will have a time-scale (12-14) defined by:

$$\tau_f = L/u \tag{1}$$

where L and u are characteristic length and velocity scales respectively. If L is some measure of the streamwise distance, then this time-scale is what Reynolds terms the 'flow-past' time (13). It represents the time required for a fluid element convected with the mean flow to travel the length of the shear layer; the wall-jet in this case. The streamwise characteristic dimension must scale on the size of the space-conditioned enclosure, and it is therefore convenient to choose the room height. In order to estimate the magnitude of τ_f, the maximum value of the supply velocity (see Table 1) may be used. Together these indicate, via equation (1), a forced convective time-scale of about 1.5 s, which shows the potentially fast rate of diffusion or mixing within the space. In contrast, diffusion on a molecular scale is very much slower, as can be illustrated by the 'viscous diffusion' time-scale (12,14):

$$\tau_v = L^2/\nu \tag{2}$$

where ν is the kinematic viscosity. If the latter is evaluated at the supply temperature, then this gives a value of about 3½ days! These forced convection and viscous diffusion timescales may be combined to yield an enclosure Reynolds number:

$$Re = \tau_v/\tau_f \simeq 2 \times 10^5 \tag{3}$$

This parameter is classically interpreted as the ratio of the inertia to viscous forces, and the high value again illustrates the dominance of the former in mechanically-ventilated spaces.

In a warm-air heated room with exterior windows there are two sources of buoyancy. These are the buoyant supply jet and the negatively-buoyant cold window downdraught. The characteristic velocity for a buoyancy- driven flow may be obtained from dimensional analysis (15):

$$u = (g\beta\Delta TL)^{\frac{1}{2}} \tag{4}$$

where g is the gravitational acceleration ($\sim 9.81ms^{-2}$), β is the coefficient of cubic expansion ($= T^{-1}$ for gases), and ΔT is the temperature difference across the thermal shear layer. Substituting for this velocity in equation (1) provides an analogous buoyant, or 'heat' (14), convection time-scale:

$$\tau_b = (L/g\beta\Delta T)^{\frac{1}{2}} \tag{5}$$

If the room height is again used for the length scale, then the time-scales for the buoyant jet and the cold downdraught are both about 2 s. Combining the buoyant convection time with its viscous diffusion counterpart, equation (2) leads to the Grashof number:

$$Gr = (\tau_v/\tau_b)^2 \tag{6}$$

which has a value of slightly less than 7×10^9. According to Alamdari & Hammond (16), this implies that the buoyancy-driven boundary layers over the cold windows are in the transitional flow regime. It also indicates, by analogy with the arguments used in connection with the Reynolds number, that buoyancy forces dominate viscous ones.

The very long time-scale for viscous diffusion, compared with either the forced or buoyant convection times, suggests that the balance between the latter will determine the nature of the airflow pattern in a warm-air heated room. Thus, it is the ratio of the convection time-scales that is important in this particular mixed convection situation, and they give rise to the so-called Archimedes number (17):

$$Ar = (\tau_f/\tau_b)^2 \tag{7}$$

This parameter is about 1.3 for the conditions studied here, indicating approximate <u>overall</u> parity between buoyancy and inertia forces. However, in reality it is the local balance that will determine flow behaviour in the room. The heated jet rapidly mixes and spreads within the enclosure and its strength will therefore have significantly weakened before, example, it reaches the cold window located in the far wall.

Mean Motion in 'Unsteady' Flow

The parameters identified above are also important when the warm-air heating system is under intermittent or cyclic control. However, in this situation further time-scales are of significance. In particular, Reynolds (13) has argued that, because the response of shear layers to change relative to an external stream is not instantaneous, the 'development time' (τ_d) is important in statistically non-stationary (or 'unsteady') flows. This is the time taken for a shear layer to reach equilibrium with impulsively imposed changes relative to external conditons. Reynolds suggests that $\tau_d \simeq 10\tau_f$, or typically about ten flow-past times. Even more significant in the present context is that development times can be extremely long in separated or recirculating flows. The data of Smith & Kline (18) suggests that τ_d = 100-250 τ_f for stalled diffuser flows, equivalent to 3-6 minutes in the present case. This is remarkably close to the time of 4 minutes which Sidaway (5) found was required to achieve steady-state conditions after impulsive supply air injection into a room.

Turbulent Motion

The structure of turbulent flows is made up of a range of length scales or eddy sizes. These range from the large-scale motions, which are of the order of the shear layer width and take their energy from the mean motion, to the smallest eddies that finally dissipate this turbulence energy in the process of microscopic-scale mixing. In shear layers where the large eddies are convected, as in wall-jets, Reynolds (13) has suggested that the characteristic time-scale of these structures is the so-called 'large-eddy turnover time':

$$\tau_e = \delta/\Delta u \tag{8}$$

where δ is the shear layer width and Δu is the maximum velocity differences across the layer. A typical value for the turnover time may be obtained from estimates of the length and velocity scales that are likely to prevail at the end of the wall-jet 'potential core' region (11). This yields, under the conditions that prevail in the warm-air heated room, a time of about 0.2 s. In contrast, the most widely used time-scale for the smallest eddies is that due to Kolmogorov (12-14):

$$\tau_k = (\nu/\varepsilon)^{\frac{1}{2}} \tag{9}$$

where ε is the rate of turbulence kinetic energy dissipated per unit mass. Substituting again values applicable at the end of the wall-jet potential core indicates that $\tau_k \simeq 0.0025$ s. The large-eddy turnover time gives a rough idea of the rate at which the large structures mix the turbulent flow macroscopically, before microscopic-scale viscous diffusion can take place on the Kolmogorov time-scale (13). Both times are seen to be very short compared to the corresponding time-scales governing the mean motion.

High-level or 'Field' Airflow Model

Background

Air flow and convective heat transfer within an enclosure are governed by the principles of conservation of mass, momentum and thermal energy (or enthalpy). These 'conservation laws' may each be expressed in terms of 'elliptic' partial differential equations, the solution of which provides the basis for a high-level flow model. A discretized form of the governing equations may be obtained by dividing the flow domain into a finite set of small sub-domains, each surrounding a node of the computational grid. The discretized equations are then formulated in such a way that integral conservation requirements are satisfied for individual sub-domains or control volumes. This approach has been called the 'control-volume method' by Patankar (19), and the discretized equations might preferably be distinguished by the prefix 'finite-domain' or 'finite-volume', rather than the term 'finite-difference' commonly employed. The description 'finite-domain equations', suggested by Spalding (7), is adopted here. They are solved in the present higher-level mathematical model by methods similar to those used in the TEACH and CHAMPION family of finite-domain programs developed by Gosman and Pun (20) and Pun and Spalding (21) respectively. Both these codes employ the SIMPLE ('semi-implicit method for pressure linked equations') algorithm of Patankar and Spalding (22), but are restricted to two-dimensional geometries. The authors and their co-workers have therefore used their past experience in applying the CHAMPION code to mechanical ventilation problems (6) in order to develop a more general computer program capable of simulating three-dimensional flow fields (Alamdari et al (7)). This is called the ESCEAT (Elliptic Equation Solver for Convection and Heat Transfer) code, and was originally employed to compute convective heat transfer in developing, square duct-flow (23). In addition to its ability to handle complex geometries, the program incorporates a number of improvements (described below) in the numerical solution procedure for the finite-domain equations.

The computation of airflow patterns within the present warm-air heated room by Alamdari et al (7) was one of the first uses of a high-level flow model to obtain surface convective heat exchange. In addition, none of the earlier studies made realistic provision for cold window effects. The influence of window downdraught on room airflow is usually very significant, even with forced convective heating or cooling systems.

Mathematical Framework

The governing time-averaged, elliptic equations for the turbulent flow and thermal field may be written in a common form, using tensor notation (7):

$$\frac{\partial}{\partial x_j}(\rho u_j \phi) = \frac{\partial}{\partial x_j}\left(\Gamma_\phi \frac{\partial \phi}{\partial x_j}\right) + S_\phi \qquad (10)$$

CONVECTION DIFFUSION SOURCES OR SINKS

Here u_j (u_1, u_2, u_3) are the time-averaged (mean) velocity components in the coordinate directions X_j (X_1, X_2, X_3 : see Figure 2), ϕ are any of the dependent variables [u_j, H ($\equiv$Cp T), k or ε], Γ_ϕ are the effective (laminar plus turbulent) diffusion coefficients for these ϕ's, S_ϕ are the sources or sinks for each ϕ, and ρ is the fluid density. Closure of the equation set was achieved in the present study by using an extended version of the energy-dissipation turbulence model (24) in order to compute an isotropic 'eddy' viscosity, or turbulent exchange coefficient for momentum (μ_t). This requires the simultaneous solution of two additional transport equations for the turbulence kinetic energy (k) and its dissipation rate (ε). The extension to the standard k-ε model (24) involved the inclusion of buoyancy generation or source terms (G_B) in these equations, using a modified form to that suggested by Rodi (25). Mathematical expressions for the diffusion coefficient and source terms for each variable are given in Table 3 and its accompanying notes. The thermal energy equation was modelled using the effective Prandtl number, σ_{eff}, approach (24), and the fluid properties for air were assigned values corresponding to those at the reference temperature.

Table 3. Diffusion coefficients and source terms in the governing elliptic equations.

CONSERVED PROPERTY	ϕ	Γ_ϕ	S_ϕ
Mass (continuity)	1	0	0
Direction - i momentum	u_i	μ_{eff}	$-\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu_{eff}\frac{\partial u_j}{\partial x_i}\right) - \rho g_i \beta \theta$
Thermal energy (enthalpy)	H	$\frac{\mu_{eff}}{\sigma_{eff}}$	0
Turbulence Kinetic energy	k	$\frac{\mu_{eff}}{\sigma_k}$	$G_K + G_B - \rho\varepsilon$
Turbulence energy dissipation	ε	$\frac{\mu_{eff}}{\sigma_\varepsilon}$	$\frac{\varepsilon}{k}\left[C_1(G_K + G_B) - C_2\rho\varepsilon\right]$

Notes

1. $\mu_{eff} \equiv \mu + \mu_t = \mu + \frac{C_\mu \rho k^2}{\varepsilon}$; $\sigma_{eff} \equiv \mu_{eff} \Big/ \left(\frac{\mu}{\sigma} + \frac{\mu_t}{\sigma_t}\right)$

2. $G_K = \mu_t \frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)$; $G_B = g_i \beta \frac{\mu_t}{\sigma_t}\frac{\partial \theta}{\partial x_i}$

3. $\theta \equiv T - T_R$, where T_R is the reference temperature

4. Values for the turbulence model 'constants':

   $C_\mu = 0.09$, $C_1 = 1.43$, $C_2 = 1.92$, $\sigma_t = 0.85$, $\sigma_k = 1.00$ and $\sigma_\varepsilon = 1.30$

In order to bridge the steep dependent variable gradients close to the room surface, the ESCEAT code employs so-called 'wall-functions' (24). These are simply based on the well-known bilogarithmic behaviour of the mean velocity and temperature near solid walls, using the log-law constants reported by Alamdari et al (7). Some modification was needed to the usual near-wall treatment of k in order to ensure that the heat balance on the warm-air heated room resulted in realistic temperatures in the occupation zone. This gave rise to occupation zone temperatures of 22.5, 23.0 and 23.5°C for the low, intermediate and high heat loads respectively. Full details of the special near-wall treatment are reported by Alamdari et al (7,23).

Numerical Solution Procedure

The generalised set of differential equations, equation (10), may be formally integrated over each cell volume of the computational grid to yield corresponding finite-domain equations. In the ESCEAT code a 'power-law' differencing scheme is utilised because it provides improved accuracy compared to some of the older approaches, such as the upwind or hybrid schemes (7,23). The velocity components are calculated at staggered locations mid-way between adjacent grid nodes (22). This practice has the advantage of ensuring that the velocities are directly available for calculating the convective fluxes of the scalar variables, as well as lying between the location of the static pressures that drive them. However, it necessitates minor changes to the coefficient expressions, as outlined in Mohammad's thesis (23).

The set of algebraic finite-domain equations are solved in the ESCEAT code in an iterative, 'line-by-line' manner (19, 21), using a tri-diagonal matrix algorithm (21,22). Here the velocities and pressures are calculated via the simplec algorithm, recently proposed by Van Doormaal and Raithby (26). This is a variant of the SIMPLE algorithm (22) which is more consistent, and consequently induces a faster rate of convergence. The latter is also enhanced by the adoption of a plane-by-plane 'block-correction' procedure applied by sweeping the flow domain in the x_3-direction (see Figure 1). These block adjustments were based on the requirements for overall mass and momentum conservation. The computational grid employed for the present simulation of the warm-air heated room utilised a 17x17x15, non-uniform nodal network, whose fineness can be judged by the velocity vector diagrams presented in Figure 2. In the previous two-dimensional high-level flow model computations by the authors (6) a jet inlet cell was utilised, over which the variation of the dependent variables was prescribed. This reduced the need for a finely-spaced grid near the supply register. However, it is difficult to prescribe the near-register flow and thermal field in a strongly buoyant situation, such as the present one. Consequently, a jet inlet cell has not been employed in the current study, for which about 700 iterations were required to obtain a converged solution. The latter was assumed to have been obtained when the finite-domain equations were satisfied to within 0.5% or less of the inlet mass flow, or the heat supplied in the case of the H-equation. Further details of the ESCEAT code, including the measures taken to ensure grid-independent solutions in the present case, are reported elsewhere (7, 23).

Flow Field Computations and Time-scales

Computations

Velocity vector diagrams illustrating the flow pattern within the warm-air heated room under full heat load conditions are shown in Figure 2. These plots were obtained using the ESCEAT code (7), and display vectors in

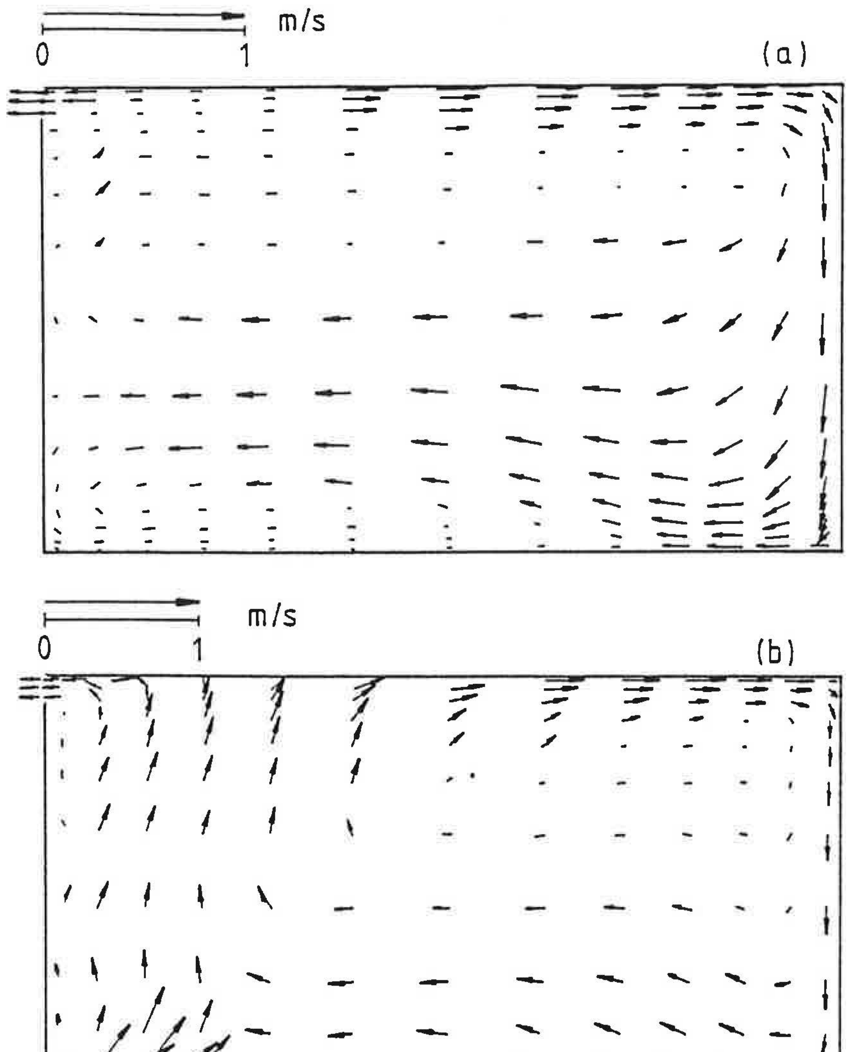

Fig. 2. Flow field velocity vectors computed using the ESCEAT code (7): (a) x_1/W = 0.25, (b) x_1/W = 0.50, (c) x_1/W = 0.,75. [Note scale change].

the x_2-x_3 plane at three positions which coincide with those of the air extract grilles. Here the tail of each velocity vector indicates the location of a node in the finite-domain computational grid. It is therefore evident that these nodes were concentrated close to the room surfaces, as well as to the jet inlet. This arrangement was adopted in order to provide more nodes in those regions where there are steep dependent variable gradients. The computed flow pattern can be seen to be strongly influenced by buoyancy effects, due to the high temperature of the supply air and the counteracting cold downdraught induced by the windows. In particular, the downdraught from the window in the right wall (viewed from the supply register) clearly damps the rigorous buoyant flow in the rest of the enclosure. The pattern in the latter region is characterised by two recirculating flow regions: one dominated by the buoyant supply jet and the other by the cold downdraught from the window in the far-wall.

Time-scales Under Modulating Control

In the course of the discussion regarding the form and physical significance of the various time-scales, it was noted that one forced convection and two buoyant convection time-scales applied to the mean motion in the present configuration. The latter arise as a result of the buoyancy of the heated supply jet and the negatively-buoyant cold downdraught from the windows. Under conditions typical of UK practice, the maximum value of all three time-scales were comparable (~ 2s). This is reflected in the Archimedes number, based on room height, of about 1.3. The numerical values of these parameters will, of course, depend on the choice of length scale. Room height was adopted in the present study, because it coincides with the gravitational vector. It is also a reasonable choice in regard to the forced convection or 'flow past' time, whose associated characteristic length must scale on the overall dimensions of the enclosure. Selecting alternative length and velocity scales for the mechanical-ventilation flow (for example, the aperture dimensions or a notional average room velocity respectively) can lead to variations in the flow past time of 0.3 - 300 s. This would give rise to a corresponding range of Archimedes numbers, defined as a time-scale ratio, of $0.06 - 1.2 \times 10^4$ with the same room conditions. It is evident that the usefulness of these parameters is restricted to comparative purposes only.

The relative influence of buoyancy and inertia forces on the airflow pattern depends on their local balance. The supply jet rapidly disperses as it spreads across the floor, and its local forced convection time consequently rises. This leads to the dominance of buoyancy effects, reflected in a rise in the local value of Ar. The hot supply jet undergoes a change to a thermal plume-like flow within a short distance from the register, as is illustrated by the velocity vector diagram in Figure 2 (b). This enables the cold downdraught from the windows to govern the airflow pattern in the rest of the room.

Time-scales Under Cyclic Control

Intermittent warm-air heating may be divided into three regimes: 'steady' mixed convection during the ON cycle similar to that which prevails with modulating control; 'steady' buoyancy-driven convection induced by window downdraught during the OFF cycle; and 'unsteady' mixed convection in the intervening periods. The mean motion convection time-scales discussed in the preceeding section will also characterise these steady-state conditions. However, in the transient situation, around the beginning and end of each cycle, the flow development time is important. This was estimated to be about 3-6 minutes for the jet-induced recirculating flow in the room. Under intermediate to low heating demand (see Table 2), this corresponds to a large proportion of the heating system ON cycle time. The airflow pattern in this period would consequently be time-dependent, requiring a more complex solution procedure than the steady-state high-level flow model employed by Alamdari et al (7). Nevertheless, the bulk of the flow field is likely to have been established in much less than the development time, and the need for a transient procedure may be unnecessary.

Concluding Remarks

The time-scales of the convection processes that dominate the motion in warm-air heated rooms have been evaluated, along with related dimensionless parameters. These may be used to gain a better physical understanding of the resulting airflow patterns. Emphasis has been placed on the modern practice of controlling the warm-air supply via a modulating control system. This operates a variable-speed fan that normally operates continuously to give a constant face velocity, whose value depends on the heat load. In contrast, the supply air temperature modulates very slightly about its load-dependent mean value. Alamdari et al (7) have computed the airflow patterns for the case of a corner, ground-floor domestic living room, where the warm-air is injected through a low side-wall register. These computations were obtained with a steady-state high-level or field flow model (the ESCEAT computer code), and includes the realistic simulation of buoyancy effects such as cold window downdraught. The computed velocity vector diagrams have been employed to illustrate the discussion of the various convective time-scales appropriate to modulating control systems. In addition, the significance of the flow development time in the case of the more traditional intermittent or cyclic warm-air systems has been discussed, although only to a limited extent.

Typical values for the time-scales in this case were found to vary from 10^{-3} seconds for the small-scale turbulent eddies to 3½ days for viscous diffusion. However, flow past times for both forced and buoyancy-driven mean motion in the warm-air heated room was found to be about two seconds. However, the magnitude of the latter times, and related parameters such as the Archimedes number, depend on the choice of length scale. The usefulness of these parameters is limited to comparative purposes only. In the present study, the room height was adopted as a suitable overall length scale, because it coincides with the gravitational vector. Nevertheless, it has been argued that the relative influence of buoyancy and inertia forces on the airflow pattern on their local balance. This is masked when overall velocity and length scales are employed.

The flow develoment time for the jet-induced recirculating room flow under intermittent operating conditions was estimated to be about 4 minutes. This is comparable to the heating system ON cycle time with intermediate to low demand, and might suggest the need for time-dependent computations. However, the bulk of the flow field is likely to have been established in much less than this time. It has therefore tentatively been argued that the need for a transient solution procedure, which would be costly in terms of computer resources, may be unnecessary.

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