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Published in:
Proceedings of the 9th Nordic Symposium on Building Physics

Publication date:
2011

Document Version
Publisher's PDF, also known as Version of record

Link to publication from Aalborg University

Citation for published version (APA):
Jensen, R. L., Nørgaard, J., Daniels, O., Justesen, R. O., Madsen, M. S., Mikkelsen, K. B., & Topp, C. (2011). Experimental Investigation of the Heat Transfer in a Room using Night-Time Coling by Mixing Ventilation. In J. Vinha, J. Piironen, & K. Salminen (Eds.), *Proceedings of the 9th Nordic Symposium on Building Physics: NSB 2011* (Vol. Volume 1, pp. 507-514). Tampere University Press.

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Experimental investigation of the heat transfer in a room using night-time cooling by mixing ventilation

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KEYWORDS: *Night-time ventilation, emissivity, mixing ventilation, convective heat transfer.*

SUMMARY:
For many years focus has been on reducing the energy need for heating in buildings. This has lead to buildings with low energy demands for heating but often at the expense of the need for cooling of the building. In order to design buildings with low or zero energy need energy efficient strategies for cooling buildings are required. One such strategy is the combination of thermal mass and night-time ventilation where the excess energy during daytime is store in the building construction and removed by night-time ventilation.

This paper addresses the efficiency of night-time ventilation by the use of full-scale measurements. The efficiency of night-time ventilation depends on the outdoor temperature and the heat transfer between the room air and the building constructions. In a full-scale test room the heat transfer was investigated during 12 hour of discharging by night-time ventilation. Three different air change rates and three different temperature differences between the inlet air and the room temperature resulting in nine different cases where conducted. For all cases the convective and radiation energy exchange was calculated for all the room surfaces.

The ceiling was subdivided into 22 areas and the convective heat transfer coefficient ranged between 5 and 30 W/m2. The ratio of convective to total heat flow from the ceiling depends on the air change rate, ranging from approximately 40% at the low air change rates to approximately 70% at the high air change rate. Even though radiation account for a large part of the energy transfer from the ceiling the results from this investigation compared to previous investigations showed that the total energy exchange from the ceiling was only slightly affected by changing the emissivity of the floor.

1. Introduction

A large part of the energy that is used is used to condition building. Due to focus on reducing the energy needed for heating buildings have become more air tight and highly insulated. The possibility to use large amount of glazing has led to an increased cooling load in buildings. An energy efficient way of cooling a building is to store the excess energy through daytime in the building constructions and discharge the constructions at night time by night time ventilation. The ventilation can be either natural mechanical or a combination. The work presented in this paper focuses on the heat transfer between the building constructions and the room air during a discharge period. The work is a continuation of previous work reported in (Artmann, 2008). The previous investigation has shown that the internal radiation evens out the expected variation of heat transfer due to different ventilation principles as e.g. mixing and displacement ventilation.

In this work the influence of the emissivity of the floor on the night cooling potential, heat transfer coefficient and overall energy distribution in a test room is presented.

2. Test room setup

The test room is a full-scale test room inside a laboratory at Aalborg University. The room is well insulated with its thermal mass placed in the ceiling. The internal dimensions of the room are 2.64 m x 3.17 m x 2.93 m (H x L x W), resulting in a volume of 24.52 m3. A cross section of the room is shown in FIG 1. The EPS is divided into three layers in order to measure the conduction in the middle layer by use of thermopiles.

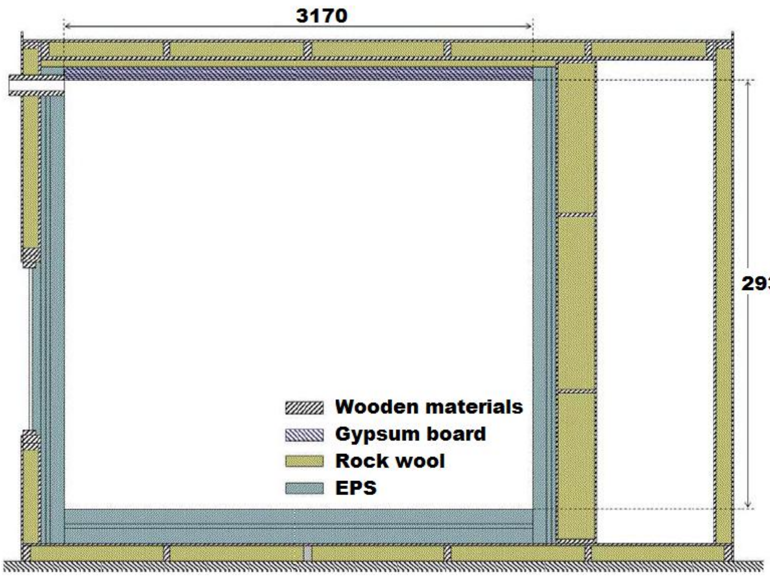

FIG 1. Cross section of test room

The test room is a wooden construction composed by several layers of insulating materials. In order to simulate a heavy ceiling element, seven layers of 12.5 mm gypsum plates are attached at the ceiling. The ceiling surface is painted white in order to achieve similar radiative heat flows between the internal surfaces made of EPS. It can be noticed that there is an empty chamber on one side of the test room. This chamber is present only due to previous experiments that were conducted at this facility and during this study the temperature in the chamber is kept the same as in the laboratory. The properties of the materials use in the test room setup can be seen in TABLE 1.

TABLE 1. Material properties

Material	k $[W/(m \cdot K)]$	ρ $[kg/m^3]$	cp $[J/(kg \cdot K)]$	ϵ [-]
Gypsum board	0.28 ± 0.01	1127 ± 104	1006 ± 100	-
Expanded polystyrene	0.037 ± 0.001	16.0 ± 0.1	1450 ± 100	0.73 ± 5 %
White paint (Ceiling)	-	-	-	0.90 ± 5 %
Mineral wool	0.037	25	800	-
Plywood	0.11	411	1800	-
Aluminium-foil	-	-	-	0.03

The values with uncertainty bands are measured by Nikolai Artmann at Empa (Swiss federal laboratories for materials testing and research) (Artmann and Jensen, 2008). The emissivity of the aluminium-foil is provided by Nikolaj S. Nielsen at Monarflex A/S. The emissivity will change under atmospheric conditions, but as the foil was protected before use and the measurements were conducted right after insertion, a small change has no effect. The important thing is that the emissivity of the floor is changed radically. The remaining values are estimates from a material database. When running mixing ventilation, the air is exhausted from two openings above the floor. The air is supplied through an inlet, which is mounted adjacent to the ceiling, visible in FIG 1. The configuration of the ventilation principle is shown in FIG 2.

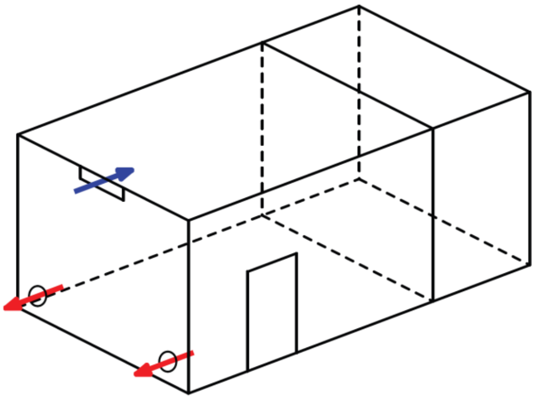

FIG 2. Configuration of air in- and outlet of the test room for mixing ventilation

FIG 3. Picture of the floor with aluminium-foil

A mechanical ventilation system is used in order to supply air at the defined temperature to the test room. The system is able to provide an air flow of 56 – 330 m3/h, which corresponds to 2.3 – 14 air change rate per hour (ACR). Air flow rate was measured using an orifice installed in the supply duct. The pressure difference on the orifice was measured by means of a micro-manometer. The total uncertainty regarding mass flow measurements were estimated to ± 12 % (Artmann and Jensen, 2008).

In FIG 3 a picture of the setup with the aluminium-foil in the floor is shown. The foil is taped to the floor to keep it from moving during experiments.

2.1 Measuring instrumentation

In order to measure the temperatures, thermocouples are attached to the ceiling, walls and floor.

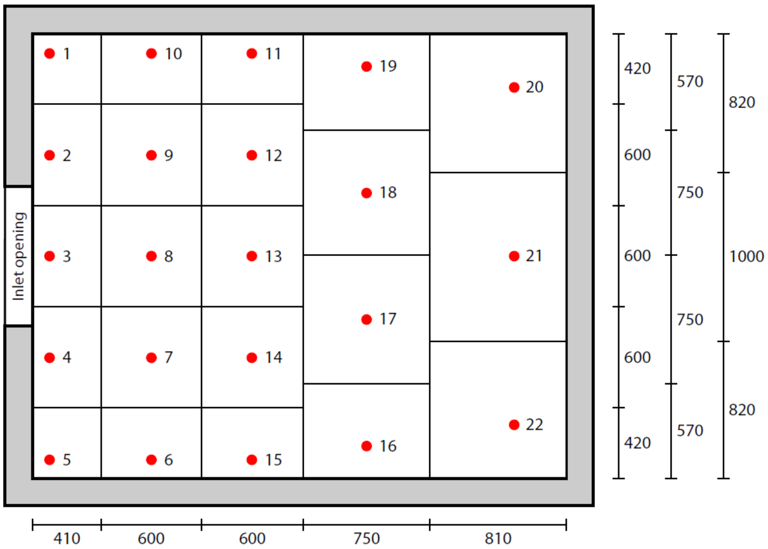

FIG 4. Ceiling element divided into 22 sections. All units are in mm.

All interior surfaces are divided into smaller sections and thermocouples are then attached in each section. Starting with the ceiling, which is divided into 22 smaller sections, five thermocouples are installed in different layers, totalling 110 thermocouples. The division of the ceiling is shown in FIG 4.

At all positions indicated in the previous figure additional thermocouples are installed to measure the internal surface and the local air temperature at a distance of 30 mm from the surface. The location of thermocouples in the five layers of the ceilings cross section is shown in FIG 5. In addition to the

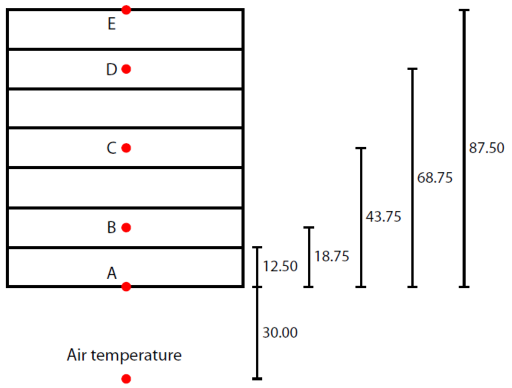

FIG 5. Attachments of thermocouples within the ceiling element, cross section view (gypsum layer). All units are in mm.

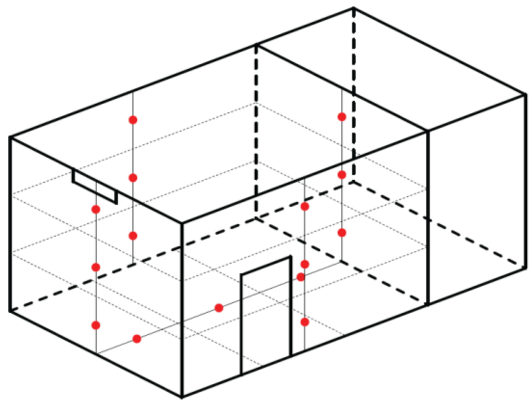

FIG 6. Subdivision of the wall and floor into three minor areas

mentioned thermocouples, a series of thermocouples were also attached 30 mm under the ceiling surfaces in order to measure the local air temperatures. As it was desirable not only to measure temperatures at the ceiling surfaces, but also at the walls and floor. These surfaces were also divided into smaller sections. All wall surfaces and the floor were divided into three minor surface areas and thermocouples were then attached likewise the ceiling. This is displayed in FIG 6.

The temperature difference over the second layer of EPS, visible in FIG 1, is measured using thermopiles for best possible accuracy (Artmann et al., 2008). The measured temperature difference is used to calculate the heat flow through the walls and the floor. To determine the temperature distribution in the room three columns of thermocouples are placed on a line in the centre of the room, visible in FIG 6 as the three dots in the floor of the test room.

2.2 Measurement procedure

A total of 9 experiments were conducted for mixing ventilation with aluminium-foil on the floor. The variables are the air change rate (n) and the temperature difference (ΔT_0) between the inlet and the room temperature, giving a variety of a simplified 'dimensional' Archimedes numbers. This describes the characteristics of the inlet flow and is introduced as:

$$Ar' = \frac{\bar{T}_{surface} - T_{inlet}}{V^2} \tag{1}$$

Where Ar' The 'dimensional' Archimedes number [Ks^2/m^6]
$\bar{T}_{surface}$ Mean surface temperature [^oC]
T_{inlet} Inlet air temperature [^oC]
V^2 Air flow [m^3/s]

The variables, with their individual case number are shown in TABLE 2.

TABLE 2. List of cases

No	1	2	3	4	5	6	7	8	9
n [h^{-1}]	13	6.7	3.3	3.3	13	6.7	13	6.7	3.3
ΔT_0 [K]	2.9	6	5	2	8	2.9	5.3	10.2	8
Ar'	370	2881	9897	3959	1020	1393	676	4898	15835

To ensure steady state starting conditions, the settings of the ventilation system are adjusted approximately 12 hours before the measurement start. This should ensure a stable air flow and a steady temperature provided by the cooling device and pre-cooled ducts from the ventilation system. At experiment start-up a bypass valve is turned, letting the air into the room. The measurements are running for 12 hours to imitate the night-time ventilation period. The balanced ventilation is adjusted so that a slight positive pressure occurs, to avoid infiltration from the laboratory to the room.

3. Results

The results obtained by adding an aluminium-foil to the floor will in this section be compared to a previous investigation with floor in EPS, reported by (Artmann et al., 2009).

3.1 Ratio of convective to total heat flow

Different air flow patterns and the development of the inlet air jet due to these affects the heat transfer at the ceiling. To show the distribution between the heat transfer caused by convection and by radiation, the ratio of the convective to the total heat flow from the ceiling is defined as in the following equation:

$$\gamma = \frac{Q_{conv,ceiling}}{Q_{cond,ceiling}} \tag{2}$$

Where γ Convection ratio [-]

$Q_{conv,ceiling}$ Heat flow by convection at the ceiling [W]

$Q_{cond,ceiling}$ Heat flow by conduction at the ceiling [W]

The ratio of the convective to the total heat flow, γ, is shown as a function of Ar' in FIG 7. Each marker represents one hour of the twelve hour measurement period. As expected, a case with low Archimedes number has heat flow dominated by convection, caused by the cold inlet jet attaching to the ceiling. A high Archimedes number has flow dominated by radiation because the air drops down short after entering the room.

It is noticeable that insertion of the aluminium-foil yields a consistently higher ratio of the convective to the total heat flow. This is believed to be due to the reflection of the foil, which reduces the ceilings heat exchange with the floor by radiation.

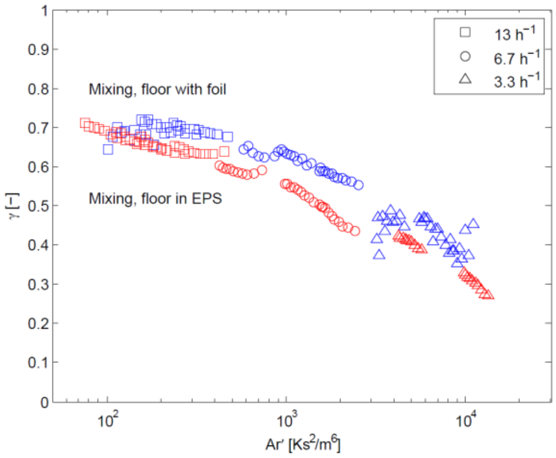

FIG 7. Convective to total heat flow as a function of Ar'.

3.2 Performance of night-time ventilation

Evaluation of the performance of night-time ventilation is done with the temperature efficiency of the ventilation, η, which is defined as in the following equation:

$$\eta = \frac{T_{outlet} - T_{inlet}}{\overline{T}_{surface} - T_{inlet}} \qquad (3)$$

Where η Night-time ventilation efficiency [-]
T_{inlet} Inlet air temperature [°C]
T_{outlet} Outlet air temperature [°C]
$\overline{T}_{surface}$ Mean surface temperature [°C]

FIG 8 illustrates the temperature efficiency, η, for mixing ventilation with the different floors. A decrease of the temperature efficiency was expected, since the ceiling´s radiation to the floor is reduced by the foil. However, the temperature efficiencies with and without foil are very similar. The uncertainty bands can disguise different results that might be as the expected. The results without foil may be positioned in the upper band, whereas the results with foil may be positioned in the lower band, yielding the expected results.

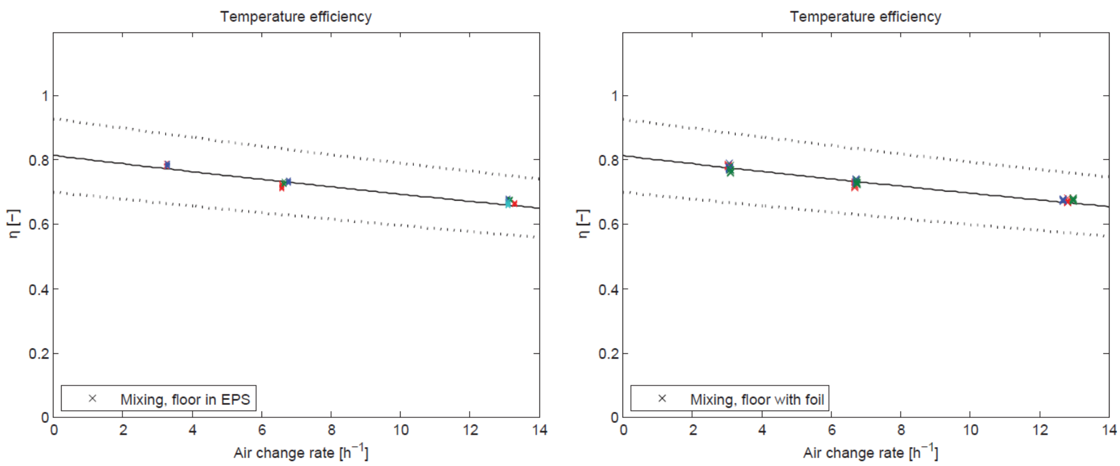

FIG 8. The temperature efficiency for floor in EPS and floor with foil. Uncertainty bands ±14 % (Artmann and Jensen, 2008).

3.3 Convective heat transfer coefficients

During each experiment, the temperature difference decreases over time and as a consequence the heat flux also decreases. For both experiments the relation is very close to linear. The gradient of the lines can be interpreted as the average convective heat transfer coefficients, which can be calculated from the following equation:

$$h' = \frac{\dot{q}_{conv,tot}}{\bar{T}_{surface} - T_{inlet}} \tag{4}$$

Where h Average convective heat transfer coefficients [W/m²K]
$\dot{q}_{conv,tot}$ Mean convective heat flux [W/m²]
$\bar{T}_{surface}$ Mean surface temperature [°C]
T_{inlet} Inlet air temperature [°C]

In FIG 9 the mean convective heat flux from all room surfaces is presented as a function of the temperature difference between the mean surface temperature and the inlet air temperature.

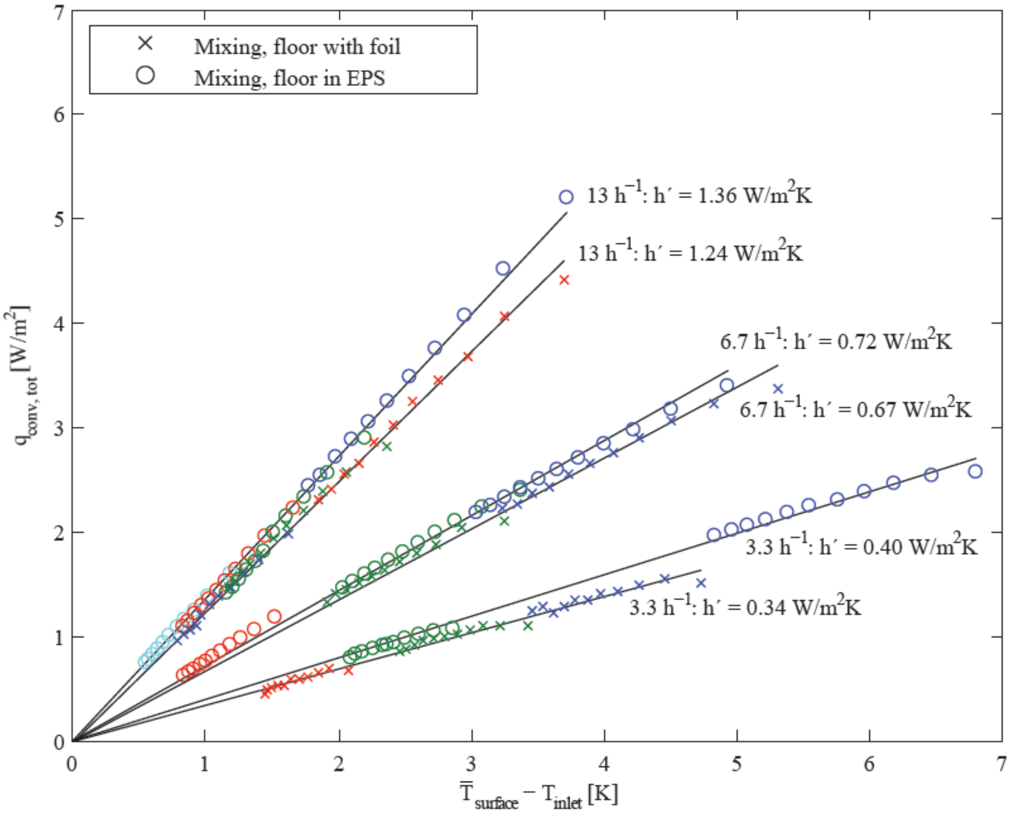

FIG 9. Mean convective heat flux as a function of temperature difference. Linear gradients represent the average convective heat transfer coefficients. The different colours represents different planned temperatures differences in each measurement series, each air change rate being a measurement series. The temperature differences and the air change rates are carefully planned to cover a wide spread of Ar'.

4. Conclusion

In this work the night-time cooling potential was investigated by cooling a light room with a heavy ceiling by a constant air change rate and inlet temperature for a period of 12 hours.

The present results where compared to previous results obtained in the same experimental room. The only difference is that in the present work the floor that was made of expanded polystyrene was covered with an aluminium-foil changing the emissivity from 0.73 to 0.03.

In general the use of the aluminium-foil had a small impact on the night-time cooling potential and the overall result including the surface average convective heat transfer coefficients. For building simulation applications this means that there is little reason to try to incorporate heat transfer coefficient that depend on the emissivity of the surfaces.

However the change emissivity at the floor resulted in some variation of the ratio of convection to total heat flow and the mean convective heat transfer coefficient. For the ratio of convection to total heat flow the results from the two investigations are scattered around each other and no clear conclusion can be drawn. Using the foil reduced the mean convective heat transfer coefficient by 0.05 – 0.12 W/ m² K corresponding to 7-15 %.

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