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Summary A programme of work involving the measurement of ventilation rates, air velocities and temperatures has been completed within a naturally ventilated auditorium in the Queens Building, De Montfort University. Measurements have been recorded for 'winter', 'mid-season' and 'summer' conditions, and average occupancy levels. Measurements carried out in 'winter' and 'mid-season' indicate that the predominant ventilation driving force is the stack effect. However, current control settings of fresh air inlets allow too great a free area to be produced, which causes high room air velocities with associated low temperatures, resulting in high discomfort levels for occupants seated in line with external openings. Results for 'summer' external conditions on particular dates indicate that ventilation rates are driven largely by wind-induced effects, which can be much higher than buoyancy-driven flows. Internal ceiling and wall surface temperatures show little variations with time, owing to the heavyweight nature of the building structure and its exposed areas.

Analysis of parameters affecting the internal environment of a naturally ventilated auditorium

E M Clancy† BE MS MSc PhD CEng MIMechE and **A T Howarth**‡ BSc, MPhil, PhD, CEng, MIMechE, MCIBSE
†School of The Built Environment, Coventry University, Coventry CV1 5FB, UK
‡School of The Built Environment, University of Nottingham, Nottingham NG7 2RD

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List of symbols

A_E	Effective opening area (m^2)
C_D	Discharge coefficient
g	Acceleration due to gravity (m s^{-2})
h_1	Height of inlets above datum (m)
h_2	Heights of outlets above datum (m)
k_{os}	Stack ventilation constant
Q_B	Stack ventilation flow rate (m^3 s^{-1})
ppm	Parts per million
ρ_a	Internal air density (kg m^{-3})
T_i	Average internal air temperature (°C)
T_o	Average external air temperature (°C)
T_{avg}	Average of internal and external temperatures (K)

1 Introduction

The Queens Building at De Montfort University, Leicester, is a three-storey building housing the Department of Mechanical and Manufacturing Engineering and contains lecture theatres, classrooms, laboratories and offices. Almost all of the building is naturally ventilated; this is facilitated by the use of large vertical ducts or stacks that use the greater buoyancy of the indoor air to 'pull' outside air through openings in the facades into the building. Warmer, internally circulated air is then discharged via the stack exits. Extensive use is made of daylighting using skylights.

The building contains two lecture theatres that have self-contained natural ventilation systems. In the theatre under investigation, outside air enters adjacent to a lightly used road into three plenums via modulated control dampers (see Figure 1). It disperses within voids underneath the seating and enters the room via vertical grilles positioned at ankle height. Air leaves the space via two large openings near the base of two stacks. Outlets are positioned at the top of each stack in the form of eight windows. Intake and outlet positions can be varied by the operation of motors controlled by the building management system (BMS). A 'ceiling' type fan is installed in one of the stacks to provide enough incoming air to satisfy fresh air requirements when the flow rates produced by natural ventilation are inadequate. Heating in winter is provided by finned tubes placed within the seating void adjacent to the vertical grilles[1, 2].

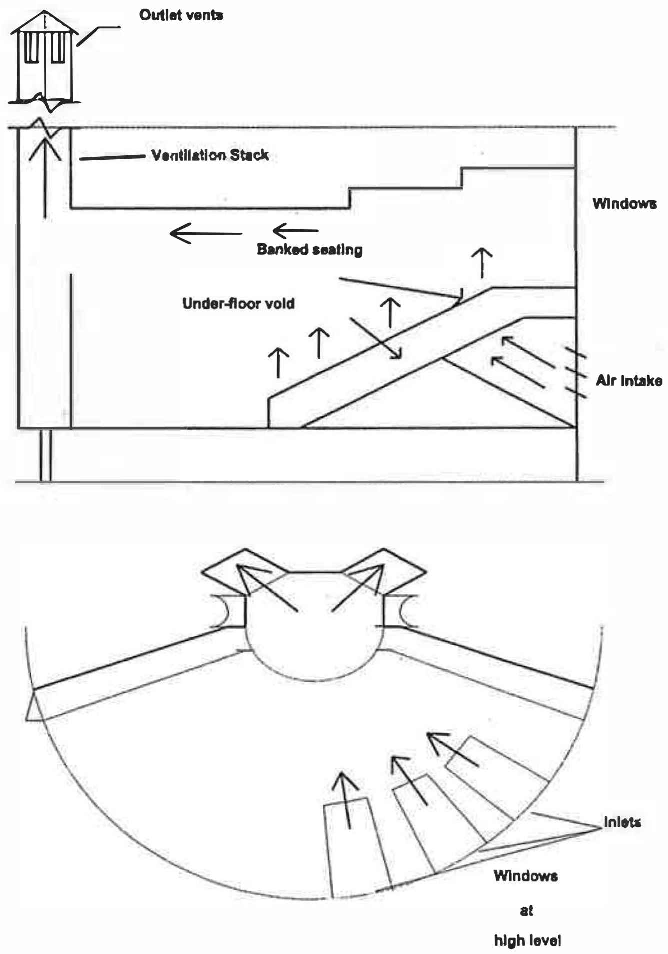

Figure 1 Section and plan of theatre 1.10 showing the general direction of air movement in the space

1.1 Aims and objectives of experimental work

The aims of the particular work discussed here were to:

- Develop an empirical model of the ventilation system used.
- Assess the ventilation performance of a naturally ventilated auditorium using appropriate experimental techniques.
- Assess thermal comfort conditions within the enclosure.

Additional work included reporting on the ventilation assessment of the auditorium within the framework and requirements of IEA Annex 26 on Energy Efficient Ventilation of Large Enclosures[3].

2 Measurements performed

2.1 Short time scale experiments

The aims of short time scale experiments (i.e. carried out over periods from 10 min to 3 h) were to measure bulk ventilation rates, local air change rates and thermal comfort levels. Bulk ventilation and local air change rates were recorded using the tracer gas decay technique and the direct measurement of air speeds in the two stacks. Flow directions in the stacks were detected using paper strips suspended across the stack inlet openings and using smoke generators. Air and surface temperatures were measured using thermistor probes connected to automatic logging devices. Air speeds and velocities were detected and recorded using low-velocity omni-directional hot-wire type sensors and ultrasonic instrumentation.

The measurement of stack air speeds involved the placing of air speed sensors in four positions in each stack; these were mounted at heights of 9.5 m above floor level. To record room air speeds, the sensors were also positioned at seven locations within the theatre, one on either side in the first row and two on the fifth row (LHS) with three in the corresponding row on the RHS (see Figure 2). Sensors were placed at ankle height 0.5 m from inlet grilles. External, wall and ceiling slab temperatures were monitored with thermistor sensors connected to data loggers.

2.2 Long time scale measurements

Data on temperatures and other important parameters were obtained over relatively long periods (6 days) to monitor how the building was performing over periods of several days. This information was used to assess the building heating system performance in winter and the effectiveness of thermal mass and night ventilation in reducing large variations in internal temperature in summer.

A number of BMS sensors were located in the auditorium under test. These monitored air temperatures at four positions, a ceiling surface temperature and in-stack air temperatures. A CO_2 sensor was installed in one of the stacks to monitor the indoor air quality and to provide a control signal for the operation of the ventilation openings. Vent opening positions were also detected; these would indicate the relative opening positions of three fresh air dampers and eight exhaust openings/windows.

External conditions, namely air temperature and relative humidity, were also recorded. The building operators were interested to know how local climatic conditions would vary on a seasonal and an annual basis. A record of external humidity would inform the operators of its local variability and the frequency with which values would be less than 40% and greater than 70% during occupied hours. Wind speed and direction sensors had been positioned 3 m above one of the concourse stacks, but the readings obtained from these devices were found to be unreliable. Probably owing to local high turbulence values, there was little or no correlation between the local measured wind speeds and those monitored at other sites within a 15-mile radius of Leicester.

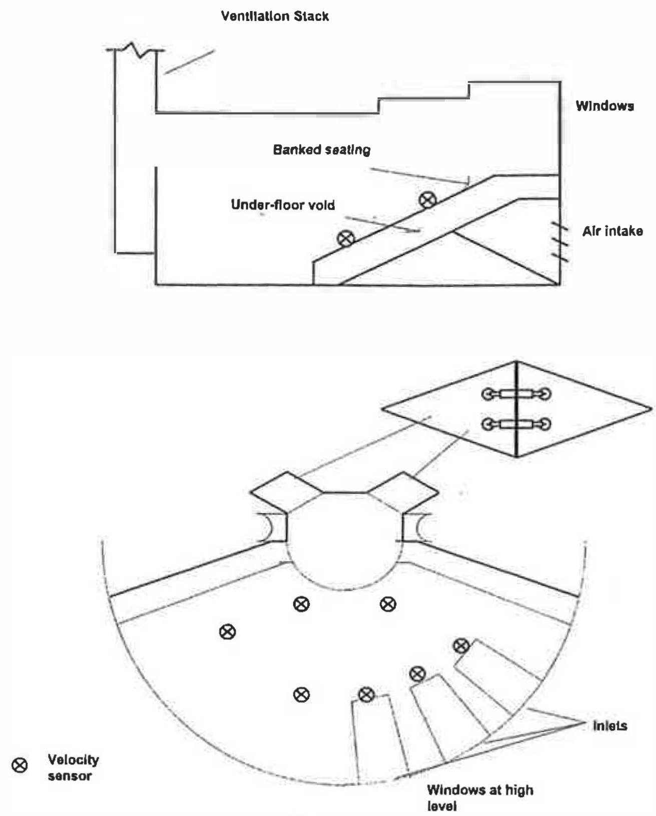

Figure 2 Section and plan of theatre 1.10 showing the positions of velocity sensors and sensor mounting orientations in a stack

However, accurate values of wind speed and direction were provided by a local airport and via a weather station at Loughborough University and these were used to examine the correlation between corrected wind speeds and ventilation rates in the auditorium.

3 Results

3.1 'Winter' and 'mid-season' ventilation rates

A formula can be derived that allows stack-generated flows within the auditorium to be calculated (see reference 4; table A4.4); i.e.

$$q_v = C_d A_E \left[2g(h_2 - h_1) \left(\frac{T_i - T_o}{\frac{T_i + T_o}{2} + 273} \right) \right]^{0.5} \tag{1}$$

where q_v is in $m^3\,s^{-1}$ and other terms are defined in the List of symbols.

It is interesting to compare the above equation with empirically derived formulae that can be generated from the experimental data. Figures 3, 4 and 5 show best-fit lines through measured data. It can be seen that when the vents are at their

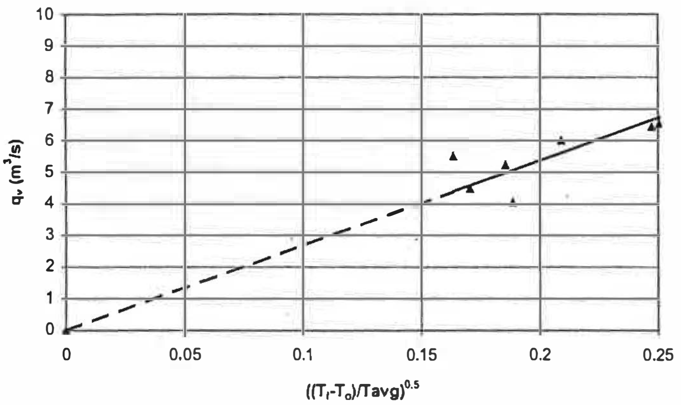

Figure 3 Measured flow rate versus $[(T_i - T_o)/T_{avg}]^{0.5}$: effective area is 68% of maximum area. $y = 26.878$; $R^2 = 0.5218$

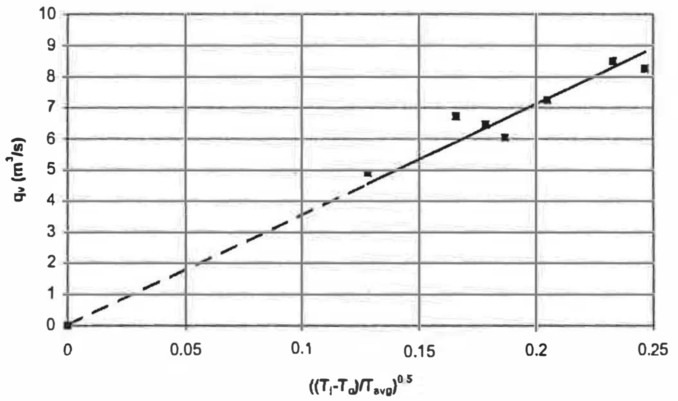

Figure 4 Measured flow rate versus $[(T_i - T_o)/T_{avg}]^{0.5}$: effective area is 89% of maximum area. $y = 35.619$; $R^2 = 0.842$

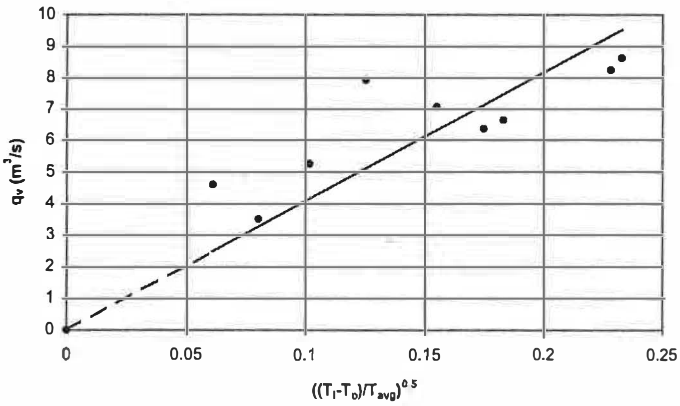

Figure 5 Measured flow rate versus $[(T_i - T_o)/T_{avg}]^{0.5}$: effective area is 100% of maximum area. $y = 40.872$; $R^2 = 0.284$

50% and 75% positions, reasonably good correlations are produced between volumetric flow rate and a function of average internal and external air temperatures only, i.e. the following relationship is valid[5]:

$$q_v = k_{os}\left(\frac{T_i - T_o}{T_{avg}}\right)^{0.5} \tag{2}$$

where k_{os} = stack ventilation constant at the particular os (opening setting) value (50%, 75% and 100%) (i.e. $k_{50} = 26.88$ m³ s⁻¹; $k_{75} = 35.62$ m³ s⁻¹; $k_{100} = 40.87$ m³ s⁻¹).

Thus the constant term, k_{os}, can be compared directly with the following, i.e.

$$k_{os} = C_d A_E[2g(h_2 - h_1)]^{0.5} \tag{3}$$

Substituting actual values for h_2 and h_1 (let $h_2 = 18.5$ m; $h_1 = 4.8$ m) gives

$$k_{os} = 16.39 C_d A_E \tag{4}$$

The openings are arranged in series (it is assumed that flow is uni-directional from the fresh air inlets to the discharge points at the top of the stacks). Hence three free areas are derived as shown in Table 1. The formula used is

$$\frac{1}{A_E^2} = \frac{1}{A_i^2} + \frac{1}{A_{ig}^2} + \frac{1}{A_{si}^2} + \frac{1}{A_{so}^2} \tag{5}$$

where A_E = total effective area (m²); A_i = fresh air inlet areas (m²); A_{ig} = room inlet grill area (m²); A_{si} = stack inlet areas (m²); and A_{so} = stack outlet areas (m²).

Using the above values of effective areas, equation (4) can be written as

$$k_{50} = 16.39C_d(2.43) = 39.83C_d \ (\mathrm{m^3\,s^{-1}})$$
(BMS setting of 50%) (6)

$$k_{75} = 16.39C_d(3.18) = 52.12C_d \ (\mathrm{m^3\,s^{-1}})$$
(BMS setting of 75%) (7)

$$k_{100} = 16.39C_d(3.57) = 58.51C_d \ (\mathrm{m^3\,s^{-1}})$$
(BMS setting of 100%) (8)

If the constants k_{50}, k_{75} and k_{100} are divided by the corresponding experimentally found values (see Figures 3, 4 and 5), discharge coefficient (C_d) values can be determined. These are 0.67, 0.68 and 0.70, respectively. This implies that the flow paths are more streamlined than might initially be assumed (i.e. with each opening acting as a sharp-edged orifice with a discharge coefficient of 0.6). The relatively good correlations between the CIBSE stack formula (equation 1) and measured ventilation rates indicate the formula can be used to accurately predict flow rates when buoyancy-driven flow predominates. This occurred (for this particular case) when the inside–outside air temperature difference was at least 10°C and when wind speeds were less than ~3 m s⁻¹).

3.2 *'Summer' ventilation rates*

Table 2 shows flow rates measured in 'summer' (i.e. when the outside temperature was within about 5°C of the inside temperature). This generally indicates much lower values than those found in 'winter' and 'mid-season', but flow rates were rarely below that required for fresh air provision for the fully occupied theatre (150 × 8 litres per second or 1.2 m³ s⁻¹). Still air conditions were never recorded, indicating that there was almost always enough wind-induced pressure difference and/or buoyancy effect to provide enough incoming air.

Table 1 Opening areas and total effective area for different opening settings

BMS setting (%)	Area of inlets (m²)	Inlet grilles (m²)	Stack openings (m²)	Outlets (stack top) (m²)	Total effective area (m²)
50	3.30	14.69	6.11	4.64	2.43
75	5.29	14.69	6.11	5.62	3.18
100	5.86	14.69	6.11	7.49	3.57

Table 2 'Summer' ventilation rates

Average temperature (°C)		Wind speed ($m\ s^{-1}$)	Wind direction	Flow rate (meas.) ($m^3\ s^{-1}$)
Internal	External			
24.6	24.9	2.8	ENE	1.6
25.0	25.9	2.8	ENE	2.3
27.5	29.8	3.4	ENE	1.2
22.8	21.7	2.6	ENE	4.6
25.3	26.2	2.2	E	1.9
26.0	26.9	2.3	E	1.6
27.8	29.0	3.6	E	1.7
27.6	29.7	3.6	E	1.8

A series of experimental runs were carried out to examine the effects of varying outlet configurations on flow rates. Four different outlet configurations were used; see Figure 6. Higher flow rates were measured when the leeward-facing outlets were open with windward-facing outlets closed (see Figure 7). In the example shown, higher stack velocities were generated when settings corresponding to S1 and S3 had leeward vents open with windward vents closed. However, under normal operation, the outlet vents used were not designed to enhance wind-induced flow rates.

It is likely that higher ventilation rates would have been recorded if internal air routes had been made more streamlined; that is, pressure losses through the stack inlets could have been reduced by the use of connecting metal ducts between the openings and the stacks, thus reducing the overall pressure drops through the ventilation system. Higher flow rates could then be generated for specific buoyancy- and wind-induced pressure differences between the internal occupied zone and external openings.

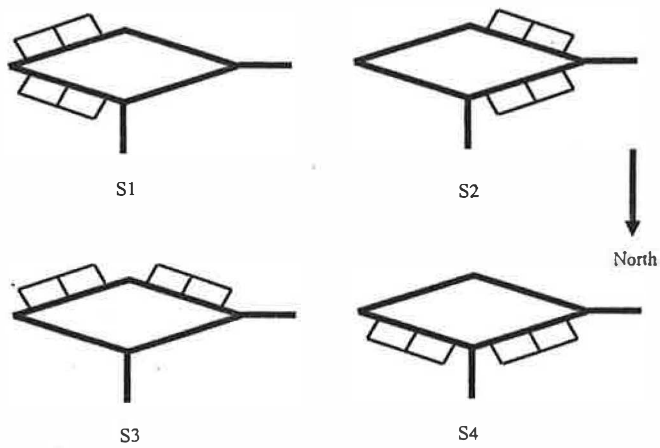

Figure 6 Positions of outlets at tops of stacks. S1: SE/NE windows open, SW/NW windows closed. S2: SE/NE windows closed, SW/NW windows open. S3: SE/SW windows open, NE/NW windows closed. S4: SE/SW windows closed, NE/NW windows open

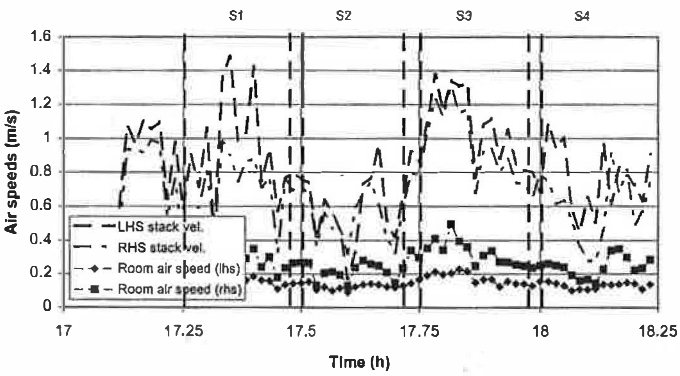

Figure 7 Air velocities in both stacks (LHS stack vel., RHS stack vel.) and room air speeds for different outlet configurations; wind direction W to WNW; inside-outside temperature difference ~0°C

3.3 *Ventilation efficiency*

Ventilation efficiency is a measure of how well a particular ventilation system distributes air within an enclosure from a separate source and how effectively it deals with the reduction of pollutant concentrations within the space. It can be determined from the local mean and room mean ages of air (which allows air change efficiencies to be found) and contaminant removal effectiveness indices. The local mean age of air at a point 'A' in an enclosure can be defined as the average time it takes air, once entering the space, to reach point 'A'. The room mean age is then the average age of air in the space. The air change efficiency is then the percentage ratio between the nominal time constant and the air change time, where the nominal time constant is the minimum time in which air, once entering the space, will remain. The contaminant removal effectiveness indices deal with the movement and dilution of pollutants within an enclosure[6].

The source of air can be a ventilation supply plant or air-handling unit, in which outside air is filtered, heated and/or cooled and humidified. In the case of the Queens Building auditorium, air is brought in directly from outside into an underfloor plenum. Air leaving the plenum is heated via finned tubes located behind supply grilles.

As can been seen from Figure 1, air is introduced at ankle height on each seating level and is extracted via stack inlets with their lower edges 3.8 m above floor level. The system design would suggest that local air change rates should be high when the vents are open and when there are sufficient pressure differentials between the room pressure and either the outside pressure at low level or that existing at high level adjacent to the outlets.

However, much higher air speeds (Figure 8) and local air change rates (Figures 9, 10 and 11) were recorded on the right-hand side of the auditorium than on the left-hand side. This would imply that the underfloor void was not acting as a true plenum and that outside air was simply short-circuiting this area and entering into occupied areas without much drop in velocity and with little transfer of heat from the heating tubes behind specific inlet grill areas.

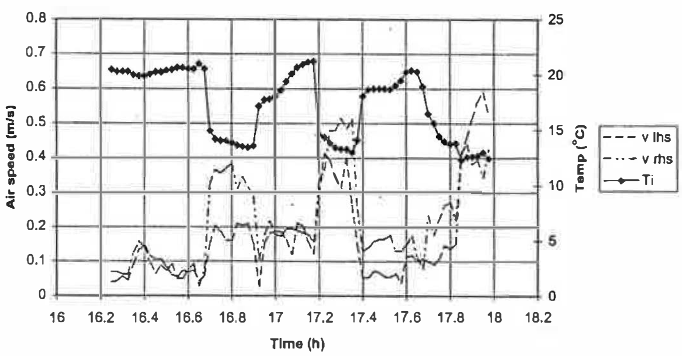

Figure 8 Internal temperature and air speeds (mid-season external conditions). v_{LHS} = room air speeds on left-hand side of auditorium; v_{RHS} = room air speeds on right-hand side of auditorium; T_i = internal air temperature

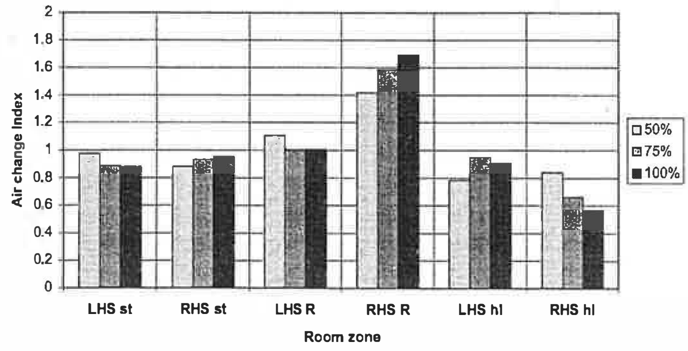

Figure 9 Average air change rate in winter for three different opening settings. LHS, left-hand side stack; RHS, right-hand side stack; st, stack; R, room; hl, high level

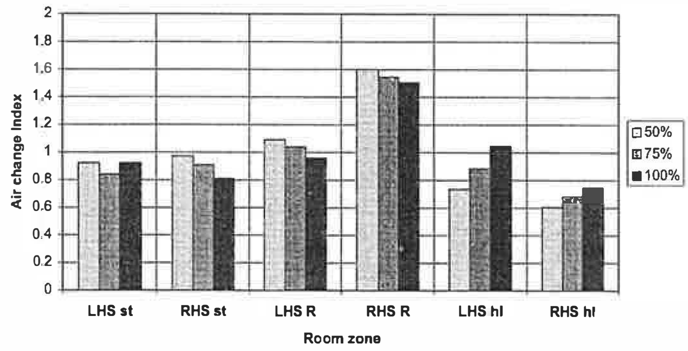

Figure 10 Average air change rate in mid-season for three different opening settings

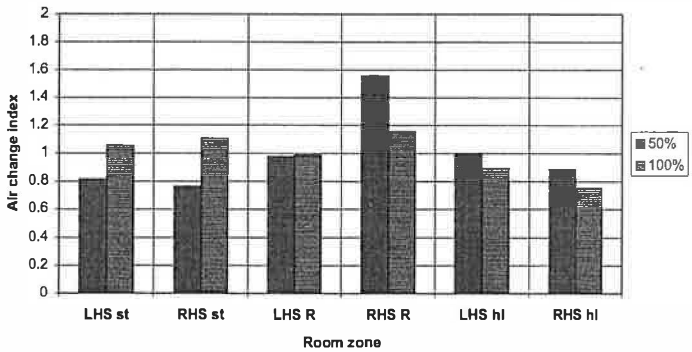

Figure 11 Average air change rate in summer for three different opening settings

For determination of ventilation efficiency, the local air change index is found. This is calculated using the local mean age of air and the nominal time constant[7]. The local mean age of air at point p is found from

$$\bar{\tau}_\mathrm{p} = \frac{\int_0^\infty C_\mathrm{p}(t)\,\mathrm{d}t}{C(0)} \tag{9}$$

where $C_\mathrm{p}(t)$ is the tracer gas concentration at a point p at time t.

In practice, the measurement period is finite, so concentrations are initiated when $C_\mathrm{p}(0)$ is at its maximum value (e.g. 50 ppm) until a concentration of 5 ppm or less is reached. Thus the nominator of equation (9) is calculated using data compiled during the measurement period plus a correction value (the summation from the end of the experimental period to a time equal to 'infinity').

The nominal time constant, τ_n, is found from the slope of the log of the average tracer gas concentration versus time. The local air change index is then found from

$$\varepsilon_\mathrm{p} = \frac{\tau_\mathrm{n}}{\bar{\tau}_\mathrm{p}} \tag{10}$$

The accuracy of obtaining local mean ages using the tracer gas decay method depended upon the following[8]:

(*a*) *Achievement of adequate mixing of the injected tracer gas*. Desk-mounted mixing fans were used at each injection point and on other areas within occupied zones. Mixing of the gas with the room air was continuously monitored at each sample point before a run was carried out; the vents were opened once local concentrations reached 50 ppm ±5 %.

(*b*) *The sampling interval used*. Maximum values for calculated air change rates were greater than 20; thus very rapid decay rates were produced. To enable detection of these rapid decays in concentrations, a very small sampling interval was required. Software that controlled the sampling hardware allowed two periods to be chosen, i.e. 2 and 6 min. The shorter period was initially chosen, but this could be halved by connecting every sample point to two channels (12 channels in total were available).

(*c*) *The length of sampling tubes used*. Ideally, each tube should have the same length so that the time taken for samples obtained at each point to reach the detector is the same. In practice, the average sample tube length was about 30 m and little deviation in length occurred. Tubes were purged once samples were obtained.

(*d*) *Positions of sampling points*. In an ideal situation, large numbers of sample points are used for large enclosures as a number of different zones are likely to be present and mixing rates will be different in each zone. Each zone will have particular heat gains, air and surface temperatures and ventilation conditions. In this particular case, it was assumed that there were two 'occupied' zones, a high-level zone, an underfloor plenum zone and two stack zones. Of primary interest were room and stack zones, and at least two intake points were positioned in each of these zones. Hence reasonably adequate coverage of gas concentrations in the entire auditorium volume was obtained. Difficulties were experienced in positioning and securing sampling points near the ceiling and within the stacks.

Measurements indicated that high air change rates were occurring and that the ventilation system was effective in distributing air within the enclosure (see Figures 9, 10 and 11). However, high local air change rates were found within the right-hand side of the space (high air speeds were also measured in the same area; see above).

3.4 *Results; measurement of internal conditions over long periods*

As stated above, data were recorded at 15 min intervals with a cycle period of one week. Data have been saved for sample winter, mid-season and summer external conditions. Internal and external temperatures, heating valve and vent positions are shown in Figures 12 and 13 (winter and mid-season conditions), while temperatures and opening positions are indicated in Figure 13 (summer conditions).

The high thermal mass of the space is indicated in Figure 12 (i.e. the decay in internal and surface temperatures is very

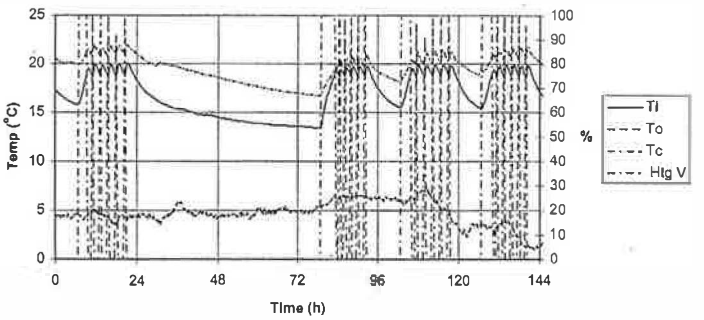

Figure 12 Internal and external temperatures and heating valve positions, 15 to 20 December 1995. T_i = internal air temperature; T_o = external air temperature; T_c = ceiling surface temperature; Htg V = relative position of heating valve

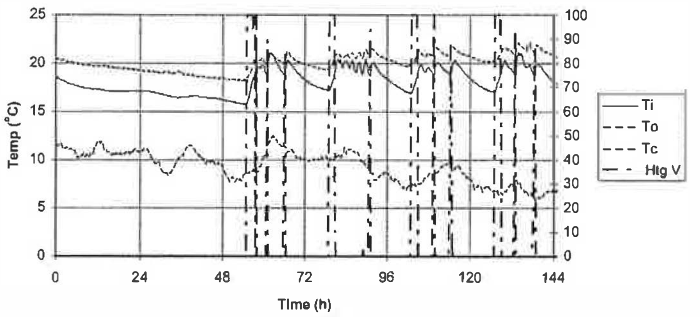

Figure 13 Internal and external temperatures and heating valve positions, 13–19 January 1996. Top curve, ceiling temperatures: middle curve, room air temperature; bottom curve, external temperatures

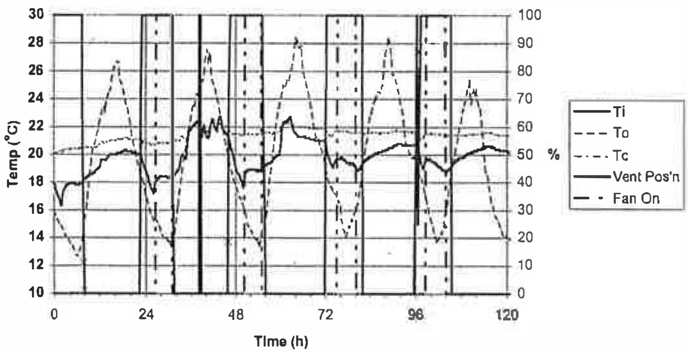

Figure 14 Internal and external temperatures and vent opening positions, 3–7 May 1995 (curve showing largest variations in temperatures; external temperatures, bold curve; room air temperatures, curve with small temperature variations; ceiling temperatures)

gradual outside pre-heat and occupied periods, e.g. from 24 h to 72 h, despite a low outside temperature of ~5°C).

Figure 14 demonstrates the effectiveness of thermal mass and night cooling in reducing internal temperatures even when external temperatures are much higher than internal temperatures. A one-degree fabric (ceiling) temperature rise over a five-day period is indicated in Figure 14 despite maximum external temperatures of 26°C and significant occupancy heat gains between 24 h and 72 h. This can be compared to a two-degree temperature rise (Figure 15) over the same time duration for similar external conditions and no occupancy heat gains when night cooling did not take place[9]. The variation in internal temperatures is only 3°C for outside temperature variations of up to 16°C (no internal heat gains; Figure 15). When internal heat gains are present with night cooling, the internal temperature variation is about 5°C with a 13 to 14 degree fluctuation in outside temperature (Figure 14).

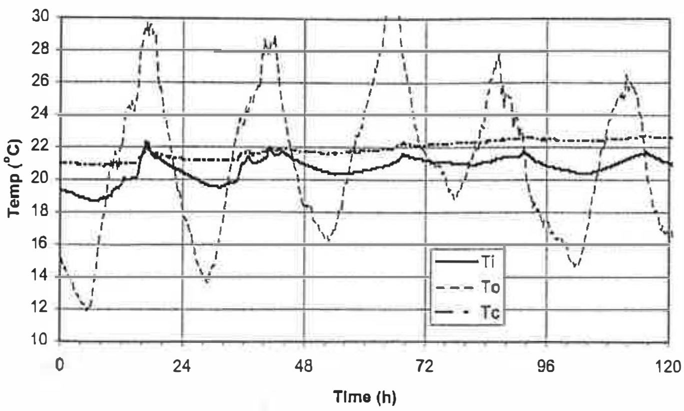

Figure 15 Internal and external temperatures and vent opening positions, 3–7 May 1995 (curve showing largest variations in temperatures; external temperatures, bold curve; room air temperatures, curve with small temperature variations; ceiling temperatures)

Further depression of the room air and surface temperatures would have occurred if the stack fan had not been switched on during the night cooling periods (note the rise in internal temperature when the fan is operated in Figure 14). The fan operation possibly causes destratification of the air within the space and the stacks and additional heat transfer from the exposed room surfaces to the air. The fan is started if the average internal temperature is 3°C above the room set point and greater than the external temperature plus two degrees. It appears that a set point of 12°C is used during unoccupied periods; hence the fan will run if the above conditions are satisfied.

4 Conclusions

Adequate ventilation is provided within the auditorium by natural ventilation throughout a 'typical' year. In winter and mid-season, stack- and wind-induced pressure differences are more than sufficient to provide for fresh air requirements when the room is fully occupied, but the current control settings of the fresh air dampers will lead to severe discomfort in winter for occupants sitting in rows 3 and 4 on the right-hand-side of the space directly in-line with external openings.

The ventilation stacks were sized for buoyancy-driven flow only (such that enough fresh air would be provided even for a temperature difference of 3°C between inside and outside). However, substantial upflows were generated when external temperatures were equal to or greater than internal temperatures (i.e. wind-induced pressures were frequently large enough to produce adequate flow rates and to overcome negative buoyancy effects).

Higher flow rates were measured when the leeward-facing outlets were open with windward-facing outlets closed. However, the particular outlet vents used were not designed to enhance wind-induced flow rates.

Ventilation flow rates in 'winter' and 'mid-season' for this enclosure can be predicted using equations derived assuming buoyancy-driven flow only. It is possible to predict 'summer' flow rates provided accurate values are obtained for local wind speeds, directions and corresponding opening surface pressure coefficients.

Significantly higher air speeds and air change rates were measured on the right-hand side of the auditorium (see Figure 8). This suggests that the underfloor void was not acting as an effective plenum and free areas between the void and the room would need to be reduced.

Internal temperatures can be predicted assuming the space behaves predominantly like a first-order system, i.e. 'heating' and 'cooling' time constants can be found from the experimental data and used in the solution equation for the system[10]. The thermal mass of the space reduces the effects of external temperature swings on internal temperatures and night ventilation reduces the rise in fabric temperatures when external temperatures are high (i.e. above 26°C).

Acknowledgement

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