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Impact of Air Distribution on Heat Transfer during Night-Time Ventilation

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ABSTRACT

Passive cooling by night-time ventilation is seen as a promising approach for energy efficient cooling of buildings. However, uncertainties in prediction of cooling potential and consequenses for thermal comfort restrain architects and engineers from applying this technique. Heat transfer at internal room surfaces determines the performance of night-time ventilation. In order to improve predictability, heat transfer mechanism in case of either mixing or displacement ventilation has been investigated in a full scale test room with an exposed ceiling as the dominating thermal mass. The influence of air distribution principle, air flow rate and inlet air temperature were investigated. Results show that for low air flow rates displacement ventilation is more efficient than mixing ventilation. For higher airflow rates the air jet flowing along the ceiling has a significant effect, and mixing ventilation becomes most efficient. A design chart to estimate the performance of night-time cooling during an early stage of building design is proposed.

INTRODUCTION

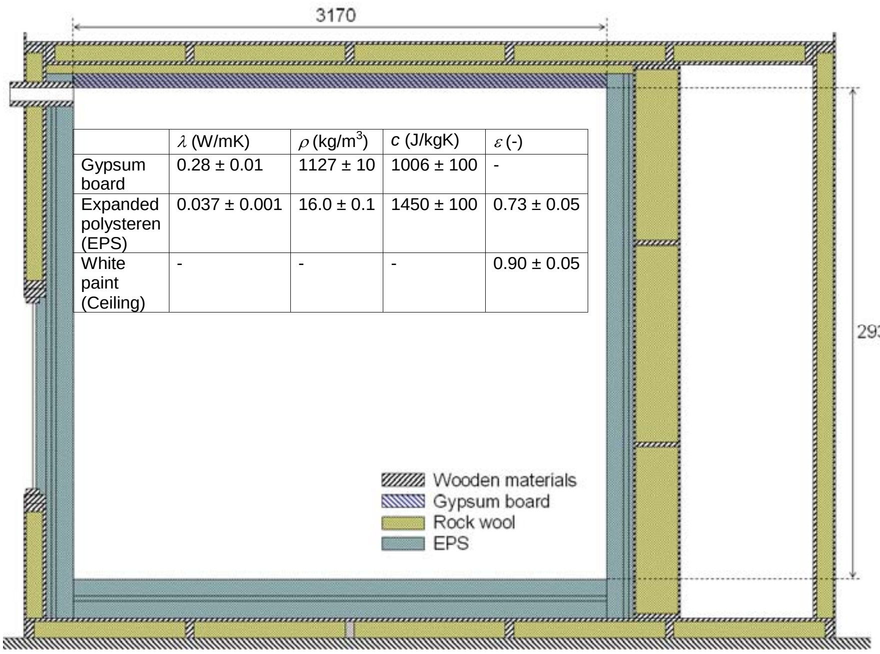
In many countries, a trend towards increasing cooling demand has been observed especially in commercial buildings in the last few decades [1]. More extreme summertime weather conditions [2], higher internal and solar heat gains and increased comfort expectations give rise to an increase in building cooling demand. One possibility to face the increasing energy demand of air conditioning systems is the passive cooling of buildings by night-time ventilation, [3], [4]. The basic idea of the concept is to ventilate a building during the night with relatively cold outdoor air. In the simplest case this can be done by opening windows or, if necessary, by using a mechanical ventilation system. By night-time ventilation heat accumulated in the thermal mass of building elements is being removed. During the next day the cool building elements absorb heat gains, which prevent an extensive increase in indoor temperature.

Despite many successful examples, architects and engineers continue to be hesitant to apply this technique in commercial buildings because of high uncertainties in thermal comfort predictions [5]. One parameter obviously affecting the efficiency of night-time ventilation is the heat transfer at the internal room surfaces [6]. Geros et al. [7] compared monitoring data from a very heavy, massive and free floating building, where night ventilation was applied by natural cross-ventilation with thermal simulations. When the measured air flow rate was used as an input, the simulation model overestimated the performance of the night ventilation. The authors attribute this mainly to the non-efficient coupling of the air flowing through the building to the thermal mass (short circuit air flow). The effective flow rate was then found by adjusting the simulation model to result in the measured indoor air temperature. The ratio between the measured and the effective flow rate was found to be close to 0.3.

This study provides a detailed analysis of convection and radiation during night-time ventilation depending on the air flow rate and the initial temperature difference between the inflowing air and the room. Heat transfer in case of mixing and displacement ventilation has been investigated in a full scale test room.

TEST ROOM SET-UP

A test room at Aalborg University was used for the experimental investigation of the heat transfer during night-time ventilation. For increased thermal mass a heavy ceiling element consisting of 7 layers of 12.5 mm gypsum boards was installed [9]. The walls and the floor were insulated with 160 mm (floor: 230 mm) expanded polystyrene (EPS). After installation of the insulation the internal dimensions were 2.64 m x 3.17 m x 2.93 m (width x length x height) resulting in a volume of 24.52 m^3. A vertical section of the test room is shown in Figure 1, a detailed description can be found in [10]. The thermal properties of the materials used at the internal surfaces of the test room were measured or taken from literature, see Figure 1 (and [10] for details).



	λ (W/mK)	ρ (kg/m^3)	c (J/kgK)	ε (-)
Gypsum board	0.28 ± 0.01	1127 ± 10	1006 ± 100	-
Expanded polysteren (EPS)	0.037 ± 0.001	16.0 ± 0.1	1450 ± 100	0.73 ± 0.05
White paint (Ceiling)	-	-	-	0.90 ± 0.05

Figure 1. Vertical section of the test room.

A mechanical ventilation system was installed to supply air at a defined temperature to the test room. The ventilation system was capable of providing an air flow rate of about 56 - 330 m^3/h, corresponding to 2.3 - 13 air changes per hour (ACH). Two different configurations of the air in- and outlet openings of the test room representing mixing and displacement ventilation were investigated (Figure 2).

In case of mixing ventilation the air inlet to the test room was a rectangular opening of 830 mm width and 80 mm height located directly below the ceiling. To obtain a more uniform velocity profile, two fleece filters were placed approximately 25 and 35 cm before the opening. For the air outlet there were two circular openings with a diameter of 110 mm close to the floor. For displacement ventilation the same rectangular opening below the ceiling was used as outlet and a semicircular displacement inlet device was placed at the floor on the same side of the test room.

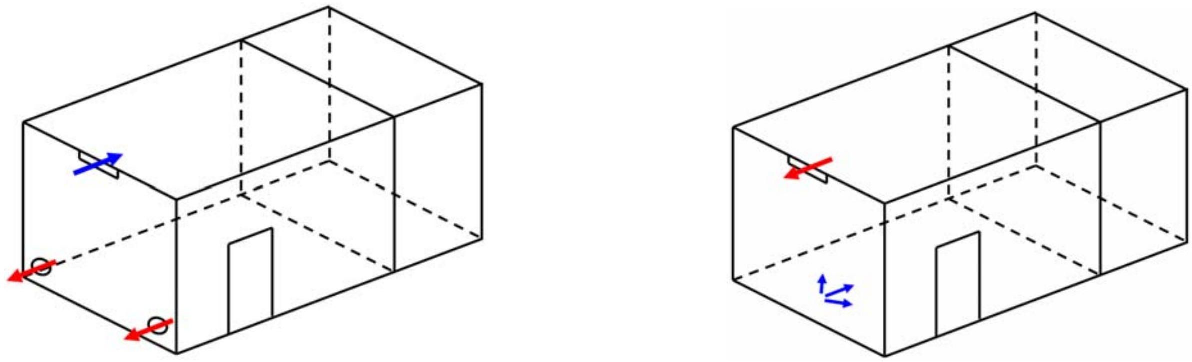


Figure 2. Configurations of the air in- and outlet openings of the test room for mixing (left) and displacement ventilation (right).

The ceiling was divided into 22 sections (Figure 3). At each of the 22 positions 5 thermocouples were installed in different layers (see Figure 4). Additional thermocouples were installed 30 mm below the ceiling to measure the local air temperature.

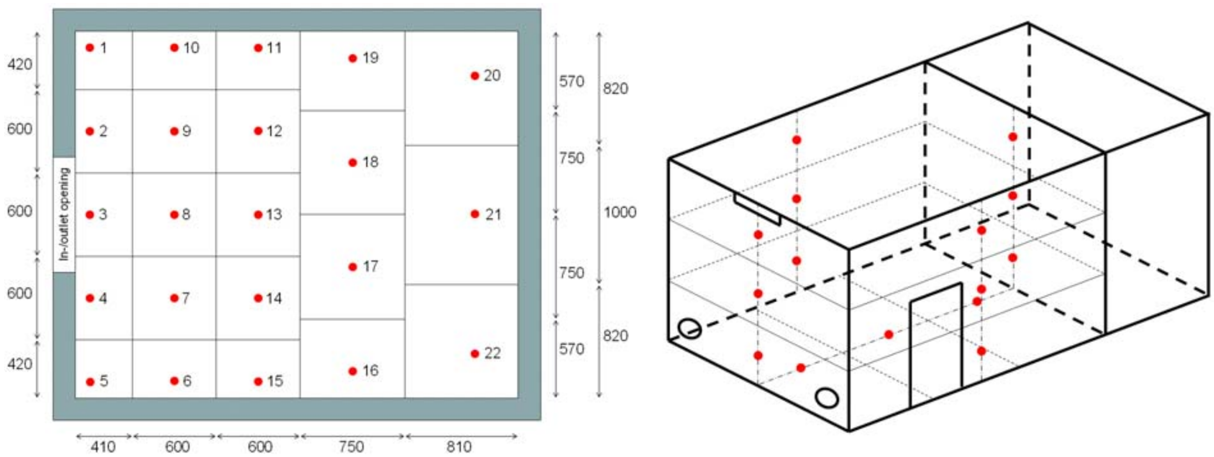


Figure 3. Left) Subdivision of the ceiling into 22 sections and location of sensors (top view), Right) Subdivision of the walls and the floor into 3 sections and location of sensors

The walls and the floor were divided into 3 sections each, and sensors were located in the centre of each section (see Figure 3). At all positions thermocouples were installed to measure the internal surface and the local air temperature at a distance of 30 mm from the surface. To determine the heat flow through the walls and the floor the temperature difference over a 30 mm layer of EPS (100 - 130 mm from the surface) was measured.

To measure the air temperature distribution in the room 3 columns of thermocouples were installed in the vertical symmetry plane. The columns were located below the 2nd, 3rd and 4th row of sensors at the ceiling, i.e. positions 8, 13, and in the middle between 17 and 18. In these columns thermocouples were at heights 0.1, 1.1, 1.7, 2.6 and 2.9 m above the floor (the height 2.9 m above the floor is equal to 30 mm below the ceiling). For determination of the total heat flow removed from the test room by ventilation, the inlet and outlet air temperatures were measured.

In each experiment the response of the test room to a step change in the air flow rate (inflow temperature below room temperature) was measured for at least 12 hours. In total 16 experiments with different air distribution modes (displacement or mixing), air change rates ($ACR = 2\text{-}13h^{-1}$) and initial temperature differences ($\Delta T_0 = 3\text{-}13^oC$) were conducted. The experiments were started with the test room having a homogeneous temperature equal to the lab temperature. The initial temperature difference, ΔT_0 was defined as the difference between the mean temperature of the ceiling element before the experiment and the mean inlet air temperature measured during the last 10 hours of the experiment.

DATA ANALYSIS

For the evaluation of the heat transfer at the ceiling surface, first the total surface heat flow (conduction in the material) for each section was calculated from the measured temperatures. Also the radiative heat flow between the surfaces was determined from the measured surface temperatures. The difference between conduction and radiation then yielded the convective heat flow for each section. By way of example, Figure 4 shows the measured temperatures of the inlet air, and for position 8 at the ceiling, the local air and 5 different layers of the ceiling. For the calculation of the conduction the temperatures measured at the internal (A) and external (E) surface of the gypsum boards were used as boundary condition for a transient 1-dimensional finite difference model using an explicit scheme. To reduce the noise in the measurement signals the moving average of 15 values (2.5 min) was applied. Running the model resulted in the spatial temperature profile for each time step. For each section, *i* the conductive heat flux, $\dot{q}_{cond,i}$ at the surface was calculated from the spatial temperature gradient.

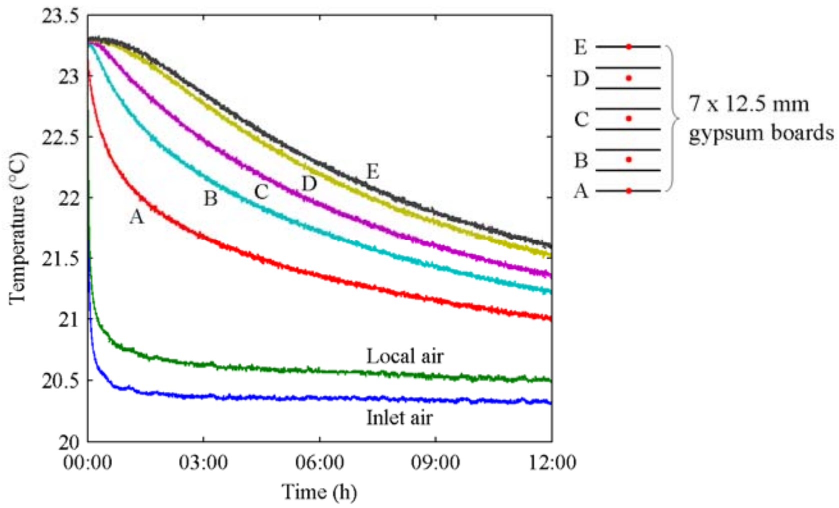


Figure 4. Temperatures measured at position 8 during an experiment with mixing ventilation with 6.7 ACH and an initial temperature difference of ΔT_0 = 2.9 K (A-E: different layers of the ceiling, A: internal surface, E: external surface).

The heat flows through the other walls and the floor were calculated from the measured temperature difference over a 30 mm layer of EPS. In order to account for the thermal mass of the EPS, the heat flux at the internal wall and floor surfaces, $\dot{q}_{cond,i}$ was calculated using the same method as at the ceiling. Here the internal surface temperature and the external heat flux were used as boundary condition for the finite difference model.

The convective heat flux, $\dot{q}_{conv,i}$ for each section, *i* was obtained from the difference between the conductive and radiative heat fluxes. Integrating the convective heat flux over all room surfaces results in the total heat flow removed from the test room, $\dot{Q}_{conv,tot}$. It should be noted, that the total convective heat flow, $\dot{Q}_{conv,tot}$ equals the total conductive heat flow, $\dot{Q}_{cond,tot}$, as by radiation heat is only transported from one surface to another ($\dot{Q}_{rad,tot} = 0$).

Alternatively, the total heat flow removed from the room can also be determined from the flow rate, $\dot{V}_{Air}$, the density, ρ_{Air}, the heat capacity, $c_{p,Air}$ and the temperature difference between the in- and outflowing air. Figure 5 compares the total heat flow obtained by the two different methods. The difference visible at the beginning of the experiment results from the thermal capacity of the air in the room. In the experiment shown in Figure 5 the two methods are in very good agreement. In other cases a difference up to 18 % was found due to measurement uncertainties.

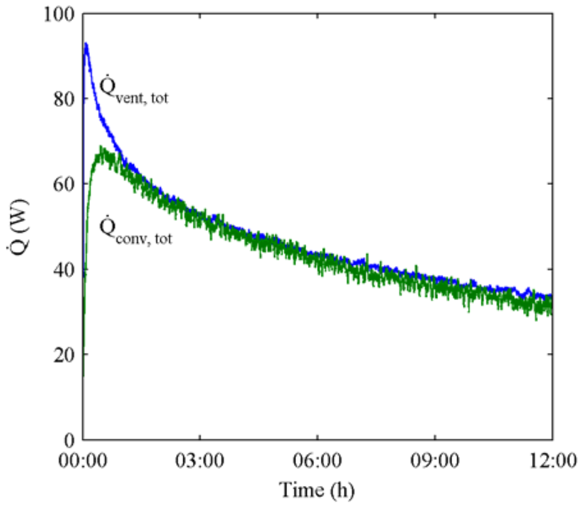


Figure 5. Total heat flow removed from the room obtained from direct measurements ($\dot{Q}_{vent,tot}$) and from integrating the convective heat flows over all surfaces ($\dot{Q}_{conv,tot}$) for an experiment with mixing ventilation, *ACR* = 6.7 ACH, ΔT_0 = 2.9 K.

RESULTS

For the evaluation of the impact of the air jet on the heat transfer at the ceiling, the ratio of the convective and the total heat flow from the ceiling was defined as:

$$\gamma = \frac{\dot{Q}_{conv,Ceiling}}{\dot{Q}_{cond,Ceiling}} \quad (1)$$

The air flow pattern in mixing ventilation is characterised by the dimensionless Archimedes number. To avoid an arbitrary definition of a characteristic length scale, only the temperature difference and the air flow rate, $\dot{V}$ (m^3/s) were used to define *Ar'*:

$$Ar' = \frac{\bar{T}_{Surface} - T_{Inlet}}{\dot{V}^2} \qquad \left(Ks^2/m^6\right) \quad (2)$$

Figure 6 shows the convection ratio γ depending on *Ar'*. During experiments with mixing ventilation, for small Archimedes numbers the inlet air jet is attached to the ceiling and the convection ratio is large. For higher Archimedes numbers the jet tends to drop down. In this case a smaller proportion of the total heat flow is due to convection and radiation becomes dominant. In displacement ventilation, the air flow pattern does not change depending on buoyancy effects and the impact of *Ar'* is small. In all experiments with displacement ventilation less than 32 % of the heat flow from the ceiling is due to convection. The convection ratio increases slightly with increasing air flow rate. During all experiments the temperature difference decreases over time (*Ar'* decreasing). Especially for mixing ventilation the convection ratio tends to increase during the experiment.

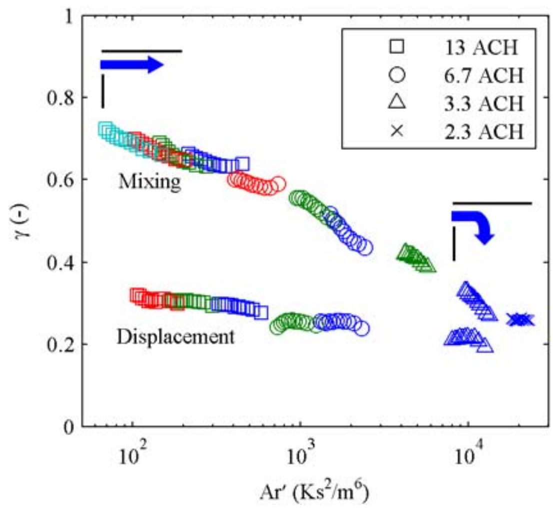


Figure 6. Ratio, γ of convective to total heat flow from the ceiling depending on *Ar'* for mixing and displacement ventilation; hourly values, first hour excluded; different colours relate to different experiments.

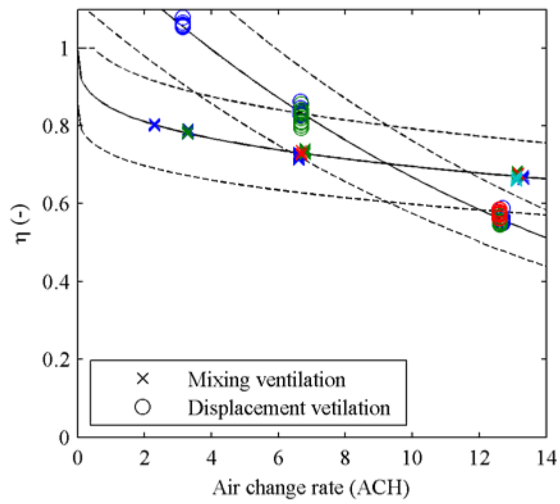


Figure 7. Temperature efficiency, η, depending on air change rate for mixing and displacement ventilation; hourly values, first hour excluded; different colours relate to different experiments. Fitted curves with estimated uncertainty bands (± 14 % [10]).

The performance of night-time ventilation can be described by the temperature efficiency of the ventilation:

$$\eta = \frac{T_{Outlet} - T_{Inlet}}{\overline{T}_{Surface} - T_{Inlet}} \quad (3)$$

The temperature efficiency yielded from the measurements mainly depends on the ventilation mode and the air change rate, Figure 7. During each experiment (excluding the first hour) and for different inlet air temperatures the efficiency is almost constant. For mixing ventilation the efficiency decreases slightly with increasing air change rate. Values between 0.8 and 0.65 were found for air change rates between 2.3 and 13.3 ACH. In a perfectly mixed room, during night-time cooling ($T_{Inlet} < \overline{T}_{Surface}$ and without internal heat sources) the temperature efficiency is limited to 1. In contrast, in displacement ventilation the temperature stratification can result in an efficiency exceeding 1. In the experiment with displacement ventilation at 3.1 ACH the temperature efficiency was 1.06. For higher air change rates, the decrease in the efficiency is more distinct for displacement ventilation than for mixing ventilation. At 12.7 ACH the efficiency was decreased to about 0.56.

DISCUSSION

The experimental results clearly demonstrate the interaction of convective and radiative heat flows contributing to the total heat flow removed from a room during night-time ventilation. For mixing ventilation, different flow characteristics significantly affect the ratio of the convective to the total heat flow from the ceiling. In cases with low convective heat transfer at the ceiling (mixing ventilation at high Archimedes number or displacement ventilation) large differences in surface temperatures cause higher radiative heat flows from the ceiling to the floor.

During experiments with mixing ventilation and an air change rate of about 6.7 ACH, depending on the temperature difference, the convection ratio varies between 43 and 60 %, and at 3.3 ACH between 27 and 42 %. The variation in the convection ratio is caused by the characteristic of the inflowing air jet. Depending on the Archimedes number – the ratio between buoyancy and momentum forces – the cold inlet air flows along the ceiling or drops down into the room. For small Archimedes numbers (small temperature difference, high flow rate) the cold air jet is attached to the ceiling and a large proportion of the heat flow from the ceiling is due to convection. For higher Archimedes numbers (high temperature difference, low flow rate) the jet covers a smaller part of the ceiling surface and the convection ratio decreases.

In the case of displacement ventilation, high local air temperatures (stratification) and low air flow velocities (no jet) result in a small convective heat transfer at the ceiling. On the other hand the radiative heat transfer from the ceiling to the floor (and the lower parts of the walls) is increased because of the large surface temperature difference. Comparing mixing and displacement ventilation at high air flow rates shows very different convection ratios but similar total heat flows. This means that in displacement ventilation the increased radiation compensates for the lower convection. At low air flow rates the inlet air jet is not strong enough to be attached to the ceiling and the convection ratio in mixing ventilation is as low as in displacement ventilation. The total heat flow is, however, higher in displacement ventilation than in mixing ventilation.

Nonetheless, it is beneficial to prevent warm air from accumulating below the ceiling. In displacement ventilation this is achieved through the location of the outlet opening close to the ceiling. At low air flow rates, the temperature stratification and the high location of the outlet opening results in a very high temperature efficiency ($\eta > 1$). In mixing ventilation warm air should be removed from the ceiling by the inflowing air jet. However, only at a relatively high air change rate (above about 10 ACH) the effect of the air jet flowing along the ceiling becomes significant and mixing ventilation is more efficient. Therefore, if a low air flow rate is expected, the outlet opening should be placed as close to the ceiling as possible. The positive effect of the outlet opening being located close to the ceiling, also becomes evident in the high temperature efficiency of displacement ventilation at low air flow rates. In this case the stratification results in an outlet temperature higher than the mean surface temperature (temperature efficiency, $\eta > 1$). With increasing air change rate the efficiency decreases. As the heat flow from the surfaces is limited by conduction in the material and heat transfer at the surface, the difference between inlet and outlet air temperature decreases with increasing mass flow rate. In mixing ventilation the decrease in the temperature efficiency is smaller than in displacement ventilation, since the effect of the jet increases with the air flow rate. In both cases, despite the decrease in efficiency, the total heat flow still increases, as it is proportional to the effective air flow rate, $\eta \cdot ACR$.

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