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Published in:
Proceedings of the 10th Nordic Symposium on Building Physics

Publication date:
2014

Experimental investigations of heat transfer in Thermo Active Building Systems in combination with suspended ceilings

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KEYWORDS: *Thermo active building systems, suspended ceilings, heat capacity*

SUMMARY:
Thermo Active Building Systems (TABS), described as radiant heating or cooling systems with pipes embedded in the building structure, represent a sustainable alternative to replace conventional systems by using source temperatures close to room temperatures. The use of suspended ceiling in office buildings to cover acoustic requirements hinders the use of TABS. To measure the reduction of the heat capacity, several experiments are performed in a room equipped with TABS in the upper deck and mixing ventilation. The heat transfer is measured for different suspended ceiling covering percentages, occupancy scenarios and ventilation rates. The gained results indicate that the heat capacity coefficient of the ceiling surface is reduced by around 30% when the suspended ceiling covering is 70% of the total ceiling area, and 45% when the covering area is up to 87%. The results also demonstrate that the ventilation rate has a high influence on the convective heat capacity. When the ventilation rate is increased from 1.7 h^{-1} to 2.9 h^{-1}, the heat transfer coefficient increases up to 16% for the same occupancy and suspended ceiling layout.

1. Introduction

Global energy-related CO_2 emissions from human activities have continuously increased over the past years. Buildings account for around 32% of the total energy consumption of the countries members of the IEA, and it is in the provision of heating, cooling and ventilation where most of the energy is consumed (International Energy Agency 2013). Therefore the building sector must address the reduction of economic and environmental costs of energy used by means of technology developments, market strategies and government policies.

In this context, Thermo Active Building Systems (TABS), commonly described as radiant systems with pipes embedded in the building structure, recently appeared as a new alternative to cool or heat a space. The main difference with other radiant systems is that they benefit from the thermal storage capacity of the building structure. Due to the heat absorption of the thermal mass, the energy is stored and released over an extended period of time reducing the peak loads and temperature fluctuations.

These systems have been successfully implemented in mainly multi-storey office buildings where the benefits of the energy storage can be distributed from storey to storey (Schmidt 2004). Frequently office buildings require the installation of suspended ceilings to cover the acoustics requirements. However, the fulfilment of these requirements compromises the thermal performance of TABS by impeding the convective and radiant heat to be transferred to the space.

Previous researches have investigated the cooling capacity of TABS when the suspended ceiling is either totally or partially covered with acoustic ceilings. Contrary to what was expected, the results of the study performed by E. Pitarello (2008) shows that it is feasible to combine both techniques and still fulfil thermal and acoustic requirements. As a matter of fact, it was discovered that when covering up to 80 % of the ceiling, the cooling capacity was only lowered by around 25-30%.

Another study performed by H. Peperkamp and M. Vercammen (2009) investigates the cooling capacity reduction in relation with the covering percentage and the sound absorption of suspended ceilings. The study concludes that there is no noticeable reduction in the average sound absorption of the 500Hz to 4Hz band when the covering area is reduced from 100% to 80%.

The same embedded pipes can be used for both heating and cooling scenarios but until now, only cooling scenarios have been investigated in detail. Therefore this work investigates the heat transfer from heated ceilings for different suspended ceiling covering percentages and different occupancy scenarios.

2. Experimental Methods

2.1 Test facility

The experiments took place in a thermo active test facility with an internal area 21.6 m^2. The test facility includes a room surrounded by a thermal guard to control the outside conditions of the room. FIG 1 shows the exact geometry of the room and its surrounded guard.

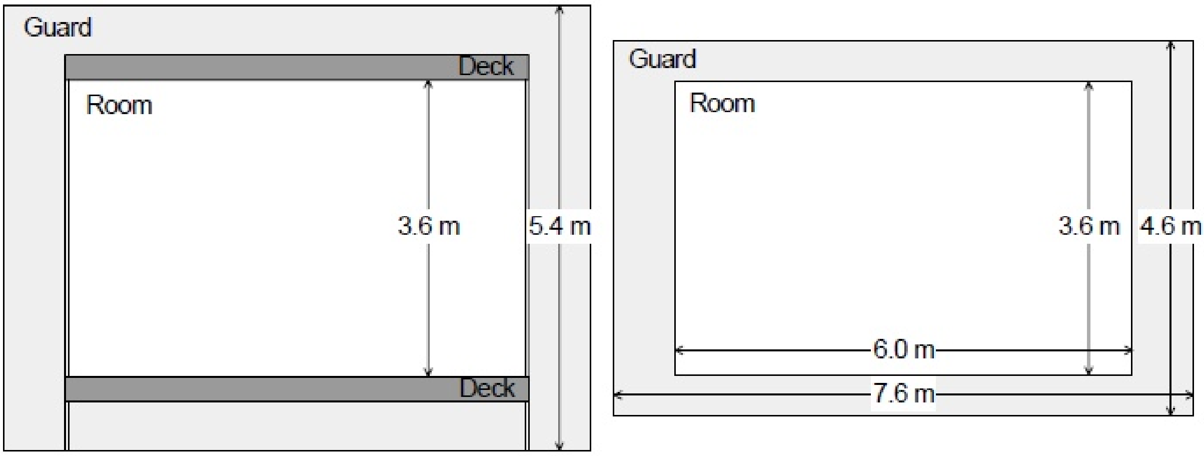

FIG 1. Front and plan view of the test facility with its corresponding dimensions

The upper deck consists of prefabricate hollow concrete slabs with integrated PEX pipes, that contains water at the desired flow rate. The height of the deck is 270 mm and the diameter of the pipes is 20 mm, positioned at 50 mm from the bottom of the deck as displayed in FIG 2.

FIG 2. Concrete hollow deck with integrated PEX pipes

During the investigations the temperature of the guard was set to be the same as the room temperature so that adiabatic conditions could be applied to room, walls and floor. .

2.2 Suspended ceiling

A suspended ceiling was installed in the test facility at a height of 2.7 m from the floor. It consisted of 600x600x15 mm mineral wool tiles mounted on wooden beams.

Three main suspended ceiling layouts were implemented varying in covering percentage (87%, 70% and 53%) as shown in FIG 3.

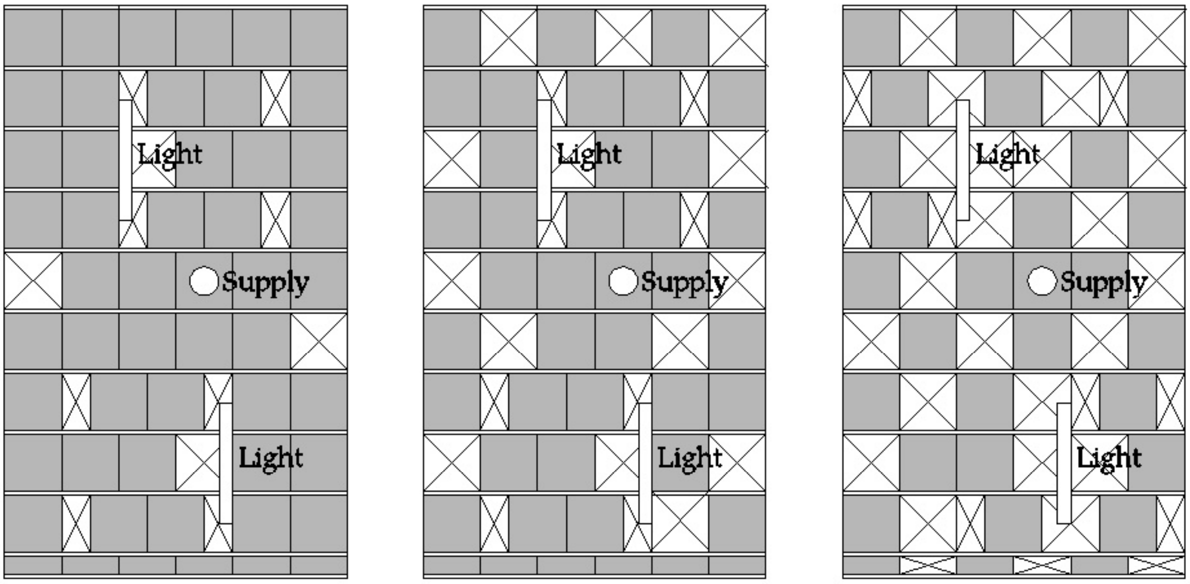

FIG 3. 87%, 70% and 53% suspended ceiling covering layouts

2.3 Equipment

A flow unit was used to provide the required supply water temperature at a constant flow (0.1 kg/s) to the upper deck pipes. A flow meter “Danfoss Mass 1100, DN10” measured the mass flow in the upper deck as an analog signal in the range of 0-20mA corresponding to 0-720kg/h.

In order to ensure the circulation of air at the desired temperature, two fans were positioned inside the guard. The temperature of the guard is controlled by a power controller with a 0-10V signal in order to obtain the desired conditions outside the room; in this case, adiabatic conditions were simulated.

A cooling coil was used to provide ventilation to the room. The temperature was controlled with a PID control system. The air was supplied to the room by a LCA ceiling diffuser.

Concerning internal loads, a double person office was simulated. by using two black barrels as dummies (100 W each), two desktop computers in stand-by mode (68 W each) and two lamps (36 W each) hanging 0.9 m from the upper deck. The total heat load during occupied hours was 18.84 W/m^2.

A LabVIEW application was used to measure and control the data collected by an Agilent 34970A data acquisition unit (data logger). Type TT (Copper/constantan) thermocouples measured absolute temperatures through a build-in function in the data logger, while thermopiles made of either three or five serially connected thermocouples measured temperature differences.

A micromanometer from Furness Controls model FCO510 measured the pressure drop across an orifice from Fläkt model EHBA-012-1 in the supply and exhaust ducts.

2.4 Measurements

The same temperature sensors were used during all the measurement series. Some of the sensors were placed during the construction process so that it was possible to measure the temperature of the concrete deck and the guard, as well as the fluid temperature. Surface- and air temperatures were measured in several locations in the occupied area and the plenum.

Steady state conditions were used during all the experiments. The heating investigations consisted of two series (series “a” and series “b”) of 8 experiments each. Both series were identical except from the fan speed, and therefore the air flow supplied to the room. Experiments number 1a to 8a corresponded to a low ventilation rate (1.7 ± 0.1 h^{-1}) while experiments number 1b to 8b corresponded to a high ventilation rate (2.9 ± 0.1 h^{-1}).

The defined ceiling covering percentages (87%, 70% and 53%) and a non covered layout were tested as well as occupancy/ non-occupancy scenarios.

Each experiment lasted a minimum of 4 days in order to ensure steady state conditions. The data was measured within intervals of 30 seconds and the average of the data collected the last 12 hours of experiment was used for the heat capacity calculations.

2.5 Heat transfer

Energy balance of the deck is assumed. The flows composing the energy balance of the upper deck are defined in Eq. (1)

$$q_{fluid} = q_{down} + q_{up} + q_{guard} \tag{1}$$

Where q_{fluid} heat flow between the pipes and the deck (W/m^2)
q_{down} heat flow through the ceiling surface (W/m^2)
q_{up} heat flow through the floor (W/m^2)
q_{guard} heat losses through the sides of the deck (W/m^2)

The sides of the decks are insulated to minimized heat loss to the thermal guard, q_{guard} . This heat loss has been measured in previous studies demonstrating that it only accounts for around 2-3% of the total heat flows (Weitzmann 2004). In the current investigation, q_{guard} is thus neglected.

To obtain the heat flow through the ceiling surface of the upper deck, it is necessary to analyze first the heat flow through the floor and the heat flow between the pipes and the deck.

Temperature sensors located across the plywood layer used as floor covering make possible to calculate q_{up} through Eq. (2).

$$q_{up} = \frac{1}{R_{floorcovering}} \cdot \Delta T_{floorcovering} \tag{2}$$

Where $R_{floorcovering}$ thermal resistance of the floor covering (m^2K/W)

$\Delta T_{floorcovering}$ temperature difference across the floor covering (K)

The heat flow from the fluid to the deck can be calculated by multiplying the properties of the fluid and the temperature difference between supply and return as shown in Eq. (3).

$$q_{fluid} = \frac{m \cdot C_p \cdot (T_{return} - T_{supply})}{A_{deck}} \tag{3}$$

Where m fluid mass flow rate (kg/s)

C_p fluid heat capacity (J/kgK)

T_{return} fluid return temperature (K)

T_{supply} fluid supply temperature (K)

A_{deck} area of the deck (m^2)

The heat flow through the ceiling surface can be found through Eq. (1), (2) and (3), as simplified in Eq. (4).

$$q_{down} = q_{fluid} - q_{up} \tag{4}$$

Eq. (5) is used to calculate the heat capacity coefficient trough the upper deck.

$$U_{hc} = \frac{q_{down}}{(T_{fluid} - T_{room})} \tag{5}$$

Where U_{hc} heat capacity coefficient through the ceiling (W/m^2K)

T_{fluid} average temperature of the fluid (K)

T_{room} room temperature (K)

The thermal capacity of a heated ceiling can also by expressed as the ratio of heat flow through the ceiling to the temperature difference between the ceiling surface and the room as specified in Eq. (6). $h_{ceiling}$ is the total heat transfer or heat exchange coefficient of the ceiling surface including radiation and convection.

$$h_{ceiling} = \frac{q_{down}}{(T_{ceilingsurface} - T_{room})} \tag{6}$$

Where $h_{ceiling}$ heat transfer coefficient of the ceiling (W/m^2K)

$T_{ceilingsurface}$ surface temperature of the deck (K)

For the calculations, the room temperature is considered as the average between air temperature and mean radiant temperature. The temperature of the fluid is the average water temperature between supply and return.

3. Results

The heat flow through the ceiling surface as a function of the temperature difference between the fluid and the room is illustrated in FIG 4. The graph classifies the data according to the setup of the experiment but there is no distinction on whether internal loads are included or not. Two trend-lines are created to distinguish the results; each of the trend-lines corresponds to one series of experiments. For an equal temperature difference between the fluid and the room, the resultant heat flow is in average 2 W/m^2 larger in the high air flow series than in the low flow ones. The tendency also indicates that the experiments performed without suspended ceilings result in a larger heat flow as function of the temperature difference. And, as expected, the opposite occurs for the layout where 87% of the ceiling area is covered; in this case the results are clearly below their respective trend.

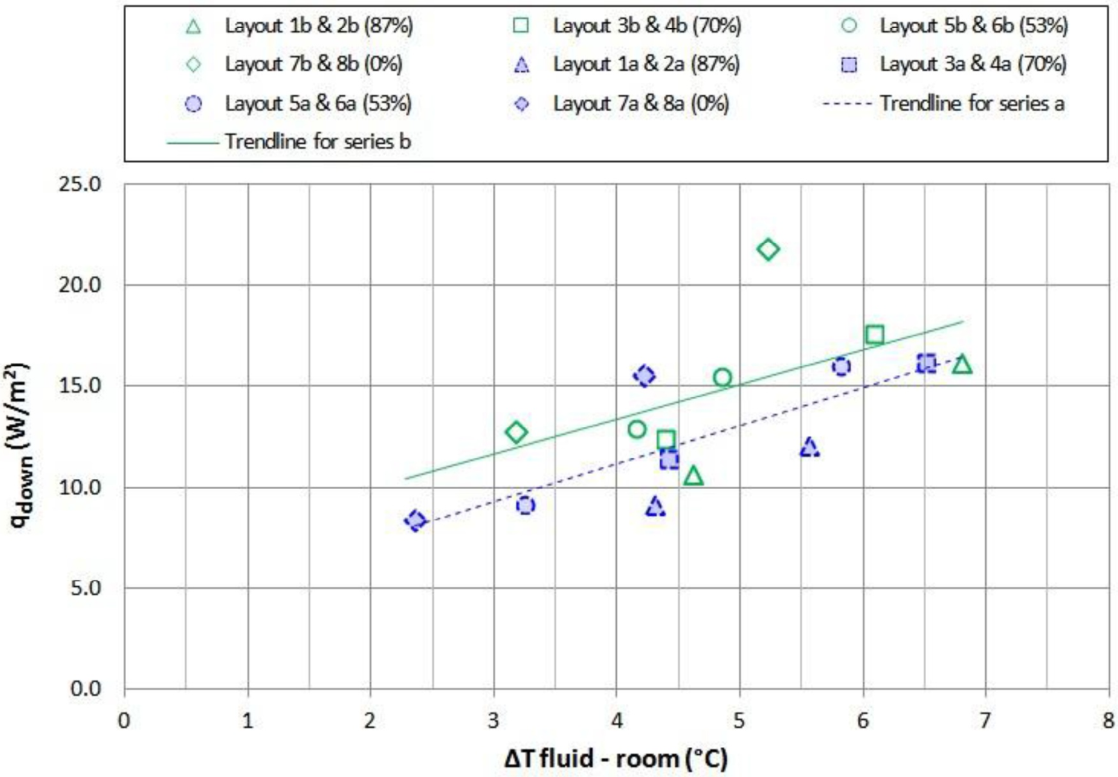

FIG 4. Heat flow through the ceiling surface, q_{down}, as function of the temperature difference between the operative temperature in the room, and the mean fluid temperature in the deck

FIG 5 combines the results of the heat capacity coefficient calculated based on Eq. 5 as function of the covering percentage and the results of the heat transfer coefficient defined in Eq. 6. In this figure, the experiments conducted with and without occupancy are not sorted since the differences between occupancy- non-occupancy layouts are not in any case larger than 5%. The heat capacity coefficient of the ceiling is reduced by 30% when the suspended ceiling covering rate is 70% of the total ceiling area. The reduction is increased to 55% when the covering rate is up to 87% of the ceiling area. It is also observed that series “a” results into a heat capacity coefficient up to 14% lower than series “b”. At the same time, the results of the heat transfer coefficient shows a variation between series “a” and series “b” which is up 16% when the ceiling is not covered. The heat transfer coefficient of series “b” for non covered layouts is up to 6.3 W/m^{2}K, whereas the same value for series “a” is 5.3 W/m^{2}K. At the same time, the results for a covering rate of 87% show a heat transfer coefficient of 3.1 W/m^{2}K and 2.7 W/m^{2}K for series “b” and series “a” respectively.

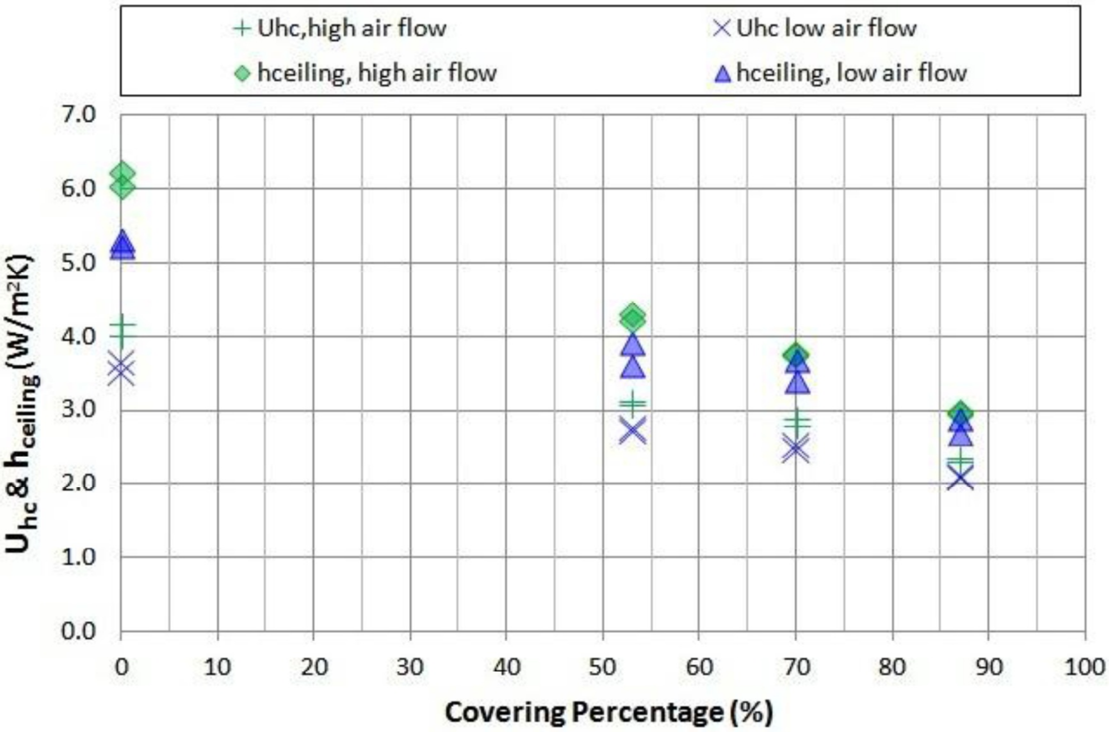

FIG 5. Heat capacity coefficient, U_{hc} and heat transfer coefficient, $h_{ceiling}$, as function of the covered percentage of the suspended ceiling

4. Conclusion

This project investigates the heat transfer of thermo active building systems in combination with suspended ceilings. The gained results show that the combination of suspended ceilings and TABS is a real alternative to conventional heating or cooling systems, especially for low energy or passive buildings where the heating and cooling demands are reduced.

The different layouts investigated show that both the covering rate and the ventilation rate have a great influence on the results. Those experiments conducted with a high ventilation rate result in up to a 16% higher heat transfer coefficient between the ceiling surface and the room than the same experiments performed with a low ventilation rate. The ventilation enhances the convective heat transfer between the ceiling surface and the room and therefore it is assumed that the variation between series correspond to an increase on the convective heat transfer coefficient.

Regarding differences between the results obtained in this project and previous investigations, it can be concluded that, as expected, the cooling capacity of the ceiling is higher than the heating capacity. The cooling capacity coefficient is reduced by 30% when using a suspended ceiling covering rate of 80%, whereas the heating capacity coefficient is reduced by the same range when using a covering rate of only 70%. This proves that the decline on the thermal performance is larger in the heating case.

It can be conclude that the thermal performance of TABS in combination with suspended ceilings is still acceptable for covering rates up to 70%. Coverings larger than that will considerably compromise the performance of the systems.

5. Acknowledgements

This research work would not have been possible without the help and support from the Civil Engineering Department in DTU as well as COWIfonden for financing the project.

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