

Measure and model of free hanging sound absorbers impact on thermal comfort

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Abstract

A current trend is to consider that the presence of free hanging sound absorbers (FHU) installed in Thermally Activated Buildings (TABS) reduces the thermal comfort by lowering radiative and convective exchanges with the cooled concrete slab. In this study we propose a simple thermal model of FHU which may be implemented into building simulation software like TRNSys. This model considers convective and radiative exchanges but also the air flows above and below FHU. Experimentally, we observe that a ceiling coverage of 50% leads to a operative temperature increase of 0.3°C. In the same conditions, the proposed model very well agrees with measurements. However, for a coverage ratio of 70%, we measure an increase of 1.0 °C and model 0.5°C. New experiments are currently running to confirm these first results and improve the model.

Keywords : TABS, Thermal comfort, Sound absorbers, Free Hanging Units

1. Introduction

1.1. Position of the problem and goals of this study

Using the thermal inertia of the concrete slab between each floor of a building allows increasing thermal comfort by reducing temperature peaks during the warmest days. The problem is then the accumulation of heat into the concrete. In Thermally Activated Buildings (hereafter TABS), different methods are used to discharge the slab. One of them is the TABS buildings with embedded pipes into the concrete. By flowing cold water inside this pipes, it is possible to remove the heat from the slab.

In such building, occupants and heat sources will exchange with the cooled ceiling by convective (40%) and radiative (60%) processes. Direct radiative exchanges are thus the main thermal exchange process. To increase the efficiency of this kind of activated ceiling, it is thus important to expose to the room the largest possible ceiling surface. However, a large surface of concrete decreases acoustic comfort by increasing sound reverberation. A solution is to install Free Hanging Units (FHU). These are usually 40 mm compressed glass wool mats, attached by a lightweight structure to the ceiling. The ratio of the ceiling surface covered by FHU to the total ceiling surface is defined as the coverage ratio (χ , dimensionless). To maintain radiative exchanges with the room, χ must be strictly lower than 100%, i.e. FHU must not totally cover the activated ceiling.

The problem is that the higher the coverage ratio, the better the acoustic comfort. To the opposite, the higher is the coverage ratio, the lower the surface of exposed concrete. This possibly leads to a reduction of thermal comfort by limiting the possibilities of radiatives or convective exchanges with the activated ceiling. The challenge for building designers or architects is thus to find the best balance between thermal comfort, TABS efficiency and acoustic performance.

The present study has two goals. First, it aims to quantify the impact of FHU on thermal comfort in TABS. Second, it aims to propose a first thermal model of FHU in TABS buildings.

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1.2. Literature on the effect of FHU on thermal comfort in TABS buildings

Several authors studied thermal comfort in TABS buildings (for example: Pfafferott et al, 2007). Influence of FHU acoustic comfort has been also extensively studied (for example: Weitzmann et al. 2009). But to our knowledge, the impact of FHU on thermal comfort has been studied in a limited number of papers, particularly in TABS buildings with embedded pipes.

R. Hoseggen et al. worked on the effect of suspended ceilings on energy performance and thermal comfort in an office building in Norway (Hoseggen et al, 2009). This building is equipped with a hybrid culvert-ventilation but do not have any pipes embedded into concrete slab between each floor. Two rooms were monitored: one with exposed concrete, the other with suspended ceilings. A simulation model of the rooms under ESP-r software is also proposed. In their discussion, authors highlight the difficulty to experimentally quantify the effect of suspended ceilings on indoor temperature. Indeed, in a real building, many parameters like occupant presence in adjacent rooms may affect the thermal behavior of a given office. Then, authors conclude that removing suspended ceiling to expose concrete slightly decreases annual space heating demand (from 3% to 7%). The larger accessible thermal mass allows reducing temperature peaks with more than 1 K during the warmest days. But discharging the additional heat stored into this concrete demands up to 17% higher air volume flow rate outside working hours compared to rooms with suspended ceilings.

H. Peperkamp and M. Vercammen aim in (Peperkamp & Vercammen, 2009) to quantify the reduction of thermal capacity of TABS by FHU. Measurements were made in a climatic chamber equipped with a concrete ceiling cooled by water inside pipes. Several gaps between concrete ceiling and FHU and several coverage ratios (40 to 100%) have been tested. They conclude that the thermal capacity of the concrete is reduced when increasing coverage ratio. Hence, if half of the concrete is covered with a suspended ceiling of 40 mm mineral wool ($\chi = 50\%$), the thermal capacity is reduced by approximately 20%. Moreover, the height of the cavity between the suspended ceiling and the concrete does not seem to significantly reduce the cooling capacity.

Similarly, H. Drotleff et al present a very complete study of absorbers strips on thermal and acoustic comfort in the same kind of TABS buildings (Drotleff et al,

2011). They conclude that covering 20% of the ceiling with strips absorbers leads to an increase of temperature lower than 1°C.

S. Morey et al propose in (Morey et al, 2010) a Computer Fluid Dynamics (CFD) study of effect of perimeter gap width around FHU on the access to thermal mass. Authors define the perimeter gap as the distance between vertical walls and FHU placed in the center of a classroom or of an office. However, the considered building does not have any pipes with water embedded into the concrete slab. Building cooling is only ensured by natural or forced ventilation. Several situations are modeled: different ventilations rates (80 or 160 L/s), different ventilations configurations (single sided or cross ventilated), different FHU configuration (single or double raft) and different heat sources (uniform or discretized). Their results show the tangential speed of air on the ceiling as a function of the perimeter gap. They obtain low air speeds, from 0.042 m.s⁻¹ to 0.16 m.s⁻¹. This speed may be, under some assumptions, linked to the thermal exchange coefficient between room air and concrete ceiling. Authors mainly conclude that, in all cases, increasing the perimeter gap produces a linear increase in mean tangential speed of air across the concrete ceiling for realistic gap sizes. This work was completed in 2012 by a set of additional simulations and measurements (Kershaw 2012). Author concludes first that FHU rafts allow convective heat transfer to thermal mass. He also precise that rafts orientation is crucial. Hence, rafts should be orientated along the short axis of the room to allow creation of convection currents. Ideally, they should be also orientated so that they span the office in the direction perpendicular to ventilation source.

Last, J. Fredriksson and M. Sandberg studied in (Fredriksson & Sandberg, 2009) the effect of false ceiling on the cooling capacity of passive chilled beams. Here, chilled beams are panels with inside flowing cold water. According to the authors, chilled beams are often installed above false ceilings for architectural reasons. Air may then flow from the room to the chilled ceiling through an opening of perforated sheet metal or a grating. But in this case, the cooling capacity of the chilled ceiling may be reduced. Authors measured the cooling power for several FHU positions around chilled ceiling and coverage ratios around chilled ceiling (from 0 to 100%). In all experiments, the chilled ceiling was never covered by the FHU. They conclude that going from a totally covered ceiling (except chilled beam area) to a totally uncovered ceiling leads to an increase of cooling capacity from 45% to 61%. FHU thus clearly affect the cooling capacity of chilled ceiling. But these results also demonstrate that the positions of the FHU relative to the walls are very important. In the case of a large uniform heating source on the floor with a chilled ceiling located in the center of the ceiling, the cooling capacity will be influenced by the size of the perimeter gap, as previously defined by S. Morey in (Morey et al, 2010).

Overall, despite this very limited number of papers, authors agree on an impact of FHU on indoor air temperature lower than 1°C, even for the highest ceiling coverage ratios. The precise effect depends on the configuration of FHU in the room and particularly to the horizontal distance between the wall and the FHU.

2. Quantification of thermal comfort in a real TABS building with FHU

The goal of these measurements was to quantify the impact of ceiling coverage ratio on the thermal comfort in a real TABS building. The complete results of these

measurements have already been published elsewhere (Le Muet et al., 2013). We'll thus only remind here the most important results.

2.1. Description of the experimental facility

The WOOPA tower is a TABS building with embedded pipes located in the suburbs of Lyon (FR). In this building, two rooms facing south-east at the same floor were used for experiments. Both approximately measure $4.3 \times 4.2 \times 2.8 \text{ m}^3$. The facade of the building is constructed of wooden profiles, with approximately 50% of triple glazing containing inside built-in blinds. During all the experiments, the blinds were half open, half closed. In the first room ('reference room') no suspended ceiling was applied during all the measurements. In the other ('test room'), no FHU were applied for 8 days in June, 2012. During this period, air and operative temperature were recorded into both rooms in order to compare their thermal behavior. Then, FHU with two ceiling coverage ratios (50% and 70%) were installed into the test room. The suspended ceilings used were $1.2 \times 1.2 \times 0.040 \text{ m}$ thick glass wool ceilings panels from Saint Gobain Ecophon. As shown on figure 1, a cavity of $\sim 220 \text{ mm}$ is placed between the FHU and the ceiling. Two electric heating elements (180 W each) were placed into each room to simulate the presence of occupants during the day (8:00 – 18:00) as described in DIN EN 14240.

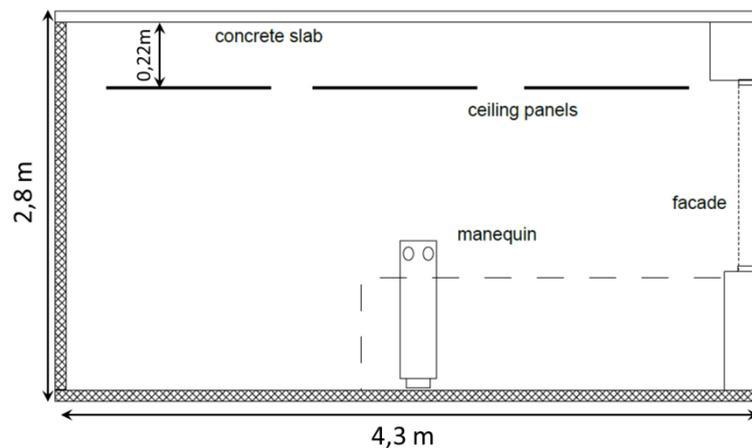


Figure 1: Sketch of the test room in the WOOPA tower

2.2. Results

The two rooms do not have exactly the same thermal behavior: the measured operative temperature in both differs by less than 1°C . This may be explained by several parameters and corresponds to the difficulties underlined by (Hoseggen et al, 2009). However, a more detailed analysis of the results allows Peperkamp to conclude that a ceiling coverage ratio of 50% by FHU leads to an increase of operative temperature inside the rooms by $\sim 0.3 \text{ K}$. And increasing the coverage ratio to 70% is traduced by an increase of operative temperature of $0.8\text{-}1.0 \text{ K}$. These values are in the range of the previous works published in literature. Unfortunately, the impact of FHU orientation or perimeter gap has not been measured.

3. Thermal model of free hanging units

3.1. Scope and limitations of the model

For research or commercial purpose, it is important to evaluate thermal comfort and energy performance of buildings before their construction. TRNSys (TRaNsient SYstem Simulation) is a simulation environment widely used by researchers, thermal consultants or architects to model the dynamic thermal behavior of a building. In TRNSys, buildings are represented by a component called Type 56 (Bradley, 2009). However, TRNSys is today not able to model a ceiling partially covered with FHU. Our goal is thus to propose here a physical model of FHU compatible with the physical description of buildings used in software like TRNSys.

Under TRNSys, a building is divided into thermal zones which are assumed to have the same thermal behavior. Each thermal zone is limited by building surfaces like walls, ceilings, floor, glazing, etc. For example, a thermal zone may represent one room. These thermal zones are themselves divided into one or more thermal air nodes. An air node represents a volume of air which is assumed to be perfectly mixed and characterized by one temperature (Bradley 2009). Under TRNSys, each node is linked to the others through a set of thermal resistances and capacitances. These thermal resistance or capacitance model the physical exchanges in the building: conductive, convective or radiative exchanges with air or surfaces, thermal inertia, etc.

In the present model, we thus consider a room from a TABS building covered with a given amount of FHU as a unique thermal zone. This room is divided into two air nodes by the layer of FHU. The first air node represents the volume of air located below FHU and including building occupants, furniture, etc. The second air node represents the air volume located above FHU. According to TRNSys conventions, both are assumed to be perfectly mixed and temperature homogenous. But because the ceiling is not totally covered by FHU, both air nodes will exchange air thanks to natural or forced convection processes. Occupants and surfaces will also exchange with the ceiling by radiative processes. Two boundary situations must be considered. The first one, represented on the left of figure 2, is a ceiling totally covered by FHU ($\chi = 100\%$). In this situation, there is a convective exchange between the air of each air node and the surrounding surfaces. If we consider the lower air node, these are for example the floor and the lower face of FHU. There is also a radiative exchange between FHU surfaces and floor and ceiling. However, in this situation, we assume that the ceiling is perfectly tight, i.e. there is not any air flow between the two air nodes. Last, it exists a direct conductive exchange through the FHU.

The second boundary situation, represented on the right of figure 2, is a ceiling not covered by FHU ($\chi = 0\%$). In this situation, there is a convective exchange on one hand between the air of the upper node and the TABS ceiling and on the other hand between the air of the lower node and the floor. There is a direct radiative exchange between the TABS ceiling and the floor. And in this case, there is an air flow exchange between the two air nodes. Its intensity depends on the ceiling coverage ratio, on the temperature difference between lower and upper nodes and on the presence in the room of a mechanical ventilation system.

As depicted on figure 2, each convective or radiative exchanges in these two boundaries cases may be represented by a thermal resistance. The value of these

thermal resistance depend on the thermal nodes temperatures, on the intensity of the convective or radiative exchange and on the surface temperatures. However, because TRNSys assumes each air node as perfectly mixed, it is not possible to precisely compute the repartition of air flows and temperature in the room. Any intermediate ceiling coverage ratio ($0 < \chi < 100$) is thus assumed to be an intermediate situation which may be traduced by a linear combination of the thermal resistances obtained in the two previous boundary cases.

In literature, some previous studies already proposed to model under TRNSys rooms with several air nodes. This gave for example the COwZ software (Stewart & Ren, 2006). Unfortunately, these model where not able to take into account the presence of solid elements like FHU between to sub-nodes into the room.

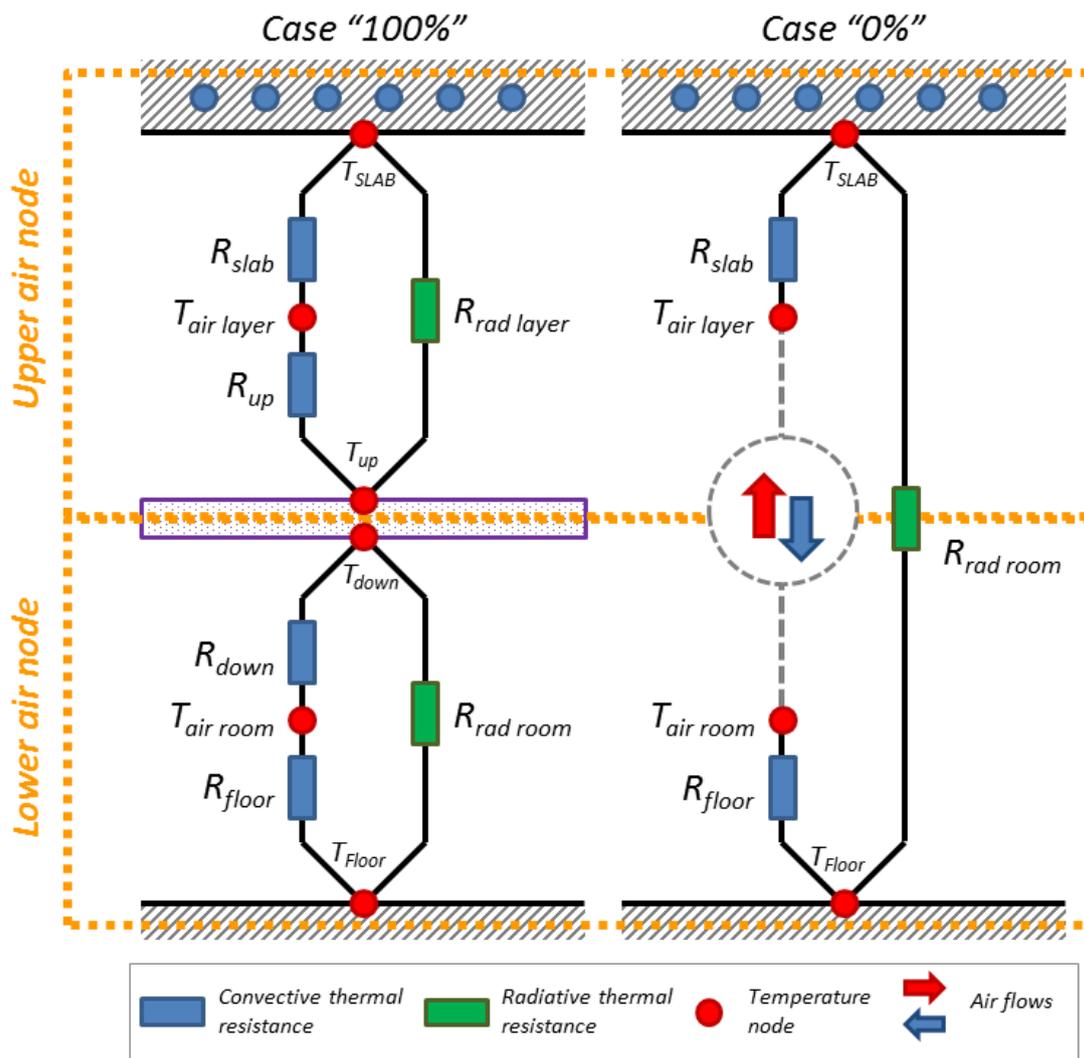


Figure 2. General overview of FHU thermal model

3.2. Modeling air flows through horizontal partitions

TRNSys proposes several ways to take into account airflows on an air node: infiltrations, mechanical ventilation or the so-called coupling. An infiltration represents the air flow entering in a building which is not perfectly air-tight. It will

depend on building orientation and wind speed. Mechanical ventilation represents an air flow imposed by a mechanical system like a cooling device or a fan. Last, a coupling represents the air flow due to natural convection between two air nodes one above the other. The problem is that TRNSys is not able to compute itself the intensity of each kind of air flow. The value must be given by the user. Between 1988 and 1997 an international collaboration in the frame of the IEA Annex 23, the Lawrence Berkley National Laboratory (LBNL, USA) and then the International Energy Agency (IEA) developed and experimentally validated the software COMIS acronym of Conjunction Of Multizone Infiltration Specialists (Feustel & Raynor-Hoosen, 1990). Coupling TRNSys and COMIS has been allowed thanks to the development of a special component called Type 157. With this Type, it is possible to model in the same simulation the thermal and the airflow behavior of a building. Since its version 17, TRNSys directly integrates COMIS in a new extension called TRNFlow. TRNFlow associates an air pressure to each air node. Each node is linked to the others by a set of non-linear conductances which model air paths. These conductances may represent windows, doors, walls, cracks, etc. In the case of FHU in a room, we want to model upward or downward air flows through the spaces between FHU, i.e. through horizontal openings. Unfortunately, COMIS nor TRNFlow are able to do it. This is mainly because, at the time of the creation of COMIS, too few scientific studies on vertical flows through horizontal openings were available to obtain an acceptable physical model (Feustel, 1990). This limitation stayed in TRNFlow.

We thus identified in literature three interesting studies which may be used to model upward or downward air flows through horizontal openings. The first is M. Epstein's pioneer work (Epstein, 1988) who studied the buoyancy-driven exchange flow through small openings in horizontal partitions. In his experiments, he used two tanks separated by a tube of diameter D (in m) and length L (in m). One tank contains clear water when the other contains a solution of brine water. The two compartments thus have a different density. M. Epstein measured the exchange rate between the two compartments for several L/D ratios and for two types of openings shapes: circular or square. Then, he plotted the Froude number Fr as a function of the L/D ratio. M. Epstein shows that four regimes exist and gave a general equation to describe them. This equation is valid for $0.01 < L/D < 20$ and $0.025 < \Delta\rho/\bar{\rho} < 0.17$ where $\Delta\rho$ is the density difference and $\bar{\rho}$ is the average density of the two compartments.

In a second interesting work is the study of L. Zhigang who studied in his PhD thesis the characteristics of buoyancy driven natural ventilation through horizontal openings (Zhigang, 2007). His experiments were carried out in a full-scale test cell divided in two rooms. He studied by velocity field measurements, air flow rate measurements and smoke visualization the characteristics of the air flow due to natural convection between the two rooms. L. Zhigang first concludes that the air flow through horizontal openings is bidirectional (upward and downward), highly transient and complex. Moreover, he mainly confirms that the four regimes identified by M. Epstein in brine water may be applied for air flows through horizontal openings in buildings.

If we consider a room of 18 m² like in the WOOPA tower covered with 70% of free hanging units, we obtain a total area uncovered by FHU of 5.4 m². Using Zhigang notations, this is equivalent to a circle characterized by a diameter of $D = 2\sqrt{5.4/\pi} = 2.622$ m. Knowing that a classical sound absorber has a thickness of $L=0.04$ m, we obtain a L/D ratio of 0.0152. This is comprised in the limits of M. Epstein's and L.

Zhigang models. However, despite their great interest, these two works are not the most adapted to model the air flow from/to the room from/to the air layer above FHU. Indeed, L. Zhigang only studied the case where the lower air volume is warmer than the upper one. Second, he does not take into account the effect of any mechanical ventilation.

S. Vera proposes in his PhD thesis a third numerical and experimental study of air and moisture transport through large horizontal openings (Vera, 2009). His experimental facility consists in two rooms, one above the other. Their temperature may be independently controlled. Both possess also ventilation inlets, ventilations outlets and a moisture source. All may independently be open or closed. The two rooms communicate through an opening with a surface of $0.91 \times 1.19 = 1.08 \text{ m}^2$ and a thickness of 0.22 m. Knowing that the slab between the two rooms measure 8.83 m^2 , we conclude that this opening covers 12% of the total floor surface. S. Vera measured the air flow between the two rooms for several temperatures differences, with or without mechanical ventilation and for several ventilations air speed. He deduces two empirical relations which express the intensity of air flow $F_{lower\ to\ upper}$ (in $\text{kg}\cdot\text{s}^{-1}$) from the lower room to the upper room in case of natural or mixed convection.

$$F_{lower\ to\ upper} = C\rho A\sqrt{gd}\left(\frac{\Delta T}{\bar{T}}\right)^n$$

$$F_{lower\ to\ upper} = C\rho A\sqrt{gd}\left(\frac{\Delta T}{\bar{T}}\right) + b$$

where C , n and b are empirical coefficients, ρ is air density ($\text{kg}\cdot\text{m}^{-3}$), A is opening area (in m^2), g is gravity acceleration ($\text{m}\cdot\text{s}^{-2}$), d is the vertical thickness of the opening (m), T_{lower} and T_{upper} are the air temperature in the lower and upper rooms (both in $^{\circ}\text{C}$), $\Delta T = T_{lower} - T_{upper}$ is the temperature difference between rooms (in $^{\circ}\text{C}$) and $\bar{T} = (T_{lower} + T_{upper})/2$ is the mean temperature between rooms (expressed in $^{\circ}\text{C}$). Depending on the scenario or the temperature difference between rooms, the quantity $F_{lower\ to\ upper}$ may be thus positive or negative. From his experiments as well from numerical simulations, S. Vera gave in (Vera, 2009) and in (Vera et al 2010) the values of the C , n and b coefficients for several airflows rates.

Because the work of S. Vera is valid for buildings, for large openings, with or without ventilation, with positive or negative temperatures differences on both sides of FHU, we conclude that it is the most adapted to model airflows across FHU for ceiling coverage. However, we must underline that S. Vera does not vary the opening area or thickness. His expressions for air flow $F_{lower\ to\ upper}$ depends on a factor $\left(\frac{D}{2}\right)^2\sqrt{d}$ when the relation from M. Epstein and L. Zhigang, obtained on smaller openings, depends on a factor $\sqrt{D^5}$. This point has to be studied for the next versions of the model.

3.3. Convective exchanges

Three different cases must be considered as a function of mechanical ventilation intensity: natural convection (without mechanical ventilation), mixed convection (low mechanical ventilation) or forced convection (high mechanical convection). To discriminate each case, we compute first the value of the Richardson number defined by:

$$Ri = \frac{g\beta(T_s - T_\infty)L}{V}$$

Where T_∞ (in K) is the fluid temperature, T_s is the surface temperature (in K), β the thermal expansion coefficient of air (in K, computed by $\beta = 1/T_\infty$) and V is air velocity (in m.s^{-1}). If $Ri \gg 1$ (i.e. $Ri > 10$ for code implementation) or if there is no mechanical ventilation ($V = 0 \text{ m.s}^{-1}$), we are in a situation of natural convection. If $Ri \ll 1$ (i.e. $Ri < 0.1$ for code implementation), we are in a situation of forced convection. Else, we are in a mixed convection situation.

For each situation, we propose below a relation to compute Nu , the Nusselt number on the surface. The thermophysical properties of air are taken in the tables proposed by (Incropera et al, 2009). From the Nusselt number, we can compute the convective exchange coefficient h_{conv} (in $\text{W.m}^{-2}.\text{K}^{-1}$) of air on the surface by:

$$h_{conv} = \frac{Nu\lambda}{L}$$

Where λ is the air thermal conductivity ($\text{W.m}^{-1}.\text{K}^{-1}$) and L is the characteristic length of the plate (in m), defined by $L = A/P$ with A the area (m^2) of the plate and P (m) its perimeter. In the present model, if we consider the upper air node, the surface will be for example the upper face of the FHU. In this case, A is the total area of the FHU and P its perimeter. Last, we obtain the value of the thermal convective resistance R (expressed in $\text{W}^{-1}.\text{m}^2.\text{K}$) shown on figure 2 by computing:

$$R = \frac{1}{h}$$

3.3.1. Natural convection

In case of natural convection many authors proposed in literature empirical relations to express the convective exchanges coefficients on horizontal surfaces as a function of surface and air temperature. Incropera et al gives in (Incropera et al, 2009) a good summary. If cold fluid falls on the upper surface of a hot plate ($T_\infty < T_s$) or if warm fluid exchange with the lower surface of a cold plate ($T_\infty > T_s$), then the Nusselt number is expressed by:

$$Nu_L = 0.54Ra_L^{1/4} \quad \text{for } 10^4 \leq Ra_L \leq 10^7$$

$$Nu_L = 0.15Ra_L^{1/3} \quad \text{for } 10^7 \leq Ra_L \leq 10^{11}$$

Ra_L is the Rayleigh number (dimensionless) defined by:

$$Ra_L = \frac{g\beta(T_s - T_\infty)L^3}{\alpha\nu}$$

Where α is the air thermal diffusivity ($\text{m}^2.\text{s}^{-1}$) and ν is the air kinematic viscosity ($\text{m}^2.\text{s}^{-1}$). If on figure 2 we consider the upper air node, we'll have $T_\infty = T_{air\ layer}$ and $T_\infty = T_{air\ room}$ for the lower air node. According to the situation, the surface temperature T_s may be T_{down} , T_{up} , T_{slab} or T_{floor} .

3.3.2. Mixed convection

In case of a mixed convection, we use the relation proposed by Taine et al for a horizontal plate submitted to an isotherm air flux (Taine et al, 2008):

$$Nu_L = 2F_1\sqrt{Re_L} \left[1 + \left(\frac{5F_3 \left(\frac{Gr_L}{Re_L^{5/2}} \right)^{1/5}}{6F_1} \right)^3 \right]$$

Where Re and Gr_L are the Reynolds and the Grashof numbers and F_1 and F_3 are two parameters computed by:

$$F_1 = 0.399Pr^{1/3} \left[1 + \left(\frac{0.0468}{Pr} \right)^{2/3} \right]^{-1/4}$$

$$F_3 = \left(\frac{Pr}{5} \right)^{1/5} Pr^{1/2} \left(0.25 + 1.6Pr^{1/2} \right)^{-1}$$

Where Pr is the Prandtl number for air. In the present study, we assume a constant value of $Pr = 0.71$ (Incropera et al, 2009).

3.3.3. Forced convection

The case of a forced convection may arise for higher ventilations, if the ventilation inlet is directly in the ceiling and thus connected with the upper air node. In this case, we use the relation proposed by (Incropera et al, 2009) :

$$Nu_L = 0.664Re_L^{1/2}Pr^{1/3}$$

Which is valid for $Pr > 0.6$, i.e. valid in the present case ($Pr = 0.7$ for air).

3.4. Radiative exchanges

Because we consider a real building, we assume that the temperature differences between each surface are low, i.e.

$$T_{down} - T_{floor} \ll T_{floor} \quad \text{or} \quad T_{up} - T_{slab} \ll T_{slab}$$

Under these conditions, we can linearize the expression of the radiative flux (Taine et al, 2008) and compute the value of the radiative exchange coefficient h_{rad} by:

$$h_{rad} = 4\varepsilon_{eq}\sigma T_m^3$$

Where s is the Stefan-Boltzmann constant, T_m is the average temperature of the two surfaces considered and ε_{eq} is the equivalent emissivity of the two surfaces. To compute its value, we assume two infinite surfaces characterized by emissivities ε_1 and ε_2 (both dimensionless) and we use (Incropera et al):

$$\varepsilon_{eq} = \frac{1}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1}$$

Knowing the value of the radiative exchange coefficient, we compute the value of the thermal radiative resistance R_{rad} (in $W^{-1}.m^2.K$) by:

$$R_{rad} = \frac{1}{h_{rad}}$$

3.5. Conductive exchanges

In the present model, thermal flux may also pass through the FHU (not shown on figure 2). We compute the thermal conductive resistance R_{cond} (in $W^{-1}.m^2.K$) by:

$$R_{cond} = \frac{e_{FHU}}{\lambda_{FHU}}$$

Where e_{FHU} is FHU thickness (m) and λ_{FHU} is FHU thermal conductivity (expressed in $W.m^{-1}.K^{-1}$).

3.6. Numerical solution and implementation of the model

All the equations of the model are implemented in a plug in for TRNSys. The total equation system is solved using the Trust-Region Dogleg method. This plug-in and its source code are freely available upon request to the corresponding author.

4. Validation of the model

We built a model of the two WOOPA rooms under TRNSys. This case uses the same room geometry, occupancy, etc. Ceiling coverage by FHU was simulated using the plug in developed in this study. We used typical weather files from Lyon (FR) and not real weather data. Running the simulation for June, July and August months allows us to simulate the operative temperature in the rooms.

With 50% ceiling coverage, we obtain an average increase of 0.33°C of the operative temperature. With 70%, this increase is 0.50 °C. The value obtained in the case of 50% ceiling coverage agrees very well with the measurements. However, the value obtained in the case 70% ceiling coverage is under evaluated.

Even if these results are good, the assumptions made during measurements and model forbids making additional conclusions. To precisely validate the model presented here as well to validate measurement results, a new experimental campaign is on-going.

5. Conclusion

Installing Free Hanging Units (FHU) on the ceiling of a Thermally Activated Building (TABS) improves acoustic comfort but will modify thermal comfort. Literature review and our measurements show that the impact on black globe temperature depends on ceiling coverage ratio, FHU orientation and distance between walls and FHU. It may reach 1°C for 70% ceiling coverage ratio and 0,3°C for 50% ceiling coverage ratio . The impact of FHU on thermal comfort thus exists but it is limited.

To better evaluate the interest of FHU in TABS buildings, we propose here a thermal model of FHU. This model was implemented in a plug-in for the thermal simulation software TRNSys. A first experimental validation shows the interest of this model, particularly for medium ceiling coverages (50%). For higher ceiling coverage (70%), the model slightly underestimates the increase of operative temperature. However, new measurements in a lab are currently on-going to complete model validation.

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