Implementing air-based thermally activated building systems in retrofitted tertiary buildings – Towards the proof of concept

M Cézard1,2, M Labat1 and S Lorente1,3

1LMDC, INSA/UPS Génie Civil, 135 Avenue de Rangueil, 31077 Toulouse cedex 04 France
2Industrielle de Chauffage Entreprises, Immeuble Le Volta, 17/19 Rue Jeanne Braconnier, 92360 Meudon-La-Forêt, France
3Department of Mechanical Engineering, Villanova University, 800 Lancaster Avenue, Villanova, PA 19085, USA

*Corresponding author: cezard@insa-toulouse.fr

Abstract. Here we document the design method of an air-based thermally activated building system (TABS) suited for the retrofitting of tertiary buildings, for cooling purposes mainly. The first phase of this work provides a general design and checks its consistency with the specifications of tertiary buildings by means of basic energy balances. Second, a numerical model of both the TABS and the room is developed under a finite element method multi-physics environment to better estimate the transient heat transfer for the proposed retrofitting solution. This results in the specifications for building a 1:1 scale prototype whose construction is documented.

Keywords. TABS, retrofitting, Phase Change Material, Finite Element Method, office building.

1. Introduction

Thermally activated building systems (TABS’s) are building parts used as a means of thermal energy storage. TABS’s make use of the building thermal inertia and help dampening indoor temperature variations [1]. Most of the time, this denomination refers to concrete floors or ceilings with embedded water pipes [2]. However, ventilation air can also be used as a heat transfer fluid [3,4,5]. In this paper we present a new system derived from an existing system for new buildings. The original system consists in embedding ventilation ducts within the concrete slab. The air is first circulated within the ducts, where it exchanges heat with the slab, and then blown into the office room. As no false ceiling is used, the slab’s lower face exchanges heat with the room. Unlike hydronic-based systems, the air-based TABS is used here for air renewal, providing indoor comfort, and stressing thermal inertia simultaneously. If air-based TABS’s can be enforced in new buildings with ease, to this day, their implementation in existing buildings for retrofitting purposes remains a hassle.

The proposed solution lies in a dropped ceiling fixed to the slab, with a phase change material (PCM) covering the dropped ceiling upper face. The ventilation air is circulated within the channel formed by the PCM and the slab, and after running all the way through the slab length, is then blown into the room.
The design of this TABS is subjected to some constraints and degrees of freedom which are summarized in Figure 1.

Figure 1. Proposed retrofitting solution with its constraints and degrees of freedom.

The objective of this study is to provide a design that meets these constraints in the framework of a small office room in summer. A pre-sizing will be proposed in Section 2 to define the channel and select a PCM material under some simplifications. Then, a transient study will be achieved in order to demonstrate the ability of the system to improve indoor conditions. These results have been used to design a prototype that is currently being constructed and that will be presented in the last section.

2. Pre-sizing

The pre-sizing will be divided in three parts, each focusing on a different element: the ventilation channel, the room and the PCM.

2.1. Ventilation channel

The channel cross-section is rectangular. However, given that the channel width \( W_s \) is much greater than its height \( d \), the system is equivalent to two parallel plates. Therefore, the system could be studied in the plane (xOy) (see Figure 2). During tests, it appeared that obtaining a turbulent flow was desirable in order to maximize the heat transfer. However, it was hard to reach the turbulent regime regarding the upper limit of the airflow rate (135 m\(^3\).h\(^{-1}\)). The solution consisted in having the TABS covering only half of the room width and using the maximal value of the airflow rate.

The lower plate is made of PCM, which means that it is intended to operate in a reduced temperature range. Thus, as a first approximation, it will be considered isothermal. As for the upper plate, the heat exchanges will be neglected, i.e., the upper boundary will be considered adiabatic. A steady-state heat balance over the \( d \) element, represented by the darker dotted line in Figure 2, gives the air temperature within the channel:

\[
T_a(x) = T_w + (T_{in} - T_w) \exp(-\frac{h_{con} W_s}{m C_p a} x)
\]

Where: \( T_a \) is the air temperature; \( T_w \) is the wall temperature; \( T_{in} \) is the inlet temperature; \( h_{con} \) is the convection heat transfer coefficient; \( W_s \) is the system width; \( m \) is the mass flow rate; \( C_p a \) is the air specific heat capacity and \( x \) is the position within the channel along the x axis.

The channel thickness proved to be of most importance: the smaller it is, the greater are the heat exchanges. The temperature variation between the inlet and the outlet of the channel (\( x = L \)) reached 50% of the maximum decrease \( (T_{in} - T_w) \), against 21% for a 10 cm width channel. Therefore, a channel thickness of 3.5 cm was selected.
2.2. Heat balance at the room scale

The room represents a typical office designed for 2 people, with a floor area of 13.5 m² and a 15 m² outdoor half-glazed wall that is south-west oriented. The solar factor is 0.2 and the glass fraction is 0.5. Such a configuration is selected to stress the need for using a TABS. To illustrate this, let us consider the same room cooled by an all-air system and no TABS.

The outdoor wall is modelled as a purely resistive element ($R = 0.5 \text{ K.W}^{-1}$), but the rather large solar gains through the glazing are taken into account. The five other walls were neglected since they were assumed to have a negligible influence on the heat balance. The total (internal and solar) heat gains are represented in Figure 3. Over the span of 24 h, the total heat gains add up to 8.7 $\text{kJ W}_\text{h}$.

Neglecting the room thermal inertia, a simple heat balance gives the supply air temperature that is needed for maintaining the ambient temperature constant. For comfort purposes however, it is required to avoid supplying some air at a temperature significantly cooler than the indoors. In the present study, it is simply accounted as a minimal temperature of 10°C at the inlet (see Figure 1). For an all-air system, it means that the targeted indoor temperature is not systematically obtained. Figure 3 shows what the ambient temperature would be with a supply temperature limited at 10°C. The TABS will add more thermal inertia to the room, this is where lies its appeal.

2.3. Phase change material

The phase change temperature will be, in a first approximation, defined as the arithmetic mean between the minimum inlet temperature and the ambient maximum temperature, that is: 17.5°C in the present case. A commercial PCM with a phase change temperature of 18°C has been selected: the RT18HC by Rubitherm®. It has the following thermo-physical properties: $\rho_{\text{PCM}} = 825 \text{ kg.m}^{-3}$, $k_{\text{PCM}} = 0.2 \text{ W.m}^{-1}.\text{K}^{-1}$, $C_{\text{PCM}} = 2 \text{ kJ.kg}^{-1}.\text{K}^{-1}$ and $L_h = 260 \text{ kJ.kg}^{-1}$. For a channel that covers 50% of the ceiling, it means that a total of 41.2 kg of this PCM can be implemented. This overload could be easily handled by an existing concrete slab, in the context of a retrofitting. The latent heat associated with such an amount of PCM covers roughly 34% of the cooling needs over the span of 24 h.

Designing a cooling ceiling questions the risk of water condensation on its lower face. Assuming that the indoor dew point is 15°C, the minimal inlet temperature can be obtained through a the computational of the heat flux at steady state, when no phase change occurs, as follows:
\[ T_{in} = \frac{T_{lf} - T_{amb}}{1 - \frac{1}{h_{tot\text{,room}}} \left( \frac{1}{h_{tot\text{,room}}} + e_{PCM} + \frac{1}{h_{conv}} \right)} + T_{amb} \]  

(2)

Where: \( T_{lf} \) is the lower face temperature; \( h_{tot\text{,room}} \) is the total heat transfer coefficient between the room and the TABS lower face; \( e_{PCM} \) is the PCM thickness; \( k_{PCM} \) is the PCM thermal conductivity and \( h_{conv} \) is the convection heat transfer coefficient occurring within the channel. Choosing the most critical parameters for this computation yields a minimal inlet temperature of 7.2°C. Therefore, it is safe to say that using a 10°C at the inlet of the channel will not lead to water condensation over the ceiling.

### 3. Numerical model

The objective here is to demonstrate the ability of the system to improve indoor temperature under a transient regime. Both the TABS and the room have been modelled under COMSOL, a finite element method multi-physics environment.

The air channel, the PCM and the covered concrete slab were modeled in 2D while the room was modeled in 0D and the concrete slab left exposed in 1D. A classical \( k-\varepsilon \) turbulent flow model was used for computing heat and mass transfer within the channel. While heat transfer within the slab were modelled using a classical approach for heat conduction, heat transfer within the PCM were modeled by using the adaptive \( C_p \) technique. See the large increase at the vicinity of the melting temperature in Figure 4.

![Figure 4](https://example.com/figure4.png)

**Figure 4.** Specific heat capacity of the PCM as a function of its temperature.

The indoor heat balance used previously was improved to include the thermal inertia of the furniture and the partition walls, the heat transfer with the newly exposed ceiling; thermal exchanges with the floor were neglected:

\[
K \rho_a C_p a \frac{\partial T_{amb}}{\partial t} = h_{tot\text{,room}} S_{TABS,lf} \left( \bar{T}_{TABS,lf} - T_{amb} \right) + h_{tot\text{,room}} S_{concrete,lf} \left( \bar{T}_{concrete,lf} - T_{amb} \right) + Q_a + \rho_a C_p a (T_{out} - T_{amb}) + U_w S_w (T_{ext} - T_{amb}) + Q_{tot} 
\]

(3)

Where: \( K \) is a coefficient accounting for the thermal inertia brought by the furniture and the partition walls; \( V_r \) is the room volume; \( S_{TABS,lf} \) and \( S_{concrete,lf} \) are the TABS and exposed concrete ceiling exchange surfaces, respectively; \( \bar{T}_{TABS,lf} \) and \( \bar{T}_{concrete,lf} \) are the mean lower face temperature of the TABS and the exposed concrete, respectively; \( Q_a \) is the volumetric airflow rate; \( T_{out} \) is the outlet temperature; \( U_w \) is the facade wall global heat transfer coefficient; \( S_w \) is the facade wall surface; \( T_{ext} \) is the outside temperature; \( Q_{tot} \) is the total heat gains.

The daily input is repeated until a periodic state is achieved, guaranteeing the results independence to initial conditions. Sensitivity analyses related to the mesh, the maximum time step and the relative tolerance have been performed as well.

Figure 5 shows the computed indoor temperature for two different configurations. The volumetric airflow rate is 135 m³. h⁻¹ irrespective of the configuration. The inlet temperature was set at 10°C from 00:30 a.m. to 6:00 p.m. and at 18°C the rest of the time. The first case (denoted \( TABS \)) corresponds to the model with the PCM being implemented and the rest of the concrete left exposed. The second case
(denoted All-air) corresponds to an all-air system, that is, with a false ceiling and no ventilated slab. The supply temperature remains the same, stressing the influence of the PCM and the exposed ceiling. For ease of reference, it is also displayed on Figure 5.

![Figure 5. Ambient temperature for different configurations.](image)

When comparing the two cases, the first one shows a reduced daily amplitude for the ambient temperature. These two configurations would theoretically have the same energetic consumption as the inlet and supply air temperatures are the same, though it might not be possible to have a supply air temperature this cold due to comfort reasons. Even though the TABS configuration yields results closer to the constraints, it does not fulfill them, especially regarding the lower limit of the ambient temperature during occupation (21°C). Improvements need to, and can be made, since only 37% percent of the PCM changes phase over the span of 24 hours.

While solving heat and mass transfer in detail provides some insights on the behaviour of the system, this approach meets its limit when it comes to tailoring a control law for the temperature at the inlet, mostly because of the high computational time. It is foreseen that this limit could be overcome by relying on an experimental set-up. This would also strengthen the feasibility of such a ventilated slab.

4. Experimental set-up
Figure 6 shows the experimental set-up. It comprises a technical part with all the equipment needed to set the inlet airflow at the temperature and the volumetric rate desired. The rest of the set-up comprises a test room and an entrance room to minimize the impact of the comings and goings. The whole is insulated: the upper slab is insulated by 200 mm of glass wool, whereas the cinder block walls comprise 100 mm of glass wool on both sides, resulting in thermal resistances of 5 W.K⁻¹ and 6.30 W.K⁻¹, respectively. Finally, the lower slab is insulated by 100 mm of extruded polystyrene foam ($R = 3$ W.K⁻¹). Furthermore, the whole experimental set-up is located inside a warehouse, thus subtracting it from weather exposure and sudden temperature changes. This should prevent the test room from being influenced by its surroundings in a noteworthy fashion. The prototype is monitored by both temperature and airflow rate probes. A total of 18 temperature probes measures the TABS thermal behavior. Additional probes measure the ambient, and some wall temperatures. The test room has no windows. The internal and solar gains will be mimicked by means of programmable heaters.

The left part of Figure 7 shows the test room hosting the prototype and the entrance room being built. The test room walls are made of cinder blocks. The right part shows a portion of the aluminum dropped ceiling bearing the PCM modules, prior to its installation. It will include insulation between the PCM modules in order to prevent the air blown into the channel from moving below the PCM elements and licking the aluminum dropped ceiling, thus inducing condensation.

5. Conclusion
A methodology was presented to size a ventilated ceiling equipped with PCM. A 3.5 cm thick air channel was retained in order to obtain a turbulent flow for a total volumetric flowrate of 135 m³.h⁻¹. A total amount of 41.2 kg of PCM whose melting point is 18°C was selected to reduce indoor temperature variations. It was verified that such a configuration would prevent water condensation over the lower
face of the dropped ceiling. These results were strengthened by the numerical simulations that highlighted the impact of the PCM against an all-air system. Further work will rely on the use of an experimental set-up being currently built and instrumented to further explore the transient behaviour of a ventilated ceiling.

Figure 6. Experimental set-up.

Figure 7. Test room being built (left), and aluminum sheet receiving the PCM modules (right).

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