NUMERICAL ANALISYS OF HUMAN THERMAL COMFORT INSIDE OCCUPIED SPACES

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ABSTRACT

In the present work, human thermal comfort is investigated within the built environment. The analysis is based on two building thermal simulation models. In the first one - Nodal Network – the air condition (such as, air temperature and velocity) within each thermal zone is assumed uniform and therefore, all zone occupants are exposed to the same condition. In the second model – Nodal Network-CFD coupling – mean radiant temperature, air temperature and velocity variations are considered. This approach allows the evaluation of comfort conditions of all individuals within the same zone. Sensation of draught can also be computed from the distributed results. The application of the developed approach is demonstrated by test cases; a window type air-conditioning unit is employed to control the air temperature of a four-zone building in both cooling and heating seasons.

INTRODUCTION

Buildings are usually constructed with the purpose of reducing the impact of severe climate changes providing comfort and safety to the occupants. In many cases, however, the occupancy is far from being adequate, as man expresses satisfaction with the ambient in a small range of climate conditions. Therefore, artificial means of air conditioning are necessary to impart thermal comfort.

Several works were conducted in order to quantify and qualify the variables that affects thermal comfort (Fanger, 1970, Gagge et al., 1986, ASHRAE, 1997). They showed that comfort can be evaluated from three different classes of variables: ambient variables (mean radiant temperature, humidity, air temperature and velocity), physical activity and clothing.

The hypothesis of uniform indoor climate is usually employed on building design, assuming that all occupants are subjected to the same condition within a thermal zone. However, in some situations, significant changes of air properties take place in the occupied space, resulting in different perception of comfort at different location. Usually, this is only verified after the occupation of the building and operation of the plant system.

After the advent of numerical techniques, the building thermal behaviour, as well as the possible air property gradients within building zones, could be predicted (Clarke, 1985 and Chen and Jiang, 1992). Consequently, this type of tool allows the evaluation of thermal comfort in different points of the same enclosure. The analysis of the simulation results may suggest modifications on the building design and/or plant system, even before construction.

The present work aims the evaluation of thermal comfort based on numerical simulation results. The air condition is considered not only uniform (mixed air) but also distributed within an occupied space. A technique, developed by Negrão (1998), is employed on the evaluation. This technique combines models of different resolution and therefore complexity.

MATHEMATICAL MODEL

The building system is composed of several components (constructions, plant systems, lighting system, occupants, etc.), each one with its own thermal characteristics. Therefore, building modelling is a complex matter because of component interactions and system size. Additionally, the thermal behaviour of buildings is strongly dependent on the heat and mass transfer interaction with the exterior environment.

Air velocity and temperature distribution, which characterise the air flow and heat transfer within building, is determined by the solution of the conservation equations of mass, momentum and energy. The conservation equation to be solved for each variable is expressed as:

$$\frac{\partial}{\partial t}(\rho\phi)+\frac{\partial}{\partial x_j}(\rho U_j \phi)=\frac{\partial}{\partial x_j}\left(\Gamma_\phi \frac{\partial \phi}{\partial x_j}\right)+S_\phi \qquad (1)$$

where ϕ represents the variable to be determined (velocity components and temperature), Γ_ϕ is the diffusion coefficient, S_ϕ is the source term, ρ is the air density and U_j is the velocity component in the x_j direction. The terms of equation (1) can be found in Table 1.

Table 1 Governing equations.

Equation	Γ_ϕ	S_ϕ
Continuity	-	-
Moment.	μ_{ef}	$-\frac{\partial P}{\partial x_j}+\frac{\partial}{\partial x_j}\left(\mu_{ef}\frac{\partial U_i}{\partial x_j}\right)-\rho g_i$
Energy	Γ_T	$q'''/c_{p,a}$
Turbul. Energy	$\frac{\mu_{ef}}{\sigma_k}$	$G-C_D\rho\varepsilon-G_b$
Energy Dissipat.	$\frac{\mu_{ef}}{\sigma_\varepsilon}$	$C_1\frac{\varepsilon}{k}G-C_2\rho\frac{\varepsilon^2}{k}G_b$
$\Gamma_T=\frac{\mu}{Pr}+\frac{\mu_t}{\sigma_T}$; $\mu_{ef}=\mu_t+\mu$; $\rho=\rho(T)$ $\mu_t=C_\mu\rho\frac{k^2}{\varepsilon}$ $G_b=g\beta_T\frac{\mu_t}{\sigma_T}\frac{\partial T}{\partial x_i}$; $G=\mu_t\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\frac{\partial u_i}{\partial x_j}$ $u_i=U,V,W$ $C_D=1.0$; $C_1=1.44$; $C_2=1.92$; $C_3=1.44$ $x_i=x,y,z$ $\sigma_k=1.0$; $\sigma_\varepsilon=1.3$; $\sigma_T=0.9$; $C_\mu=0.09$ k - turbulent kinetic energy ε - Dissipation of turbulent kinetic energy		

The equations are discretized and solved by the finite volume method (Patankar, 1980). Two grids are considered: the first one (global model) is less resolved – the air is considered mixed (only one cell represents the room air condition and only the energy equation is solved) – and the second one (detailed model) is more refined – the air volume is divided in several cells (all conservation equations are solved for each cell). In the detailed method, turbulence – characteristic of indoor airflow – is considered by the k-ε model (Rodi, 1984).

The heat convection on internal and external surfaces, heat conduction through constructions, solar heat gain, radiation between internal surfaces, etc. are also modelled by the global method. In the more refined model, only the airflow within a zone is solved.

Two possibilities are considered: i) solution of the global domain and ii) coupling between the more and less resolved models. The coupling between domains takes place at internal surfaces of walls. This coupling allows evaluation of not only global (in some zones) but also distributed comfort (in other zones). The details about this coupling can be found in Negrão (1998).

THERMAL COMFORT

By applying heat balance to human body as a whole, Fanger (1970) has identified the following parameters which affects thermal comfort: air velocity, air temperature, moisture content and mean radiant temperature (environment factors), physical activity and clothing. Fanger correlated the body thermal load to an index, PMV (Predicted Mean Vote). This index reveals the thermal sensation of a group of individuals (cold, warm or neutrality) regarding an ambient condition. The PMV equation can be found in the literature (Fanger, 1970) and can be generically written as:

$$PMV=\left(0.303e^{-0.036M}+0.028\right)L \qquad (2)$$

where L is the body thermal load and can be expressed as L=L(M, I_{cl}, T_{mr}, v_a, T_a, w). M is the metabolic rate, I_{cl} is the thermal resistance of clothing, T_{mr} is the mean radiant temperature, v_a is the relative air velocity, T_a is the air temperature and w is the moisture content. Fanger has also derived a second index which quantify the percentage of dissatisfied people with the ambient, the PPD (Predicted Percentage of Dissatisfied). This index is obtained as a function of PMV:

$$PPD=100-95e^{-\left(0.03353PMV^4+0.2179PMV^2\right)} \qquad (3)$$

Mean Radiant Temperature

The mean radiant temperature (T_{mr}) is defined as the uniform temperature of a black enclosure that results in the same heat transfer by radiation as the real case (radiation between the person and the surrounding surfaces). T_{mr} is a function of person posture in the thermal zone.

The accurate determination of T_{mr} is however complicated because of the complex human anatomy. In the present work, however, the T_{mr} is evaluated for a standing mannequin, placed in several points within a thermal zone. This mannequin (Figure 1) is composed of seven cubic figures, which represents the different parts of the human body (head, trunk, legs and arms). The cubic shapes are sized (Table 2) according to the dimensions of a thin adult person. Its dimensions are obtained as a function of the mannequin height, H_M, and the dimension P_M is the only one chosen independently from the others. Additionally, these cubes can be rearranged to model other postures (e.g. sitting, lying, etc.).

For each visible face of the cube, a plane radiant temperature (T_{rp}) is determined as a function of view factors, F_{p-i}, (between face p and an internal surface i of a zone) and of surface temperatures (T_i):

$$T_{rp} = \left[\sum_{i=1}^{n} F_{p-i} T_i^4 \right]^{1/4} \qquad (4)$$

The view factors F_{p-i} are obtained from the combination of view factors of parallel or perpendicular rectangular planes. These expressions can be found in the literature (Clarke, 1985).

$h_{M,c}/H$	0.125
$h_{M,t}/H$	0.344
$h_{M,p}/H$	0.531
$l_{M,c}/H$	0.188
$l_{M,t}/H$	0.263
$l_{M,p}/H$	0.088
P_M	20 cm

Table 1 – Mannequin dimensions.

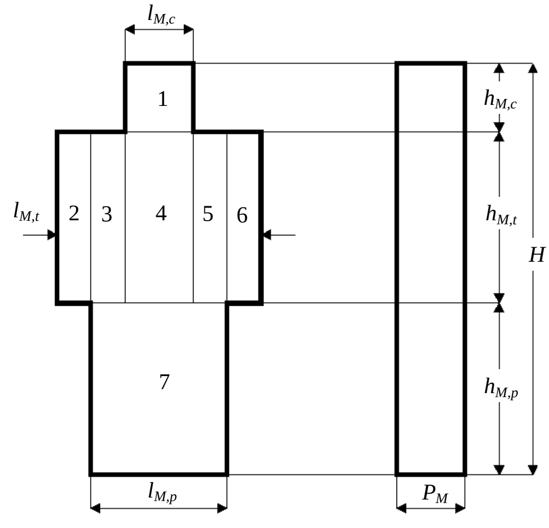

Figure 1 – Mannequin.

Once the plane radiant temperatures of all visible faces are known, the T_{mr} is then calculated as an average of the T_{rp} weighed by their respective areas:

$$T_{mr} = \left[\sum_{j=1}^{m} A_j T_{rp,j}^{\ 4} \Big/ A_T \right]^{1/4} \qquad (5)$$

where m is the number of the mannequin exposed areas (in this case, m=27). A_j corresponds to the face area and A_T is the mannequin total surface area. Carvalho F$^{\underline{o}}$ (1998) presents the determination of equations (4) and (5) in detail.

Risk of Draught

The thermal neutrality (PMV=0) is a necessary condition but not sufficient to establish comfort. Although the whole body thermal load can be zero, the individual may feel uncomfortable if one part of his body is cold and the other is warm. Fanger et al. (1988) established a complementary index, PD (Percentage of Dissatisfied), in order to quantify the risk of draught. This index is associated to local climate parameters and to turbulence intensity of the airflow. The authors emphasised that this risk exists particularly in sedentary and rarely in high intensity activity. The percentage of dissatisfied due to draught is obtained as a function of air temperature (T_a), mean air speed ($\bar{v}_a$) and the air turbulence intensity (T_u):

$$\begin{aligned} PD = 3.143(34 - T_a)(\bar{v}_a - 0{,}05)^{0.6223} + \\ 0.3696\,\bar{v}_a\, T_u\,(34 - T_a)(\bar{v}_a - 0{,}05)^{0.6223} \end{aligned} \qquad (6)$$

Turbulence intensity, T_u, is computed as function of turbulence energy and air velocity ($\sqrt{2/3\,k} / u_a \cdot 100\%$).

In a study of displacement ventilation system, Melikov et al. (1990) established that PD should be less than 15% for a comfortable environment. There are little information regarding other type of system, and this limit is usually accepted in such cases.

Evaluation of the Comfort Indices

The indices proposed by Fanger (PMV and PPD) are computed in several points in a horizontal plane within a zone. As the heat balance is considered for the body as a whole, PMV and PPD are based on mean values of air temperature and velocity. The averages are computed from values under 1.80m high.

RESULTS

The building under analysis, located in Florianópolis-SC (Latitude = 27^{o} South), is composed of three occupied zones and one attic (Figure 2). The air volume is considered mixed in all zones but zone 1. In zone 1, a window type air-conditioner is employed and the zone air temperature and velocity distributions are computed. The room depth, width and height are, respectively, 8.0m, 4.0m and 2.85m, as shown in Figure 3. The building walls are made of perforated bricks. In the north wall, there is a heat source under the window, which dissipate 500 W (the dissipation is by convection and radiation). The air-conditioner was originally located in the room west wall, 2.4 m from the floor and equidistant from north and south wall. The total cooling capacity of the equipment is 2635 W (10 000 BTU/h). The occupants perform sedentary activities (1met).

Simulations were performed for typical summer and winter days. For each day, two simulations were considered: a) global simulation and b) combined simulation in only one simulation time-step[1] in zone 1 (in the other zones, the air is assumed mixed). In the summer time, the combined approach was performed at 3pm and in the winter at 11am.

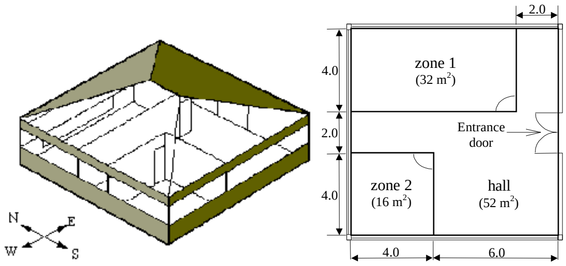

Figure 2 – Building geometry.

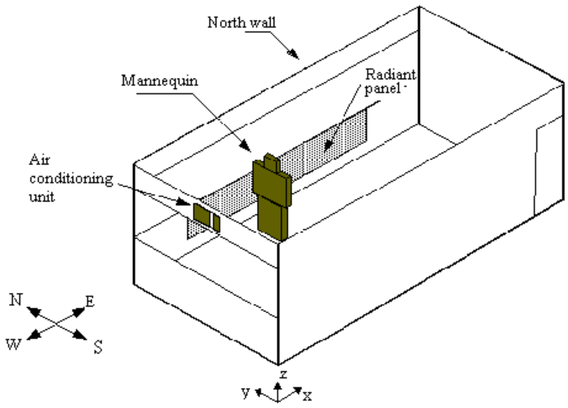

Figure 3 – Zone 1 geometry.

Additional information about problem configuration and climate data can be found in Carvalho Fº (1998).

Although the air-conditioner performance is dependent on the return air and outside conditions, the cooling capacity was considered constant during operation. The air flows perpendicular to the unit frontal plane, and therefore the louver deflexion angle is zero.

Summer Case

In this case, the occupants wear typical summer clothes (0.6 clo). The sensible cooling capacity of the air conditioning unit is 2000W (sensible heat factor of the cooling coil is equal to 0.76). During the room occupation period (8:00 to 18:00 h, the unit may cool and ventilate (on) or only ventilate (off) the ambient, since the cooling capacity is larger than the ambient cooling load.

The zone 1 return temperature is controlled within 23 to 25 ^{o}C. The simulation results of the global model (air completely mixed) can be observed in Figure 4. Note that, during the first hours of occupation, the unit operation time is shorter than the time it is not in operation. At about midday, the on and off periods are about the same order of magnitude. On the other hand, the opposite takes place during the afternoon; the operation time is much larger than the off period. The reason is that the cooling load increases during the day. In the morning, the cooling load is small and progressively increases until the maximum is reached at 4:00pm. This is also justified by the cooling capacity variation, seen in Figure 4.

Based on the simulation results, the comfort indices can be evaluated, as can be seen in Figure 5. Although the PMV fluctuate as fast as the air temperature, its average value increases during the day. At the beginning of operation, the average PMV is approximately –0.5 and close to 4:00pm, this value reaches +0.5. This change is related to the variation of the mean radiant temperature, as shown in Figure 5. Despite the PMV variation, the percentage of dissatisfied people (PPD) changes from 5 to 15%, which can be considered acceptable.

Fanger´s model was developed for steady state situations, and therefore the changes of PMV and PPD at the same frequency of the air temperature may not be representative. Thermal comfort under transient regime should be investigated in order to verify how adequate is the model on the treatment of such cases.

The mean air temperature within zone 1, observed during the combined simulation, is presented in Figure 6. The blue colour at middle of the room indicates the lower temperature at that region. The temperature variation within the zone is approximately 1^{o}C. The higher temperature takes place close to the west wall and the lower temperature at the region where the air jet falls.

The PMV distribution is shown in Figure 7. This distribution indicates the people located under the fall of the cold air jet may feel cold while those close to the heat source are slightly warmer. This variation is related not only to the air temperature but also to two other factors: i) Adjacent to the jet, people feel a relatively high air speed, ii) while close to the heat source, the effect of mean radiant temperature is more pronounced (see Figure 8).

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[1] A five seconds time-step was employed – above this value, the results are sensitive to the time-step.

Although the variation of PMV between –0.55 and +0.33 shows that 90% of people are satisfied with the condition, the same change (≅0.9) may indicate a dissatisfaction of 5% to 20% for a different clothing or physical activity.

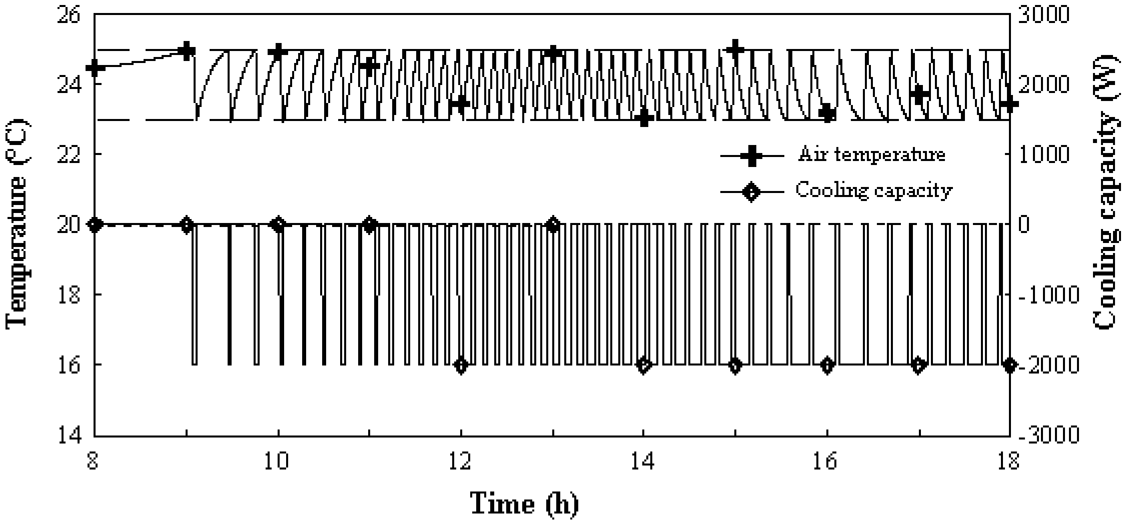

Figure 4 – Air temperature and cooling capacity for the summer case.

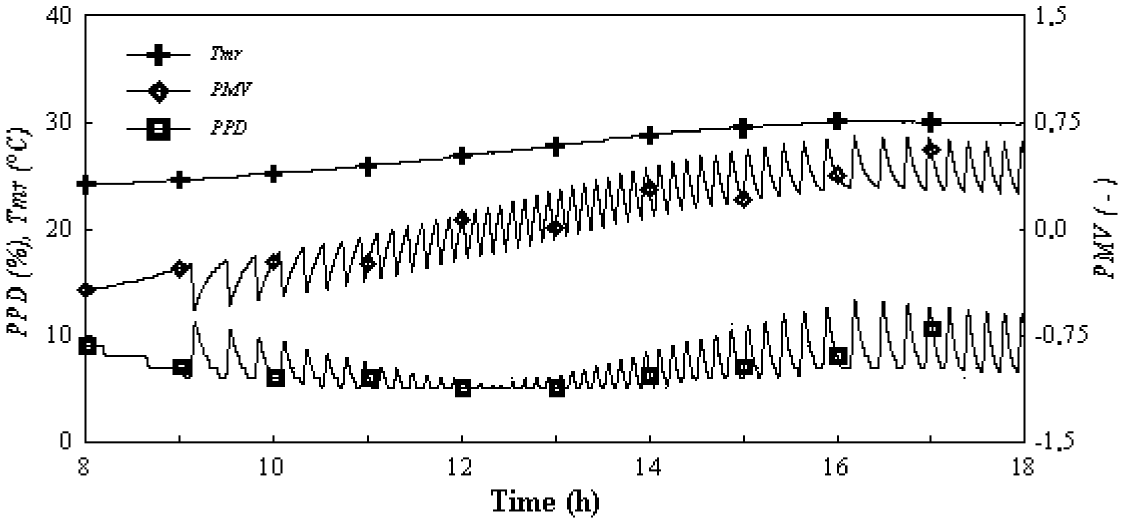

Figure 5 – Changes of mean radiant temperature, PMV and PPD for the summer case.

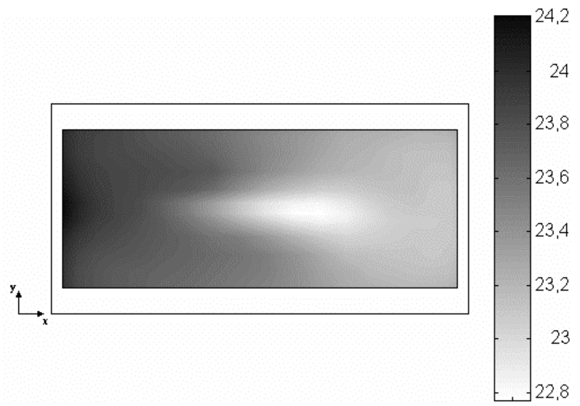

Figure 6 – Distributed mean air temperature (^oC) in zone 1 (summer).

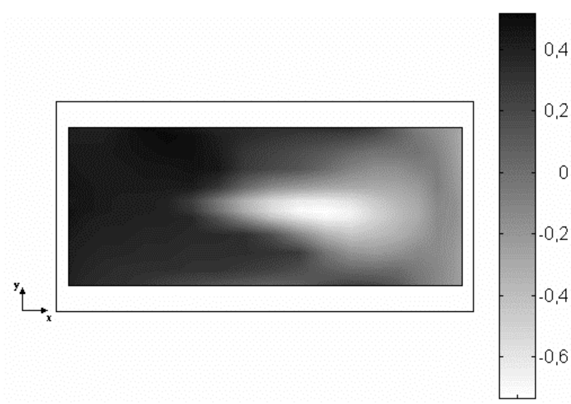

Figure 7 – PMV distribution in zone 1. Summer case.

Although the individual global discomfort is small in the air jet region, the percentage of dissatisfied due to draught is significantly high. Figure 9 shows the PD distribution inside zone 1 in the middle vertical plane. At the height of the head (1,75m), eighty to ninety percent of people under the air jet will be uncomfortable. The only region where the risk of draught is not significant is that between the west wall and the fall of the air jet.

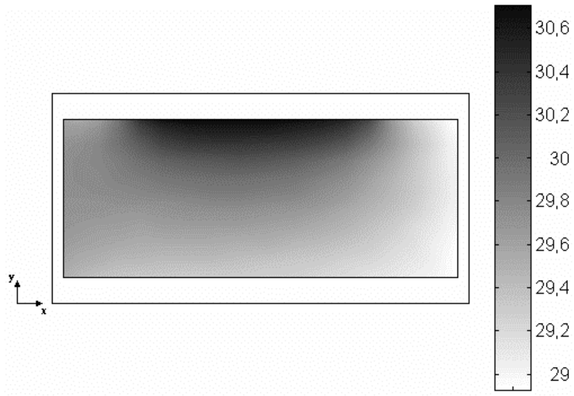

Figure 8 – Mean radiant temperature (^oC) distribution. Summer case.

Winter case

A typical winter day was also simulated and the mixed air temperature is shown in Figure 10. Unlike the summer situation, the maximum heating load takes place within the first operating hours of the unit, showing that the "on" and "off" periods are approximately the same. As the hours pass, the operating periods decreases until 1pm when the equipment does not work any more. Even though, the air temperature is kept within the control limit (23 to 25^oC). Different from summer, the "on" and "off" times are approximately the same. This indicates that the unit capacity is much higher than the heating load. In the winter, the equipment works as a heat pump with a heating capacity of 2500W.

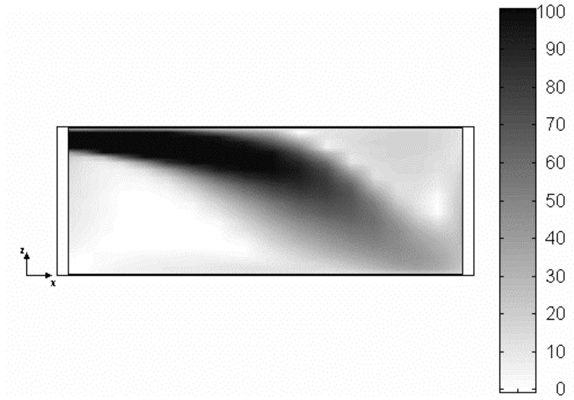

Figure 9 – PD distribution (%) in a vertical plane. Summer case.

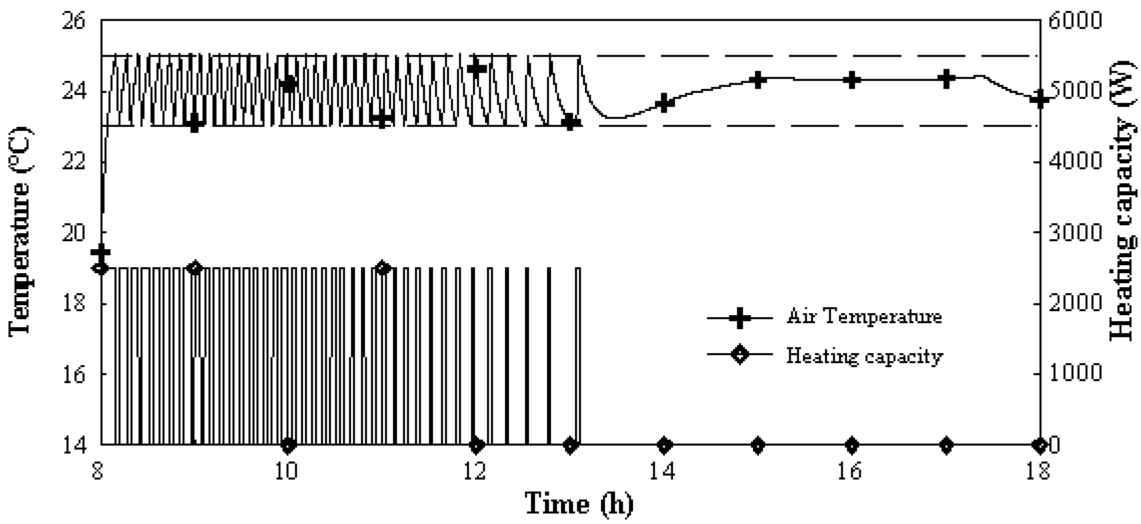

Figure 10 – Air temperature and heating capacity of zone 1. Winter case.

On the other hand, Figure 11 shows that the average thermal comfort changes within a wider range when compared to the summer case. The reason is that the internal surface temperatures are much lower in the morning, resulting in a lower mean radiant temperature during that period. For the clothing type (0.8 clo), the percentage of dissatisfied (people feeling cold) may reach 30%.

The air temperature distribution within zone 1 for the combined simulation is shown in Figure 12. Once more, the temperature variation is small inside the zone (maximum change is 1.3^oC). Unlike summer, mean radiant temperature varies 2.5^oC inside the zone, as can be seen in Figure 13. Figure 14 shows that the PMV changes from –0.31 to –0.73, indicating that the percentage of dissatisfied are within the limit of 8 to 18% (all feeling cold). Note that the larger number of dissatisfied are placed in the lower T_{mr} regions. However, the variation (ΔPMV=0,39) is not as large as in the summer case, showing much more uniform conditions. This means that an increase of clothing thermal resistance may keep the percentage of dissatisfied fewer than 5%. This small change of PMV is mainly related to the fact the mean air velocity changes little in the occupation area (0.07 to 0.15m/s).

Even in the heating season, the risk of draught is still possible, as can be seen in Figure 15. The distribution of PD in a horizontal plane (1,10m high) shows that close to the east wall, where the air jet falls, 23% of people expresses dissatisfaction with draught. In other location, the percentage of dissatisfied is kept under 15%.

DISCUSSION AND CONCLUSIONS

In the present work, an analysis of thermal comfort within occupied spaces was conducted based on numerical simulation of buildings. Two simulation models are considered: a global and a combined model (the global model coupled to a distributed one). The simulation was performed for typical winter and summer days.

In the global analysis, despite the air temperature being controlled between 23 to 25^{o}C, the level of comfort changes considerably during the occupation period. This change takes place because of mean radiant temperature variation during the day. The effect of the mean radiant temperature is more pronounced in the winter since the surface temperatures are much colder in the morning.

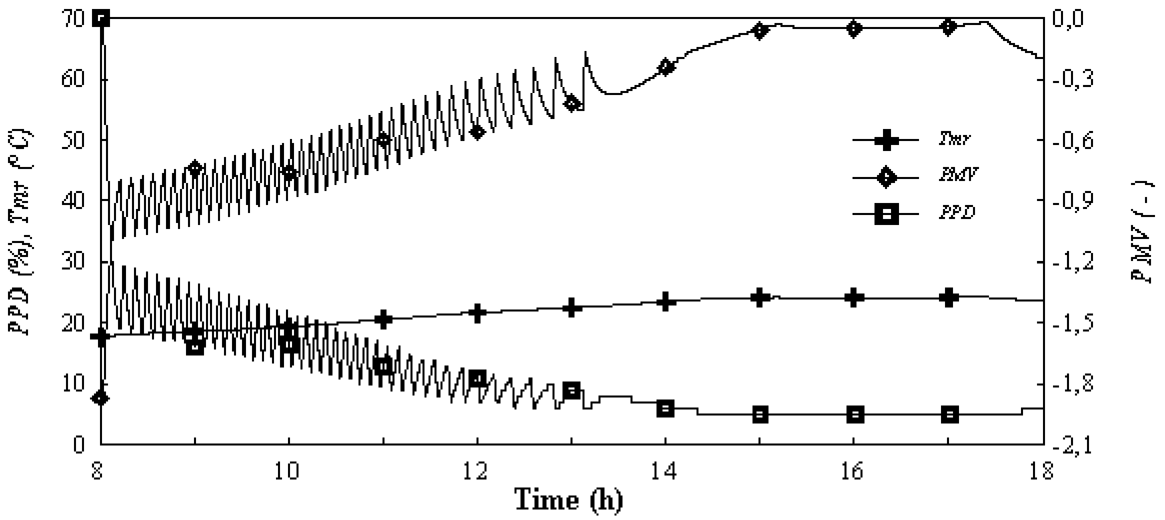

Figure 11 – Mean radiant temperature, PMV and PPD for winter.

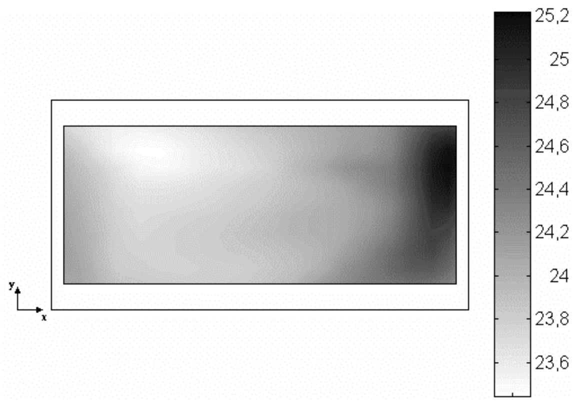

Figure 12 – Mean air temperature (^{o}C) distribution in zone 1 (winter).

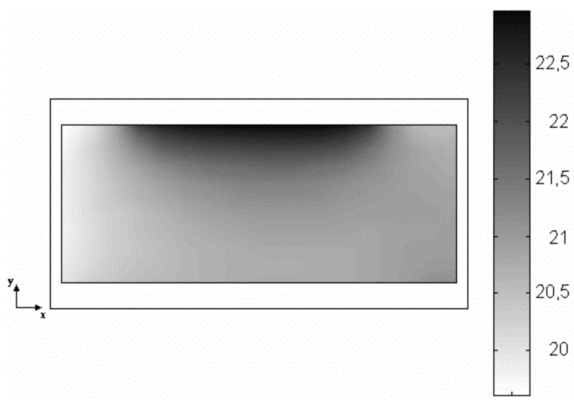

Figure 13 – Mean radiant temperature (^{o}C) distribution in zone 1 (winter).

The results show that the comfort indices oscillate as fast as the temperature of the air. The air temperature may vary between 23 to 25^{o}C during a period of 3 minutes, as in the winter period. Nevertheless, Fanger´s indices were developed for conditions of thermal equilibrium of the human body. Transient analysis is necessary to verify the influence of the oscillation of temperature in thermal comfort.

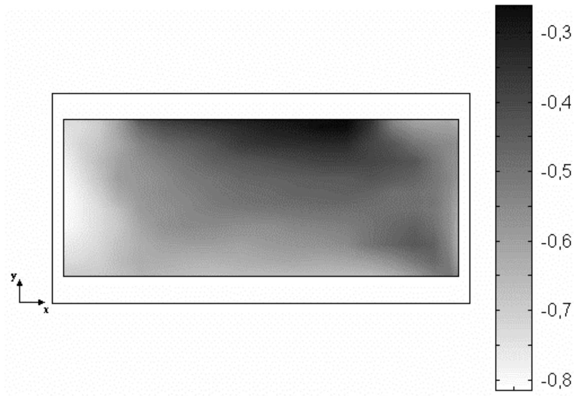

Figure 14 – Distribution of PMV in zone 1 (winter).

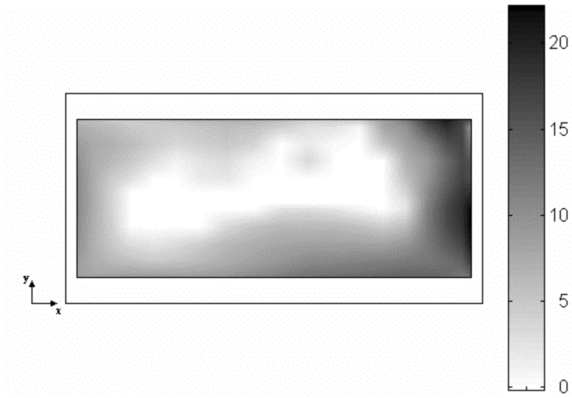

Figure 15 – PD distribution (%) in a horizontal plane. Winter case.

The winter situation presents a higher distribution of air velocity inside the zone than the summer case. This is justified by the fact that buoyancy effect produces a faster fall of the air jet in the summer. In the winter, the air is maintained in higher regions of the room until it is cooled off close to the ceiling. After that, the air falls. This fact decelerates the airflow resulting in a more uniform velocity distribution. On the other hand, the air temperature variation is higher in the winter than in the summer, just because of air stratification.

Although the air temperature variation is not so large (about 1.0°C in the summer and 1.3°C in the winter), the temperature distribution affects natural convection which consequently changes the velocity field. As seen, these changes have a strong influence in comfort.

The air temperatures of the other occupied zones were maintained artificially fixed. The indices of comfort and mean radiant temperatures for such zones, not shown because of space constrain, may be obtained from Carvalho Fº (1998).

In the summer, the risk of draught is quite high for those located under the fall of the air jet. Deflection of the inlet airflow can reduce this. Although this risk is still present in the winter, it is much reduced even because the air velocity is smaller.

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