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Evaluation of thermal comfort using combined CFD and experimentation study in a test room equipped with a cooling ceiling

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Abstract

This paper reports a full-scale experimental campaign and a computational fluid dynamics (CFD) study of a radiant cooling ceiling installed in a test room, under controlled conditions. This research aims to use the results obtained from the two studies to analyze the indoor thermal comfort using the predicted mean vote (PMV). During the whole experimental tests the indoor humidity was kept at a level where the condensation risk was minimized and no condensation was detected on the chilled surface of the ceiling. Detailed experimental measurements on the air temperature distribution, surface temperature and globe temperature were realized for different cases where the cooling ceiling temperature varied from 16.9° to 18.9°C. The boundary conditions necessary for the CFD study were obtained from the experimental data measurements. The results of the simulations were first validated with the data from the experiments and then the air velocity fields were investigated. It was found that in the ankle/feet zone the air velocity could pass 0.2 m/s but for the rest of the zones it took values less than 0.1 m/s. The obtained experimental results for different chilled ceiling temperature showed that with a cooling ceiling the vertical temperature gradient is less than 1°C/m, which corresponds to the standards recommendations. A comparison between globe temperature and the indoor air temperature showed a maximum difference of 0.8°C being noticed. This paper also presents the radiosity method that it was used to calculate the mean radiant temperature for different positions along different axes. The method was based on the calculation of the view factors and on the surface temperatures obtained from the experiments. PMV plots showed that the thermal comfort is achieved and is uniformly distributed within the test room.

Keywords: cooling ceiling, **full-scale** experimental study, CFD simulations, thermal comfort

1. Introduction

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The majority of air-conditioning devices function using the principle of pulsed air, where the hot air of the room is partly recycled, cooled and returned into the room. The increase of the thermal loads in the buildings, mainly due to the arrival of office computers and lighting requirements, causes the installation of air-conditioning systems necessary to neutralize these loads and to create a good indoor thermal comfort. Heating, ventilating and air conditioning (HVAC) systems, which consume large quantities of energy, have become a necessity for almost all the buildings [1] to provide a comfortable indoor environment.

Currently, the evacuation of these quantities of latent and sensible heats is done mainly with air treated, introduced by air diffusers. To maintain comfort under these conditions, a greater volume of cooled air must be provided to the working area. Many complaints about air-conditioning systems have been claimed, especially in summer by female occupants [2]. Disadvantages such as noise, cold-drafts, air temperature differences between the human head and foot or energy wasting in certain cases, demand the investigation of other cooling systems which can create better indoor conditions and in the same time to be energy efficiently.

The use of water to cool the surfaces of the buildings is consequently a tempting alternative solution. It is even more appealing as water cooling requires much lower flow rates and higher temperatures of the water, so an energy consumption reduction could be achieved [3,4]. The chilled ceiling radiant panels are room cooling systems for placement in the ceiling zone, their cooling surfaces being connected with closed circuit heat conducting pipework containing flowing chilled water.

The main difference between cooling ceilings and air-conditioning systems is the mechanism of heat transfer. Classic air-conditioning systems are based only on the convection, while cooled ceilings employ a combination of radiation and convection. With cold ceilings, the transfer of radiant heat occurs by a clear emission of electromagnetic waves from the hot occupants and their environments to the radiant ceiling. The hot air arriving in contact with the cooled surface is cooled below the average temperature of the room and therefore falls down the zone of occupation [5]. With a cooling ceiling the temperature of water or its flow rate can be modified so that the surface temperature adapt to the desired conditions.

Ceiling radiant cooling systems can be more comfortable than conventional air cooling systems due to small vertical temperature gradient, few air movements and reduced local discomfort for occupants during long stays in cooled room environments. Catalina et al. [6] found after a series of simulations that the cooling ceilings offer good thermal comfort, the mean radiant temperature being an important parameter in the comfort evaluation.

Nagano and Mochida [7] analyzed the thermal comfort of five subjects in an experimental test room equipped with cooling ceilings. Their study aim was to clarify the control conditions of ceiling radiant cooling systems for human subjects in a supine position. They concluded that the mean radiant temperature for a supine human body should be used in the design of ceiling radiant cooling. It was also found that some of the subjects voted the strong radiant sensation even though the temperature difference between the ceiling and the room air was less than 5°C in these experiments.

Kulpmann [8] reported an investigation of thermal comfort in a test room equipped with a cooled ceiling surface and supplied with upward displacement ventilation air. The results showed that a good value of thermal comfort was obtained and that the temperature of the room surfaces was lower or at least equal to the air temperature in the room.

Hodder et al. [9] investigated and determined thermal comfort design conditions for combined chilled ceilings/displacement ventilation environments. A typical office room was reproduced in a laboratory test room in which the ceiling temperatures were varied. It was found that vertical radiant temperature asymmetry has an insignificant effect on the overall thermal comfort of the eight subjects that participated in this experimentation. Hodder et al. [8] reported that the existing guidance [10] is valid with no modification, for thermal comfort design in such design environments.

Imanari et al. [2] compared the radiant cooling systems with air-conditioning systems in terms of thermal comfort, energy consumption and costs. The thermal environment as well as human response was tested by using a small meeting room equipped with radiant ceiling panels. It was shown in their work that the radiant ceiling panel system is capable of creating smaller vertical variation of air temperature and a more comfortable environment than conventional systems.

Simmonds [11] found that the traditional design criteria such as dry-bulb temperature and operative temperature were not always sufficient, mean radiant temperature having an important influence on the comfort results.

Kitagawa et al. [12] investigated the effect of humidity and small air movements on the thermal comfort of subjects in a climatic chamber equipped with radiant cooling panels. The results of their questionnaires showed that the most comfortable sensation vote was obtained in the conditions whose thermal sensation vote was not neutral but approximately -0.5.

The thermal comfort is, by definition, a subjective sensation; different people will express different preferences and it is preferable to use real persons to evaluate this notion. However, this estimation method requires the use of a representative large number of subjects and hence can become expensive and time consuming. The approach used in this article study is based on experimental data and computational fluid dynamics simulations in order to obtain results on the parameters that influence the thermal comfort.

Myhren et al. [13] used Computational Fluid Dynamics (CFD) simulations to investigate possible cold draught problems, the differences in vertical temperature gradients and air speed levels in two office rooms equipped with different space heating and ventilation systems. CFD technique was also used by Rohdin et al. [14] research study on the prediction of indoor environment in large and complex industrial premises. Stamou et al. [15] have studied the thermal comfort in an Olympic Arena by using a CFD model. Calculated mean velocities and temperatures were used to determine the thermal comfort indices, predicted mean vote (PMV) and predicted percentage of dissatisfied person (PPD) and to evaluate the thermal conditions in the various regions of the Arena. CFD simulations were also found to be a good solution to determine the indoor thermal comfort in different research studies [16, 17] by using CFD simulations of virtual human manikins.

For the present research article, CFD simulations were realized with the aim to analyze the air flow pattern and the velocity fields inside the test room. The thermal comfort was evaluated by using the PMV index [18] that was calculated based using the data obtained from the experimental and CFD analysis. For the thermal comfort study, experimental results on the air temperature, mean radiant temperature and relative humidity were used, while the CFD study gave valuable results on the air velocity and air pattern. The particularities of this research study are related to a real-size controlled experimental set-up of using radiant ceiling panels equipped with

capillary tubes. Another particularity of this article was the use of CFD simulations to examine the air flow pattern and to obtain conclusions on the eventual local discomfort that could appear under a cooling ceiling.

2. Experimentations

2.1. Test room presentation

Studying the influence of certain indoor parameters on the thermal comfort when using a cooling ceiling could be a delicate issue, an experimental investigation being mandatory. The experimental cell MINIBAT (INSA de Lyon, France), presented in Figure 1, is composed of two identical enclosures (Cell 1 and Cell 2), which dimensions are 3.10, 3.10, 2.50 m according to the coordinate directions (x, y, z). A single glazed façade isolates the test cell 1 from a climatic chamber where the air temperature is controlled by the means of an air-treatment system. The climatic chamber temperature can vary between -10°C and 40°C as a function of time. Using a battery of 12 spotlights, of 1000 W each, it is possible to simulate an artificial sunning for the test Cell 1. The spectrum of the gas-discharge lamps with metal halide is similar to the sun one. For this study, only in the test cell 2 were installed the radiant ceiling panels and were taken measurements. The cooling ceiling surface temperature was varied during the experiments, but the rest of the five exterior walls of the MINIBAT were kept at a constant temperature of 26°C, by the mean of a thermal guard controlled with a precision of $\pm 0.5^\circ\text{C}$. The air distribution in the thermal guard was made by a ventilation network equipped with lateral diffusers and placed in the upper part of the controlled volumes. This distribution was chosen to avoid dead zones and to create a uniform air flow in the technical spaces. In Table 1 are presented the main elements of the MINIBAT envelope and in Table 2 are summarized the physical characteristics of the materials. For the data acquisition it was used a multiplexer-multimeter with more than 100 connections. The data acquisition central was linked using the software Labview with a computer, located outside the test facility. The time step chose for the measurement was 45 seconds and the total duration of the experiments was extended on several days.

2.2. Radiant cooling system description

In the test cell 2 were installed 6 radiant cooling panels (1.5 x 0.9 m – dimension for one panel), covering the entire ceiling area. The particularities of the installed system are the capillary tubes which are in contact with the interior part of the panels. With an exterior diameter of 3 mm and spaced between them by 1.5 cm, the capillary tubes offer good temperature uniformity along the panels' surfaces by comparison to similar systems.

The very small thickness of the heat exchangers capillary pipes allows them to be embedded into the surface of the walls, ceilings or floors. Passive elements of construction are thus transformed into heating and/or cooling surfaces by the use of the energy transferred between two radiant bodies. Each mat element was connected in parallel to the overall cold water collector in order to induce a homogeneous distribution in the ceiling's surface temperature. The water flowed through the mat at a temperature between 17°C and 20°C, lower values would have increased the risk of condensation. Such temperatures would not lead to condensation of moisture from air, but could offer opportunity for substantial energy savings since water at such temperatures could be provided without using refrigeration or active cooling. For our experiment it was used an air-water heat pump to obtain the desired water temperature for the cooling ceiling, along with a pipework system and a water pump.

2.3. Measurements

The test cell indoor air temperature was measured using 6 K-type thermocouples (Nickel-Aluminum) at different heights starting from 0.4 to 2.3 m and placed in different positions of the room. The test facility walls and floor have been equipped with K-type thermocouples in order to measure the internal/external surface temperatures; nine measurements were taken for each face of the room, a total of 108 surface temperatures being scrutinized. The radiant ceiling temperatures were measured using Pt100 probes in 9 different points. The measurements error for the thermocouples was of $\pm 0.5^{\circ}\text{C}$ and for the Pt100 probes of $\pm 0.3^{\circ}\text{C}$. The water temperature in the system was also measured at inlet/outlet position using Pt100 probes. The indoor humidity was measured using a sensor with a precision of $\pm 2.5\%$ for measurements of relative humidity between 10 and 90%. The black globe temperature was analyzed using a thermocouple positioned in a black globe with a diameter of 15 cm at a height of 1.2 m. The test cell was divided in four symmetric spaces, the analyzed measurement area being one of these parts (see Figure 2). The thermal guard temperature was kept at 26°C during the whole experimental campaign, but the ceiling temperature varied from 16.9°C to 18.9°C. The air temperature, relative humidity and globe temperature were measured in different locations of the rectangular grid of 0.43 x 0.43 m presented in Figure 2. The measurements were taken also on different heights, a total number of 96 measurements (on x, y, z-axis) being realized. The experiments were conducted under natural convection conditions and no humidity sources were added. All of the experiments were carried out under steady-state conditions.

2.4. Experimental results

2.4.1. Vertical air temperature profile

Three experimental cases were carried out to investigate the effect of radiant cooling ceiling temperature (T_{RCC}) on the indoor environment. Figure 3 shows the vertical distribution of air temperature in the center of the test room. If the temperature difference between the head and the feet is sufficiently large, a person can have local warm discomfort and/or cold discomfort even though the overall average is thermally neutral. Nominal vertical air temperature gradients between 0.4 and 2.1 m heights were found to be around 0.71-0.77°C/m for the cooling ceiling at temperatures of 16.9°C, 17.6° and 18.9°C. The obtained experimental data are more than satisfied, in terms when the comfort standards call for a maximum temperature difference between head and feet of 3°C, regardless of the operative temperature.

2.4.2. Surface temperature

The MINIBAT test cell was equipped with 108 thermocouples to measure the surface temperature on the interior and exterior part of the room. It wasn't notice any major wall temperature fluctuation due to air treatment system of the thermal guard. The impact of surface's temperature on the human thermal comfort is considerable and a particular attention have be paid to this aspect. A 0.5°C difference was observed between the lower and the upper part of the walls (see Figure 4).

2.4.3. Mean radiant temperature

Even though they are not in direct contact with the body, cold surfaces (like radiant cooling ceiling) still greatly affect human perception of temperature. With radiant cooling, the energy balance on the human body is different than without the cooled ceiling in several ways. First, the heat losses from the human body by radiation are increased from about 35% without radiant cooling to 50% with radiant cooling [8]. Likewise, the heat losses due to convection decrease from about 40% without the chilled ceiling to 30% with radiant cooling.

The globe temperature, which integrates the effect of air temperature, radiation and air movement, was measured at $z=1.2$ m for different locations of the (x, y) plane. It was found that the globe temperature had lower values than the air temperature with 0.6°C to 0.9°C (see Figure 5).

Mean radiant temperature (MRT) is the most important parameter governing human energy balance, with the strongest influence on thermo-physiological comfort indexes such as PMV, its meaningful scope being the thermal comfort. To evaluate the MRT for all the axes and heights we used a radiosity method. Considering a spherical point located at a spatial position in a room, the radiative balance equation can be written for this point.

Taking into account the reciprocal relation of the view factor, it is obtained, by using a linearization of the radiative exchanges between the spherical point and the walls and neglecting the internal reflections, the thermal balance of the point [19]:

$$MRT = \sqrt[4]{\sum_{j=1}^N F_{0j} T_{Sj}^4} \quad (1)$$

We suppose to have only rectangular surfaces for the different envelope components to consider. F_{0j} , the diffuse view factor between a spherical point and a rectangular surface can be calculated with the Eq.4 according to the notations of Figure 6. In the Eq.3 the surface temperatures (T_{Sj}) temperature were obtained from the experimental campaigns.

$$F_{oJ} = \frac{1}{4\pi} \left(\arctg \left[\frac{ab}{c \cdot \sqrt{a^2 + b^2 + c^2}} \right] \right) \quad (2)$$

A comparison between the measurements taken on the mean radiant temperature and the results obtained using the radiosity method is presented in Figure 7. The results correspond to the position (x_1, y_1, z_3) to (x_4, y_1, z_3) as presented previously in Section 2.3. The results obtained using the radiosity model were in good agreement with the data obtained from measurements, a maximum difference of 0.4°C being noticed, value which can be explained by the measurements errors from the thermocouple sensors. The results of MRT calculations are presented as two temperature fields at position x_1, y_1 and x_4, y_1 on the y-axis. The analysed cases correspond to the position in the center of the test room and in the proximity of the wall. Figure 8 shows the mean radiant temperature distribution calculated using the Eq.3 on the two analysed cases. The effect of the radiant cooling ceiling on the radiant temperature is more noticeable for the x_1, y_1 position and is reduced when approaching the walls, due to radiant effect of the walls temperature.

2.4.4. Indoor humidity levels

One of main inconvenient of the radiant cooling ceiling is that it's presenting a relative high risk of condensation on the chilled area. Even if the indoor humidity has a small effect on the global thermal comfort, for the cooling ceiling systems it's a crucial design parameter. Usually the dew point temperature is calculated for each zone by monitoring all the time the ambient air temperature and the relative humidity. The inlet water temperature is

controlled individually with values higher than the dew point of the ambient air. Therefore, if a risk of condensation is present, the water temperature is raised. A good solution to reduce the risk of condensation is to dehumidify the air but this solution will increase considerably the price of the system and it cannot be used alone, a control system being necessary for perfect safety against the condensation. For our experiment the relative humidity was kept at values between 42% and 43% (see Figure 9), so the risk of condensation on the ceiling was minimized.

3. Mathematical model of airflow

During the last years, Computational Fluid Dynamics (CFD) technology has progressed and made it possible to analyze complex situations where HVAC systems were simulated and compared with real data obtained from experiments [20].

The mean air flow modelling is divided in two parts: a) mathematical model (governing equations, turbulence model, boundary conditions, et.) and b) numerical solution (discretisation, numerical scheme, solution convergence, etc.)

The CFD model was based on the description of the experimental test room. The airflow pattern and temperature distribution in the facility are governed by the conservation laws of mass, momentum and energy. The flow is assumed to be steady state, three-dimensional, incompressible and turbulent. The buoyancy effect is invoked in the momentum equation, k and ϵ . The Boussinesq approximation hypothesis is used for the buoyant force term. The radiation heat transfer is not integrated in the CFD model due to the constant temperature of surrounding surfaces. The numerical model is based on commercial CFD code called STAR-CCM+ [21]. The Rayleigh-number has been calculated and its value of around 10^{10} showed that the airflow is turbulent.

3.1 Turbulence modelling

For this study the turbulence is modelled with the AKN Low-Re $K-\epsilon$ model [22]. This turbulence model was evaluated numerically stable with correct precision on the results [23]. Hashimoto Y. [23] indicated that the standard $k-\epsilon$ turbulence model could be applied for a fully turbulent flow but it is inappropriate for a buoyant flow. In his research article he studied the airflow in an office room with displacement ventilation system using a low-Re number $k-\epsilon$ turbulence model with a damping function to reduce the turbulent viscosity near a wall (AKN model).

The AKN model introduced Kolmogorov velocity scale, instead of the friction velocity, to account for the near-wall and low-Reynolds number effects in both attached and detached flows. The fluid is considered like an incompressible one with density computed via the ideal gas law considered as varying only with temperature.

The model used in this study can be written in a general form as:

$$\rho \frac{\partial \bar{\phi}}{\partial t} + \rho u_j \frac{\partial \bar{\phi}}{\partial x_j} - \frac{\partial}{\partial x_j} \left[\Gamma_\phi \frac{\partial \bar{\phi}}{\partial x_j} \right] = S_\phi \quad (3)$$

where ϕ is the variable, Γ_ϕ the effective coefficient and S_ϕ the source term of the equation. The expressions of the variables for the turbulence model used in this paper are summarized Table 3.

3.2 Boundary conditions

The CFD model boundary conditions were taken from the experimental data results of the walls', cooling ceiling and floor surface temperature. Table 4 resumes the surface boundary conditions used for the CFD simulations based on the three experimental series where the cooling ceiling temperature was varied from 16.9°C to 18.9°C. The wall boundaries have been modelled using the no-slip condition with constant wall temperature. The low-y+ wall treatment was applied for our model. The wall treatment assumes that the viscous sublayer is well resolved and thus wall laws are not needed.

3.3 Discretization

The mesh is designed using the pre-processor STAR-DESIGN [21] and the discretization of the computational domain is achieved by means of an unstructured mesh. In the boundary layer next to a non-slip wall, there are high gradients within a small region, so to capture these gradients accurately it was necessary to have fine mesh spacing normal to the wall. The volume mesh process consisted of several steps, including surface improvement, subsurface generation and then the actual interior volume meshing. The grid contains polyhedral elements, the final mesh being composed of 116,762 cells. The volume range was between 3.386E-6 to 1.5351E-3 and the minimum distance between centroids of neighbour cells was of 3.374e-3. A grid dependency analysis was conducted to ensure that the resolution of the mesh was not influencing the results.

3.4 Numerical scheme

The solution method is based on the following main hypothesis: the diffusion terms are second-order central-differenced, and the second-order upwind scheme for convective terms is used to reduce the numerical diffusion.

The velocity-pressure coupling method is the SIMPLE algorithm. The multigrid scheme allows to accelerate the convergence as our model contains a very large number of control volumes [24].

3.5 CFD simulations results

3.5.1 Grid independence analysis

Even if a convergent solution was obtained an additional step was necessary to achieve good accuracy. Chen et al.[25] recommends a verification of the model results by systematically refining the grid size realized generally by doubling the grid number and compare the two solutions. A grid independence test was performed and the data results showed that the mesh density chosen was sufficient for a grid independent solution of velocity fields, no noticeable error being observed. Figure 10 shows the velocity results difference between the two analysed grid mesh (1) 116,762 cells and (2) 233,524 cells.

3.5.2 Comparison of the experimental data to CFD simulations

The model validation is important to ensure adequate results and was a necessary step before analyzing the air flow pattern obtained from the CFD simulations. The validation was focused on the comparison of air temperature for the three experimental series. The simulated results were analyzed and compared with the experimental data on the position (x1, y1) along the z-axis according to the analyzed experimental configuration presented previously in the article.

The analyzed data points are the most relevant in terms of temperature distribution in the test room. In Figure 11 the air temperature distribution is confronted with the experiment data. The comparisons between the experimental and the numerical data confirm the validity of the numerical model used in the CFD simulation; the differences between the CFD and experimental results could be explained by the measurements errors of $\pm 0.5^{\circ}\text{C}$. Furthermore the accuracy of the results was analyzed by a sensitivity study of the air velocities values and air pattern flow to a change in the cooling ceiling temperature. This temperature was modified accordingly to the measurements errors of the Pt probes used in the experimental campaign ($\pm 0.3^{\circ}\text{C}$). The conclusions of this sensitivity analysis showed that the air flow pattern is not modified and the differences between the air velocities were insignificant.

3.5.3 Air velocity fields

The air speed could be an important parameter for the human thermal comfort; an increased air speed will aid the evaporation of sweat thus leading to a cooling effect, particularly if loose clothing is worn. However, if the air

velocity is too high it may cause discomfort and a sensation of draughtiness. The thermal comfort cannot be ameliorated by an increase of air velocity if the temperature or humidity is too high. Mean air speed should be less than 0.15 m/s in the occupied zone to prevent a feeling of discomfort for the most sensitive. The CFD technique was found a valuable technique to complement the experimental results, allowing a detail access to velocity fields on different planes. In Figure 12, the flow fields at the two analyzed position (x1, y1) and (x4, y1) on the yz-axis are plotted for the experimental study where the radiant cooling temperature was set at 17.6°C. The hot air heated by the floor rises due to Archimedes effect; once the ceiling is reached, the air is cooled and is then goes down. It can be noticed that the air speed is higher than 0.2 m/s in the feet and ankles zone and may cause problems for the occupants. Because the vertical walls temperatures are not strictly equal, the air flow is not symmetric in the test room. The results showed that the air velocity levels were still below the standards limits in most of the occupied zones.

5. Thermal comfort analysis

The use of standardized scales to gather ratings of thermal sensation from groups of individuals has permitted the researchers to identify the vital elements of the thermal environment and to evaluate the average response of the subjects. Thermal sensation ratings given by real subjects are recommended, but these studies are expensive and demand a large number of subjects in order to get correct results. For our study the predicted mean vote PMV and the predicted percentage of dissatisfied PPD [13] were used to evaluate the thermal comfort. Both indices have been used with success during the past decades and their valid range extends to our case of air conditioning with cooling ceiling panels.

The PMV model assumes a relation between optimal thermal conditions, using the steady state heat balance equation for the human body and thermal comfort ratings from panels of subjects. The PMV is related by a 7-point thermal sensation scale of a group of people exposed to a certain environment to the calculated result of the basic heat balance equation. The PMV range values are within -3 and +3, where the positive values stand for hot feeling and the negative for cold feeling, 0 being a neutral value.

The elements necessary to calculate the PMV are the air temperature, the mean radiant temperature, the air velocity, the relative humidity and two subjective parameters which are the human metabolism rate and the clothing insulation of the person. For this study these parameters were obtained by the mean of an experimental

study, by calculations and by a CFD analysis. Figure 13 shows how the data obtained from the different studies are connected with the thermal comfort analysis. The main purpose of the CFD simulation was to give important information on the air flow pattern and on the air velocity fields. The experimental study was necessary to obtain data on relative humidity and air temperature, but also to provide the boundary conditions mandatory for the CFD analysis. The mean radiant temperature was calculated by using a radiosity method and had its support on the results of surface temperatures provided by the experimental tests.

In Figure 14 the PMV distribution on the planes (x_1, y_1) and (x_4, y_1) on the yz -axis) are investigated. Figure 15 presents the PMV index when the radiant cooling ceiling is set to 17.6°C , for different metabolism rates and clothing insulations. The hypothesis for the analysis were that the metabolism (M) corresponds to an office work and its value was 1.2 met (70 W/m^2) and the clothing insulation (I_{cl}) of 0.58 clo ($0.089 \text{ m}^2\text{K/W}$). The effect of the radiant ceiling is more intense in the center part of the room where the mean radiant temperature is lower with an overall thermal comfort which is acceptable and with values between -0.2 and 0.2 . The two plots of PMV demonstrate that it can be obtain a uniform thermal comfort distribution in the room when using the chilled ceiling to cool the indoor air.

Based on the data obtained from the studies, the PMV/PPD were calculated for different heights and for different levels of metabolism rate and clothing insulation. Even for higher levels of I_{cl} the thermal comfort is still good, exceeding the comfort range described in ASHRAE 55 ($-0.5 \div 0.5$) [26] only in the case where the metabolism rate takes higher values than 1.3 met .

6. Conclusions

The air cooling using chilled ceiling panels was found to be an interesting alternative to traditional air-conditioning system. The research work was based on a fully experimental study which had the support of an extended analysis on the air flow pattern by using the CFD technique. The boundary conditions necessary for the CFD simulation were taken from the experimental data and then a validation on the air temperature was realized.

The air velocity fields obtained from the CFD study showed that a local discomfort at feet/ankle zone was observed but with good values of air velocity for the rest of the test room. The mean radiant temperature, a necessary element for the PMV calculations, was evaluated using a radiosity method by calculating the view

factors in each point of the analyzed planes and with the data from the experiment. The experimental results obtained for different chilled ceiling temperature showed that with a cooling ceiling the vertical temperature asymmetry is less than 1°C, value which corresponds to the standards recommendations. The globe temperature was measured and compared with the air temperature, a maximum difference of 0.8°C being observed.

The relative humidity was kept at normal values of 43% in order to avoid the condensation that could have been produced on the cold surface of the ceiling. The thermal comfort was studied using the PMV index scale. The elements necessary to calculate the PMV were taken from the two studies, CFD and experiment. The PMV distribution plots showed that thermal comfort is achieved and is uniformly distributed no matter the position in the room for the different planes in the test room. The results reported in this paper confirm that the cooling ceiling has advantages like low vertical air gradient and that the thermal comfort is obtained even for higher metabolism rates or clothing insulation.

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Nomenclature

d	distance to the wall [m]
F_{oj}	diffuse view factor
f_{μ}, f_2	damping functions
g	acceleration due to gravity [ms^{-2}]
k	turbulent kinetic energy [m^2s^{-2}]
I_{cl}	clothing insulation [clo]

Pr	Prandtl number
Re	Reynolds number
M	metabolism [met]
MRT	mean radiant temperature [$^{\circ}\text{C}$]
PMV	predicted mean vote
PPD	predicted percentage of dissatisfied [%]
T	mean temperature [$^{\circ}\text{C}$]
T_{sj}	surface temperature [$^{\circ}\text{C}$]
U	mean velocity magnitude [ms^{-1}]
u_i'	velocity fluctuation component [ms^{-1}]
x,y,z	coordinate [m]

Greek letters

β	coefficient of thermal expansion [K^{-1}]
μ	dynamic viscosity [Pas]
ν	kinematic viscosity [m^2s^{-1}]
ε	dissipation rate of k [m^2s^{-3}]
ρ	mass density [kgm^{-3}]
τ	turbulent time scale [s]