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Influence of the Boundary Thermal Conditions on the Air Change Efficiency Indexes

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Abstract The influence of a thermal heterogeneity boundary conditions on the air change efficiency (ACE) of a mechanical ventilation system in a test room was experimentally evaluated by means of the "step-down" tracer gas technique in 24 different experimental conditions. The experiments were performed under isothermal condition, varying the air supply temperature with respect to the walls and varying the surface temperature of a wall with respect to the other walls and the supply air, simulating both heating and cooling situations. Changing the position of the outlet grid two different configurations of the ventilation system were tested. The nominal supply air velocity varied between 0.04 and 0.11 m/s, corresponding to a range from 1 to 3 ach, and the temperature differences varied from 0 to 5°C. Results are reported in terms of air change efficiency indexes, both local and global. The global air change efficiency (ACE), values are presented as a function of the Archimedes number (Ar), whose values were in the range 0 to 181. The reported results suggest that the Ar number may be used to organize the ACE values when in the presence of thermal heterogeneity, both in the external envelope and in the supplied air. The obtained results show that there is a logarithmic relation between Ar and ACE. In particular, for both ventilation strategies tested, the increase of the absolute value of Ar leads to an increase of ACE when the supply air is warmer than the walls, and to a decrease of ACE when the supply air is colder than the walls. Under isothermal conditions the Reynolds number (Re) fairly correlates the experimental results.

Key words Thermal boundary conditions; Step down; Buoyant air; Age of the air; Air change efficiency; Archimedes number.

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Introduction

Using ventilation systems to heat or cool a room implies that the supplied ventilation air is warmer or colder than the inside air. The inlet air distribution within the room depends on the thermal boundary condition, both surface wall temperatures of the room and inlet air temperature. Also, the presence of thermal heterogeneity due to a wall at a different temperature (colder or warmer) than the other one and the supplied air induces a secondary convective flow interfering with the main ventilation flow. The interfering secondary convective flow can significantly modify the internal flow and the local and global ventilation quality indexes.

In the 1980s, glazed facades and atria became popular as architectural features in building design. However, in winter and in summer, cold or warm surfaces are often the cause of thermal discomfort due to natural convective flows from these surfaces.

Research has been done concerning natural convection boundary layer flows along a vertical surface (Heiselberg 1994); Nielsen (1991) investigated the effect of a cold wall on room air distribution; Shillinglaw (1977), Chung and Lee (1996) studied comfort problems that occur in the occupied zone.

Research about buoyancy effects on fresh air distribution inside a room was done by Sutcliffe (1991); experiments and theoretical development were done by Sandberg et al. (1981, 1991) for displacement ventilation and Zhang et al. (1996) for air jets. Computations, using computational fluid dynamics (CFD), were done by Gan (1995) to predict indoor air distribution and thermal comfort for different air distribution systems. Also, experimental investigations were performed by Fisk et al. (1997) in a room simulating workstations under warming and cooling conditions.

The aim of the experiment reported in the present paper was to investigate the influence of boundary thermal conditions, varied by changing the tempera-

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ture of inlet air, the temperature of a wall and the ventilation strategies, on the fresh air distribution in a ventilated test room. The results are reported in terms of air change efficiency (ACE).

Experimental Set-Up

The experiments were performed using a test facility based on the controlled ventilation chamber (CVC) (Figure 1), a full size room with dimensions of 2.4×2.4×4.0 m well described in Di Tommaso et al. (1996).

The ventilation system of the CVC includes two variable speed fans with flow rates ranging from 0 to 70 m^3/h (corresponding to about 3 ach), ductwork connecting the fans to the CVC, equipped with ball valves, and four grids alternatively used for air emission or extraction.

The grids, measuring 0.96 (width)×0.18 m (height), are located on the two smaller opposite walls, one close to the ceiling (0.15 m), the other to the floor (0.15 m). Air can be introduced and extracted into or out of the CVC through each of the grids, allowing for different ventilation strategies.

To measure the concentration of tracer gas in air, an infrared gas analyzer able to analyze SF_6, N_2O and CH_4 in the range from 0 to 200 ppm is used, having an accuracy of 1% of the reading and 2% on the full scale.

The CVC has been equipped by a temperature control system (acting only on the four walls with dimensions 2.4×4.0 m) able to keep the walls and the supplied air temperature at a uniform values or with a moderate difference of the order of not more than 5°C. The surface temperatures of all the walls, including the two which are not thermally controlled, and the air

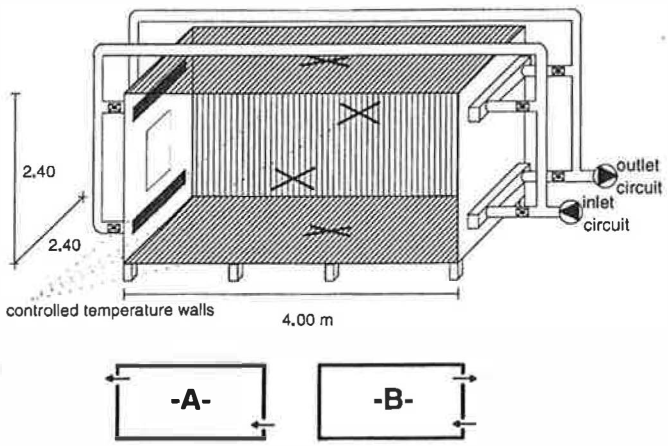

Fig. 1 The controlled ventilation chamber (CVC) and the investigated ventilation strategies

temperature inside the room, in the inlet, and outlet ducts, are monitored by means of 24 K-type thermocouples.

Measurements

In order to measure the air change efficiency of the ventilation system (Sandberg, 1981; Sutcliffe, 1991), the step-down method, using methane (CH_4) as tracer gas, has been adopted.

Two ventilation strategies (Figure 1) were investigated: (-A-), with the inlet grid near the floor and the exhaust on the opposite wall, adjacent to the ceiling; and (-B-), with the inlet and exhaust grids on the same wall, with the inlet positioned near the floor. Under isothermal conditions both configurations reproduce, with different flow patterns, a substantial "mixing" situation.

The measurements have been performed at three different air flow rates: 23 m^3/h (1 ach); 46 m^3/h (2 ach); and 69 m^3/h (3 ach). Air concentration was sampled (by means of a rotary sampling valve) in 25 points uniformly distributed on a moveable plane grid with about the same dimensions of the transversal room section, a 26th sampling point was allocated in the exhaust duct. Measurements were repeated positioning the sampling grid in an orthogonal position respect to the major axis of the CVC, at 0.5, 1.5, 2.5, and 3.5 m (x/L=0.125, 0.375, 0.625 and 0.875) from the wall on which the inlet grid is located.

The measurement protocol consisted of the following steps: first, the value of air flow rate, the ventilation strategy, and sampling grid position were selected; and then the tracer gas was injected inside the CVC and a mixing fan was switched on. When a uniform tracer gas concentration (of about 130 ppm) was reached in the room, the mixing fan was stopped and the inlet fan activated. At this point the tracer gas, present inside the CVC, started to decay.

The total monitoring time was 1.5 hours for flow rates equal to 2 and 3 ach, and of 2 h for a flow rate equal to 1 ach. The sampling duration for each point was 9 s and the total sampling time for completing the 26 consecutive points was of about 4 min (the sampling valve and the sampling procedure is fully described in Di Tommaso et al., 1996).

Permutation of ventilation strategies, air flow rates and sampling grid position, gives a series of 24 measurements that have been repeated changing the thermal boundary conditions of the CVC.

The temperatures of the inlet air and of the walls of the CVC for all the experimental conditions investigated are reported in Table 1.

Table 1 Thermal conditions of the CVC during the measurement series

case	inlet air temperature (°C)	wall temperature (°C)	all other walls temperature (°C)	inside air temperature (°C)
isothermal	20.0	20.0	20.0	20.0
warm air	25.0	20.0	20.0	20.7
warm wall	20.0	23.1 22.4 21.8 21.2 20.7	20.0	20.5 20.3 20.3 20.2 20.2
cold air	15.0	20.0	20.0	19.4
cold wall	20.0	17.0 17.6 18.2 18.8 19.3	20.0	19.4 19.5 19.6 19.7 19.8

Air Change Efficiency Concepts

Sandberg and Skaret (1985) use the terms air change efficiency and ventilation efficiency with different meanings. Air change efficiency is a measure of how effectively the air present in a room is replaced by fresh air from the ventilation system, whereas ventilation efficiency is a measure of how quickly an air-borne contaminant is removed from the room.

The HVAC performances in air change efficiency terms are usually evaluated measuring the well known indexes of local mean age, relative ventilation efficiency and air change efficiency.

The local mean age of air is defined as the average time it takes for air to travel from the inlet to any point P in the room. The local mean age of air within the room can be measured using three different tracer gas techniques (Sutcliffe, 1991). These are the pulse method, the tracer step-up method and the tracer decay method. In the present work, only the decay method has been used and the local mean age can be calculated as follows:

$$\tau_p = \frac{\int_0^\infty C_p(t) \cdot dt}{C_0} \qquad (1)$$

where $C_p(t)$ is the concentration of tracer gas at point p at time t and C_0 is the initial uniform concentration. Local mean ages apply only to the room in which they have been measured and local mean ages for points in different rooms are not comparable.

The relative ventilation efficiency is the ratio of the local mean age that would exist if the air in the room where completely mixed (i.e., τ_n =room volume/flow rate) to the local mean age which is actually measured at a point (τ_p).

$$\varepsilon_p = \frac{\tau_n}{\tau_p} \qquad (2)$$

This index also gives a measure of spatial variations of air distribution in a room. However, since it is normalized respect to τ_n, values obtained in different rooms can be compared.

The air change efficiency is the ratio of the room mean age that would exist if the air in the room were

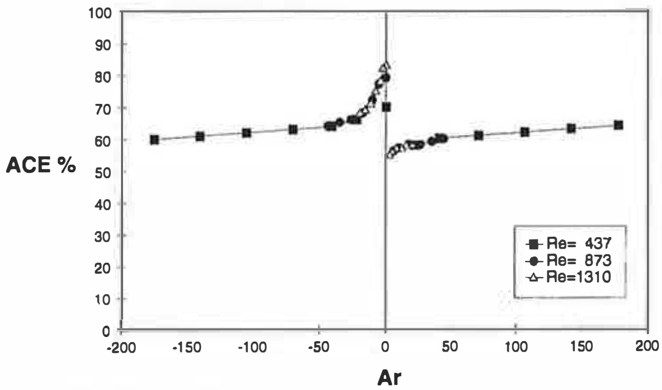

Fig. 2 Air change efficiency vs. Archimedes number at three different Reynolds number for ventilation strategy -A-. Ar positive heating condition; Ar negative cooling conditions

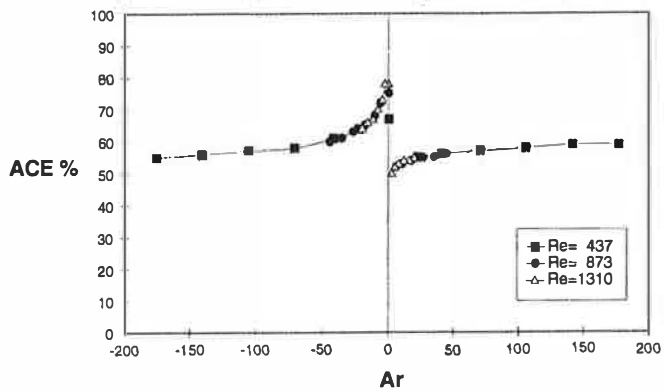

Fig. 3 Air change efficiency vs. Archimedes number at three different Reynolds number for ventilation strategy -B-. Ar positive heating condition; Ar negative cooling conditions

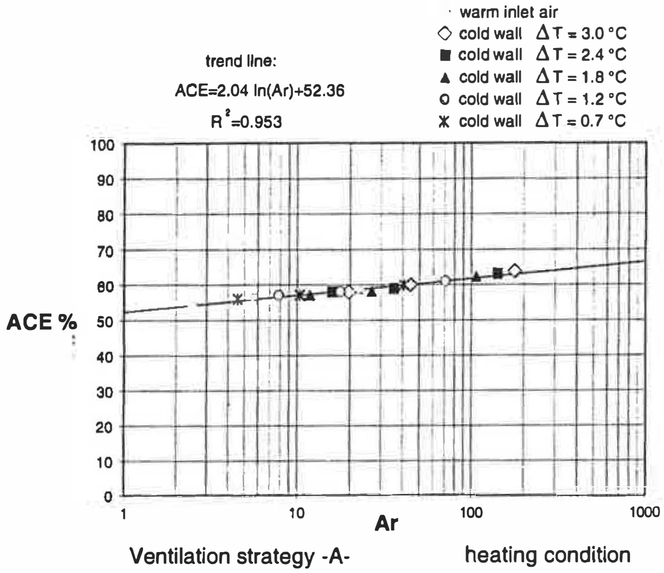

Fig. 4 Curve fitting of air change efficiency vs. Archimedes number for ventilation strategy -A- and heating conditions

completely mixed (τ_n) to the average time of replacement of the room (τ_{exc}) (Sandberg and Sjoberg, 1983).

$$ACE = \frac{\tau_n}{\tau_{exc}} \cdot 100 \tag{3}$$

with:

$$\tau_{exc} = 2 \cdot \frac{\int_0^\infty t \cdot C_e(t) \cdot dt}{\int_0^\infty C_e(t) \cdot dt} \tag{4}$$

where $C_e(t)$ is the measured concentration inside the exhaust duct.

ACE values obtained in different rooms are comparable. A value of 50% indicates fully mixed conditions, a value of 100% implies piston flow and a value less than 50% implies short circuit conditions. These reference values, when compared with the measured value for a particular room, provide an indication of the nature of air distribution in that room.

Measurement Results

The number of combinations of the influencing factors (i.e., supply air velocity, number of air changes, supply air temperature, walls temperatures, geometry of the room and of air inlets/outlets, etc.) is very large, so an attempt has been made to identify a non-dimensional parameter which can correlate the physical phenomenon. It has been observed that for ΔT's "far" from zero there is a good relation between Archimedes number, a non-dimensional indicator that accounts the effect of buoyancy forces to inertial forces, and ACE.

As could be expected for isothermal conditions, the phenomenon is better described by using the Reynolds number, in fact if Ar is equal to 0 (isothermal conditions), and the value of ACE was observed to depend on the amount of supplied air.

Archimedes number has been commonly employed in the study of the so-called "gravitational currents"; for example, it has been adopted (Wilson and Kiel, 1990; Sandberg and Mattsson, 1991) to characterize the study of gravity currents induced from opening a door when there is a temperature difference across it, or to study the jet drop distances into a ventilated air space (Zhang et al., 1996).

Archimedes number is given by the following expression:

$$Ar = \frac{\beta \cdot g \cdot \Delta T \cdot L}{u^2} \tag{5}$$

with:

g-acceleration of gravity
β-volume expansion coefficient
L-characteristic length
u-air velocity

In the present study, the characteristic length was assumed to be equal to the height of the inlet grid (0.18 m) in the cases of cold and hot inlet air, and to the height of the hot or cold wall (2.4 m) in the other cases. The velocity u is the average velocity at the inlet grid. The volume expansion coefficient has been evaluated

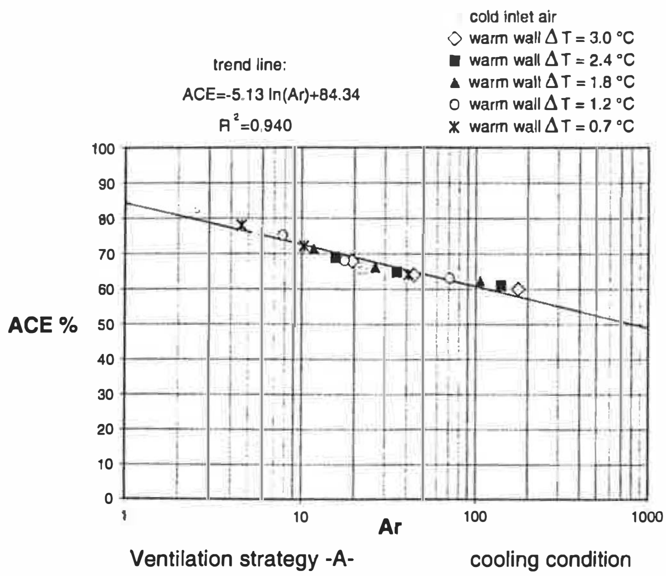

Fig. 5 Curve fitting of air change efficiency vs. Archimedes number for ventilation strategy -A- and cooling conditions

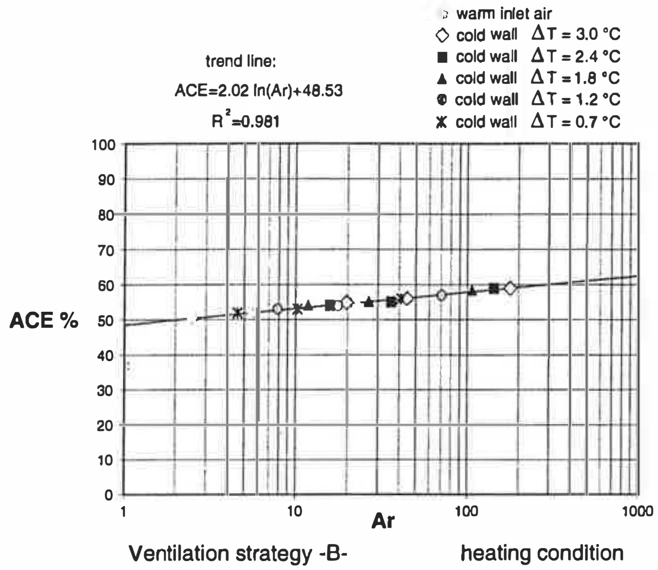

Fig. 6 Curve fitting of air change efficiency values vs. Archimedes number for ventilation strategy -B- and heating conditions

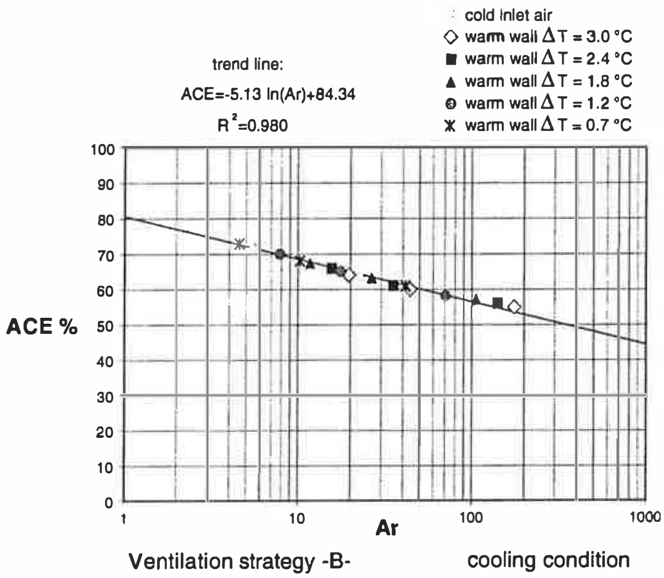

Fig. 7 Curve fitting of air change efficiency values vs. Archimedes number for ventilation strategy -B- and cooling conditions

at the mean temperature between the inlet air and the air close to the wall. The temperature difference is that between the inlet air and the coldest (or hottest) wall of the enclosure. Therefore, in cooling conditions the temperature difference and Ar will be considered negative, and vice-versa, heating conditions will be identified by a positive Ar.

In Figures 2 and 3, all the experimental results (expressed in terms of ACE) are reported for the two investigated ventilation strategies as a function of Ar. The values are also parametrized by the Reynolds number calculated at the exit of the inlet grid (characteristic length equal to the square root of inlet grid surface).

For both ventilation strategies, in heating conditions the ACE values tend to increase with increasing Ar, and vice-versa, to decrease, in cooling conditions, with the absolute value of Ar. There is a noticeable discontinuity around Ar=0, apparently more marked for higher Re values.

The measured ACE vs. Ar is reported in the semilogarithmic scale (in the presence of thermal heterogeneity cases) in Figures 4 to 7. The trends appear approximately linear for all the experimental situations investigated. Curve-fitting with a logarithmic law leads to:

$$ACE = A_1 \ln(Ar) + A_2 \qquad (6)$$

Coefficients A_1 and A_2, and the correlation coefficient (R^2) values, are also reported in the above mentioned figures for the four examined cases. The values of R^2 for the logarithmic interpolation are sufficiently close to 1 (values from 0.94 to 0.98).

Discussion

In heating situations, the ACE values appear positively correlated to Ar. This means that lower velocity and/or higher temperature of inlet air produce a slight increase of the ACE from about 55 to 65%. The same effect may be observed reducing one of the walls temperature respect to the other walls and to the inlet air.

There is a marked decrease of ACE with the absolute value of Ar in cooling conditions, i.e., with lower velocity and temperature of the inlet air. Experiments show that the same effect is produced when one of the walls is warmer than the enclosure and than the inlet air.

Both findings are contrary to the conventional belief and require a detailed explanation. This is presented (only in one experimental condition, for the sake of

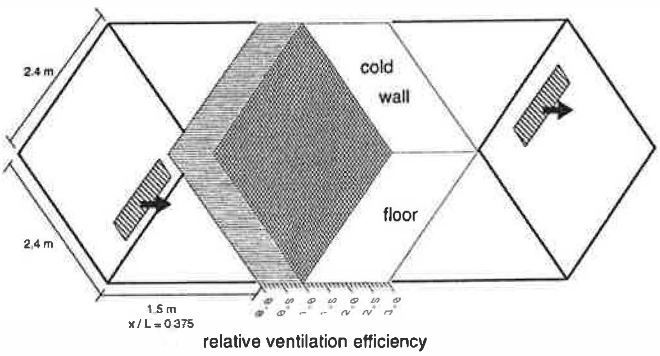

Fig. 8 Relative ventilation efficiency measured in the conditions of "cold wall". Case of ventilation strategy -A-, 3 ach and sampling grid position at 1.5 m from the inlet (x/L=0.375)

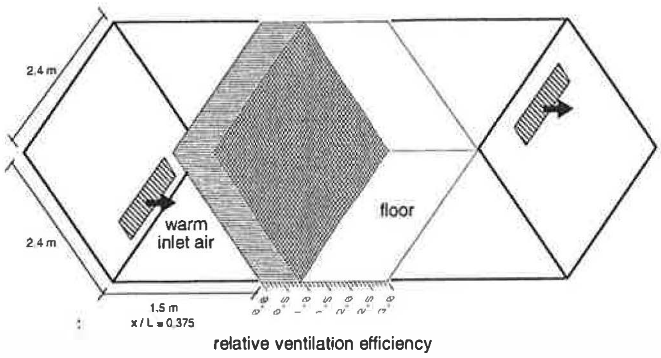

Fig. 9 Relative ventilation efficiency measured in the conditions of "warm inlet air". Case of ventilation strategy -A-, 3 ach and sampling grid position at 1.5 m from the inlet (x/L=0.375)

brevity) in Figures 8 and 9 (heating conditions) and 10 and 11 (cooling conditions). These figures report the relative ventilation efficiency values measured in the case of: 3 ach's, ventilation strategy -A- and position of the sampling grid at 1.5 m (x/L=0.375) from the inlet grid. The measured local indexes are presented in a three-dimensional interpolation form.

Figure 8 is relative to the conditions of "cold wall" in which the flow induced by the ventilation system interacts with a secondary flow, induced by the presence of the cold wall in the region near the floor where the secondary flow reaches the maximum velocity. This implies an increase in the mixing, with local ages of the air equal to the nominal time constant (20 min) and relative ventilation efficiency values equal to one (perfect mixing). In the case of "warm inlet air" (Figure 9), due to the buoyancy effect, a mixing situation similar to the previous one is realized.

In general terms, the mixing effect will increase as the supplied momentum increases, i.e., as Ar decreases. Therefore, the ACE values tend to 50% for Ar tending to zero. Since all measured ACE values are greater than 50%, they are positively correlated with Ar. This interpretation would also imply that ACE would be negatively correlated with Ar if ACE were less than 50%. These are not general results but they are particular of the ventilation strategies investigated and the test chamber configuration. In fact, data measured by Fisk et al. (1997) indicate that even when the ACE is less than 50% there is a positive correlation between ACE and Ar. It would be very interesting to perform heating tests with ACE less than 50% in a future research project, in order to validate this hypothesis. In-cooling conditions we have different situations.

In the case of the "warm wall" (Figure 10) near the floor, the ventilation flow weakly interacts with the convective one that reaches the minimum velocity values there. The ventilation flow is almost undisturbed by the convective one and so, near the floor, relative ventilation efficiency values higher than 2.0 were measured; this is in contrast with the values of about 0.8−1.0 measured in the upper region. These values, typical of a perfect mixing situation, are caused by the convective flow that mix the air near the room ceiling. This situation was observed in all the measurements performed under these conditions. In fact, a perfect mixing situation was observed near the ceiling also changing the inlet flow rate, while near the floor the relative ventilation efficiency increased with increasing inlet flow rate (1.6 for 1 ach and 1.8 for 2 ach). In global terms, this means that ACE will increase when Ar decreases.

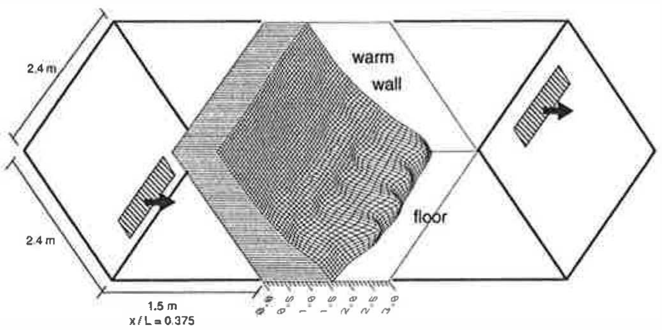

Fig. 10 Relative ventilation efficiency, measured in the conditions of "warm wall". Case of ventilation strategy -A-, 3 ach and sampling grid position at 1.5 m from the inlet (x/L=0.375)

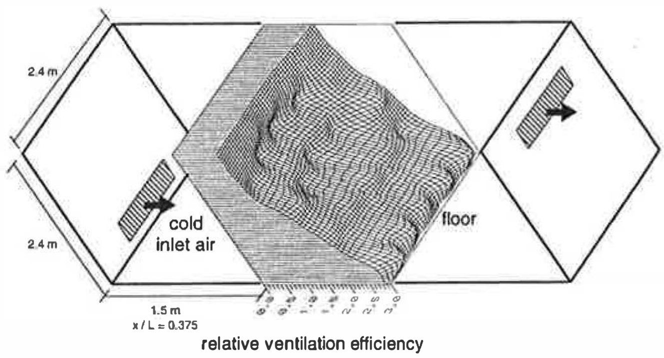

Fig. 11 Relative ventilation efficiency measured in the conditions of "cold inlet air". Case of ventilation strategy -A-, 3 ach and sampling grid position at 1.5 m from the inlet (x/L=0.375)

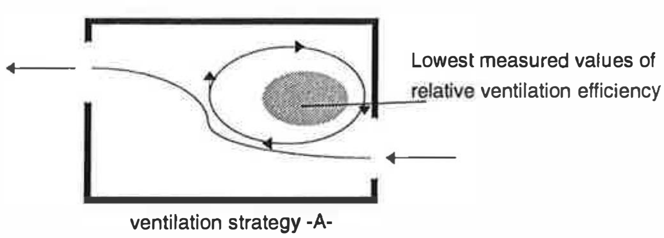

Fig. 12 Hypothesis of "fresh" air distribution for cold inlet air case and ventilation strategy -A-

In the case of "cold inlet air" (Figure 11) the ventilation flow is localized near the floor due to the buoyancy effect. In fact, the relative ventilation efficiency values near the floor are higher than 3.0 for 3 ach (2.0 for 1 ach and 2.6 for 2 ach). Near the ceiling the values are approximately equal to 1.0, except in a small central region in which the values are slightly less than 1.0 (0.7–0.8).

This kind of situation near the ceiling was verified for all the flow rates. There may be in this region a secondary vortex caused by the primary flow (see Figure 12 in which a schematic illustration of the phenomenon is reported). In practice, when the inlet velocity is increased, the situation near the ceiling does not substantially change, but it will change near the floor (the local efficiencies are increased). In global terms ACE values will increase when the velocities are increased, i.e., when Ar decreases.

Conclusions

The control of the thermal boundary conditions in a controlled ventilation chamber (CVC) allowed us to perform 24 experiments simulating both heating (with inlet air warmer than the enclosure or a wall colder than the inlet air) and cooling conditions (inlet air colder than the enclosure or a wall warmer than the inlet air) with two different ventilation strategies.

With both strategies, the increase of the absolute value of Ar leads to an increase of ACE when the supply air is warmer than the walls, and to a marked decrease of ACE when the supply air is colder than the walls. The limit to which ACE tends for Ar tending to zero is different depending on whether it approaches zero from left or right, and the difference is noticeably dependent on Reynolds number (the higher Re, the greater the difference).

Due to the limited number of configurations of the ventilation system which have been investigated, the conclusions which may be drawn from the experimental results should not be considered generally valid, but they are indeed evidence that temperature differences may play an important and unexpected role on the performance of a ventilation system. They also show that Archimedes number, as defined by the authors, is a useful tool for predicting the ACE values in a wide range of experimental conditions.

The experimental results may also lead to the conclusion that, whenever the HVAC system has to accomplish both heating and refrigeration, the optimal position of inlet and outlet grids may substantially differ from the heating to the cooling case because slight variations of Ar around zero may produce quite large differences both in the air flow pattern and in the ACE value.

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