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NUMERICAL ANALYSYS OF A SPLIT AIR-CONDITIONING SYSTEM USING TRANSIENT BOUNDARY CONDITIONS

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Abstract. *Energy consumption has been modified in recent years, especially because air-conditioning devices has become largely responsible for the increase in demand, particularly in tropical countries. This increase is due to the reduction in the air-conditioners price for homes, commercial and office facilities. An attenuation of this consumption issue would be to develop devices with the same refrigeration quality of the present, but that would consume much less energy. In addition, it is worth mentioning that, the most common devices used in residence, small shops and offices are Split Air Conditioning System, with a High wall internal unit. The evaporator control is based on the return temperature that implies in a waste of energy for not controlling areas of greater interest or even thermal discomfort in some regions. Considering this aspect, a numerical solution, using a CFD code, will be proposed using the energy-difference equations combined with the Navier-Stokes equations, considering a turbulent flow based in a semi-empirical k-ϵ turbulent model of two equations, to evaluate the behavior of the temperature profile in a standardized room. A numerical model was elaborated for analyzing the thermal behavior of the room with the swing mode. Analyzing the results, one can conclude the high performance of this mode when compared with operation using a fixed angle.*

Keywords: *Split ACS, Computational Fluid Dynamics, OpenFOAM^{TM}, Transient BCs*

1. INTRODUCTION

The popularization of air conditioning is nowadays the main responsible for the increase of the electrical energy demand in tropical countries. To illustrate that, in Brazil residential consumption represents approximately 37% of total electricity consumption. In this total amount, about 20% is due to air conditioning systems (Eletrobras, 2007). In general, according to Pessoa and Ghisi (2015), these systems represents up to 47% of final energy consumption in the country at certain times.

Therefore, there are relevant benefits in increasing the energy efficiency for air conditioning systems. Shah et al. (2013) estimates that up to 30% reduction in energy consumption could be achieved using new products using the technology already known by industry. Another aspect highlighted in this work is the improvement in the air distribution in the environment, which can determine how long the appliance will work at its maximum power.

An important tool to evaluate air distribution in environments is numerical modeling by CFD software.This tool can solve problems with for complex geometries and boundary conditions. Considering this aspect, it is possible to find several studies in literature. Among then, is was highlited a numerical study of an air-conditioning system with floor inflation based on the pressurized zone with air diffuser (POMA) proposed by Fang (2017). In this same theme, Mao et al. (2014) used a numerical simulation to solve a three-dimensional quarter with a fine mesh.

Other studies were also presented with analysis of the turbulence model that best represents the air circulation systems in numerical simulations. Yongson (2007) points out that a turbulent flow numerical study, such as those currently developed for air-conditioning systems, was impossible a few decades ago. To demonstrate this fact, in his work were modeled an individual office room with some furniture, comparing the models of turbulence k-epsilon and RNG. In this same context, Chen, Zhang e Zuo (2007) show an overview of the turbulent models applicability for closed-space modeling. Another work that evaluates this aspect was presented by Chen and Moshfegh (2011) where the standard k-ϵ models, k-ϵ RNG and Realizable k-ϵ models are compared; through an office simulation with the SIMPLE algorithm.

Considering the studies already developed, this paper will evaluate the thermal behavior of a standard room for different situations. The main changes involving the position and parameters of operation of Hi-wall evaporators. The results in each case will be proposed a physical model and simulated through OpenFOAM.

Based on the simulation results it is possible to estimate the thermal behavior in the different regions of the room according to the position and operating conditions of the Hi-wall evaporator. It will be studied an environment with three

Nomenclature

Acronyms

ACS	Air-Conditioning System
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
PISO	Pressure Implicit with Split Operator
RANS	Reynolds Averaged Navier-Stokes
SIMPLE	Semi-Implicit Method for Pressure Linked Equations

Greek Symbols

ϵ	turbulent kinectic energy dissipation [m^2/s^3]
μ	fluid viscosity [m^2/s]
Φ	viscosity dissipation [W/m^3]
ρ	fluid density [kg/m^3]
σ	turbulence model constants
θ	inflow angle [rad]

Roman Symbols

$\dot{q}$	heat generation [W/m^3]
$\mathbf{q}''$	heat transfer [W/m^2]
$\mathbf{u}$	fluid velocity vector [m/s]
C	turbulence model constants
g	gravity acceleration [m/s^2]
H	distance from the top [m]
h	enthalpy [J/kg]
k	turbulent kinectic energy [m^2/s^2]
L	distance from the side wall [m]
p	pressure [Pa]
t	time [s]

Subscripts

t	turbulent parameter

sitting people and considering the loss of heat through certain walls by the global coefficient method. This environment geometry is similar to several environments such as bedrooms, residential rooms, collective offices and classrooms.

2. METHODOLOGY

The development of proposed study was defined based on setting the temperature and an insufflation speed at the exit of the air conditioning system. The numerical solution for the flow was considered as turbulent and the air was treated as Newtonian fluid. The solution for air flow and temperature profile in the room is based on the discrete and simultaneous solution of the continuity, Navier Stokes and energy equations:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \qquad (1)$$

$$\rho \left(\frac{\partial \mathbf{u}}{\partial t} + \mathbf{u} \cdot \nabla \mathbf{u} \right) = -\nabla \mathbf{p} + \nabla \cdot \left[\mu \left(\nabla \mathbf{u} + (\nabla \mathbf{u})^T \right) \right] + \nabla \cdot \tau + \rho \mathbf{g} \qquad (2)$$

$$\frac{\partial (\rho \mathbf{h})}{\partial t} + \nabla \cdot (\rho \mathbf{h} \mathbf{u}) = \nabla \mathbf{q} + \Phi + \dot{q} \qquad (3)$$

The turbulence model adopted was the standard RANS / $k - \epsilon$. In this semi-empirical turbulence model, k is the turbulent kinetic energy defined as the variation of velocity fluctuations. On the other hand, ϵ is the dissipation rate of turbulent kinetic energy that is responsible for the disappearance of speed fluctuations. On the other hand, ϵ is the viscous turbulent dissipation of kinetic energy that is responsible for the disappearance of speed fluctuations. This model originally developed by Launder and Spalding (1974), introduces the concept of turbulent viscosity (μ_t) based on values for k and ϵ:

$$\mu_t = \rho C_\mu \frac{k^2}{\epsilon} \qquad (4)$$

Values for k and ϵ are obtained using a two equation method based on the set following equations:

$$\frac{\partial (\rho k)}{\partial t} + \nabla \cdot (\rho k u_i) \quad = \quad \nabla \cdot \left[\frac{\mu_t}{\sigma_k} \nabla k \right] + 2\mu_t S_{ij} S_{ij} - \rho \epsilon \qquad (5)$$

$$\frac{\partial (\rho \epsilon)}{\partial t} + \nabla \cdot (\rho \epsilon u_i) \quad = \quad \nabla \cdot \left[\frac{\mu_t}{\sigma_\epsilon} \nabla \epsilon \right] + C_{1\epsilon} \frac{\epsilon}{k} 2\mu_t S_{ij} S_{ij} - C_{2\epsilon} \rho \frac{\epsilon^2}{k} \qquad (6)$$

with the empirical constants proposed as:

$C_{1\epsilon}$	$C_{2\epsilon}$	C_μ	σ_k	σ_ϵ
1.44	1.92	1.44	1	1.3

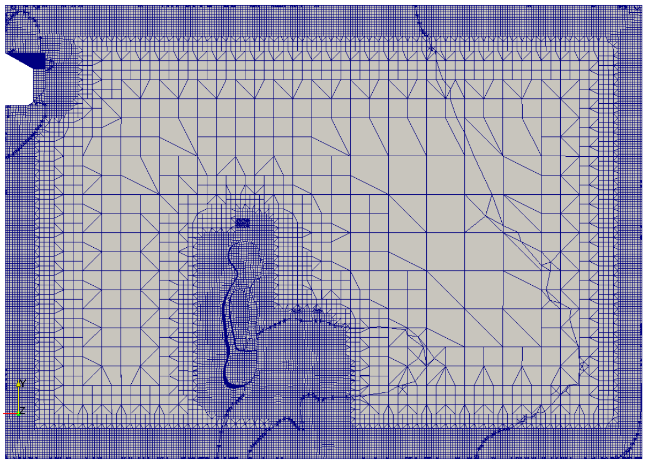

Figure 1. Mesh generated using the *snappyHexMesh*

This set of equations that solves the turbulent flow, previously described, is already implemented in the open source software, OpenFOAM™, including the necessary algorithms for the velocities and pressures field integration. In this study the SIMPLE pressure-velocity coupling algorithm was used, with the Boussinesq hypothesis to consider the fluid density variation.

In the light of these characteristics, the chosen solver for this situation was*buoyantSimpleFoam*. For non-permanent flow, there is another possibility using a modified PISO pressure coupling algorithm present in *buoyantPimpleFoam*, which was used for comparison purposes.

The mesh generator used was *snappyHexMesh* is also supplied with the OpenFOAM™ package. In this case, it is possible to use STL solids to generate an unstructured mesh from a previously established structured mesh as shown in Figure 1.

Based on this mesh generation scheme, it was possible to establish different mesh sizes in order to verify the solution consistency. Thus, four different meshes were analyzed for this verification. A mesh size and required computational time overview for the standard case convergence is shown in Table 1. Based on the obtained values analysis, the option was made by the mesh 4 that presents a shorter simulation time than the meshes 1 and 2, whereas for the mesh 3 there was no results convergence.

The relaxation factors used in *buoyantSimpleFoam* simulations for a good approximation of the transient solution are 0.7 for pressure, 0.6 for velocity and 0.8 for enthalpy. Using these values with the return temperature at evaporator, it was able to verified a similar behavior with the *buoyantPimpleFoam* solver. However, the use of SIMPLE algorithm implied in a computational time reduction to 145,000s in the Standard case in the range of 3600s. In addition to the relaxation factors, we also modified the time increment values to 0.2 s instead of 1 s, used in the mesh tests.

2.1 Boundary conditions

For the proposed problem analysis, the static and transient boundary conditions were evaluated and the one is used as reference data and called the standard. The boundary conditions used in this problem were established from a typical air conditioning systems conditions and can be seen in Figure 2.

It should also be noted that the air condition in the room is established from the velocity represented in this vector form. However, the set module value is equals to 2 m/s with variable angle θ, which is a typical condition observed in a Hi-wall evaporator of a mini-split with operation in swing mode. Figure 3 shows the behaviour of outlet angle for swing conditions. The time period of a complete cycle is 60 s and the limits for inclination angles are from 0° to 60°. Fo this

Table 1. Mesh size tests OpenFoam™

Mesh	Points	Runtime [s]
1	7720146	38480
2	3237318	25151
3	3250573	8508
4	4911055	13416

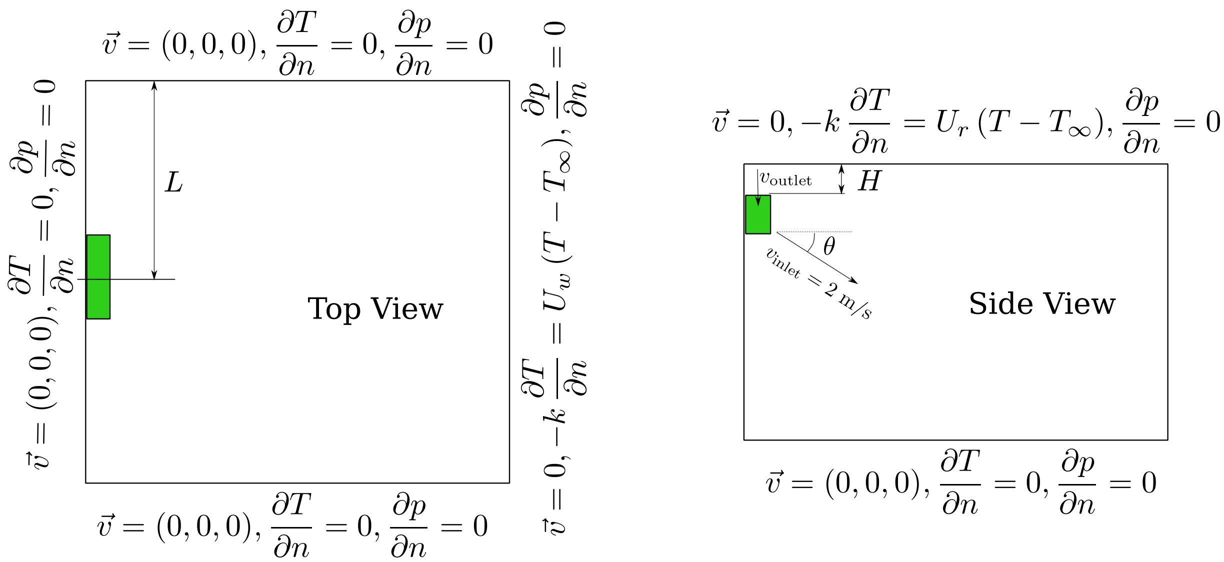

Figure 2. Boundary conditions used on the room limits.

case, the variation is continuos and when one cycle ends another one starts.

Likewise, the outlet temperature was set at 9°C (282K) for the case where it is desired to obtain a temperature room around 18 °C (291K). This procedure is common in offices and classrooms, to quickly achieve the thermal comfort condition. This temperature difference of 9°C is based on the values suggested by ASHRAE, which indicates in the summer a range of 6.7°C to 11.1°C.

3. RESULTS

Based on the considerations presented above, the behavior of an air conditioning system was analyzed in a square room measuring 4 m x 4 m with height of 2.9 m. The evaporator used is similar to hi-wall type, with front insufflation. Outlet angles are variable with time according with previous presented model. In this system the air return occurs on the top of the equipment. The device positioned at a height $H = 0.3$ m, of the rear wall, and at a distance $L = 2$ m of side wall.

Due variation in the outlet angle, the velocity in the region of interest varies significantly as shown in Figures 4 and 5. Considering Figure 4, it is noted that the time to achieve the comfort temperature 295K (22°C) is approximately 200s. Considering the temperature axis, there is a good uniformity throughout, with a significant reduction only in the direct

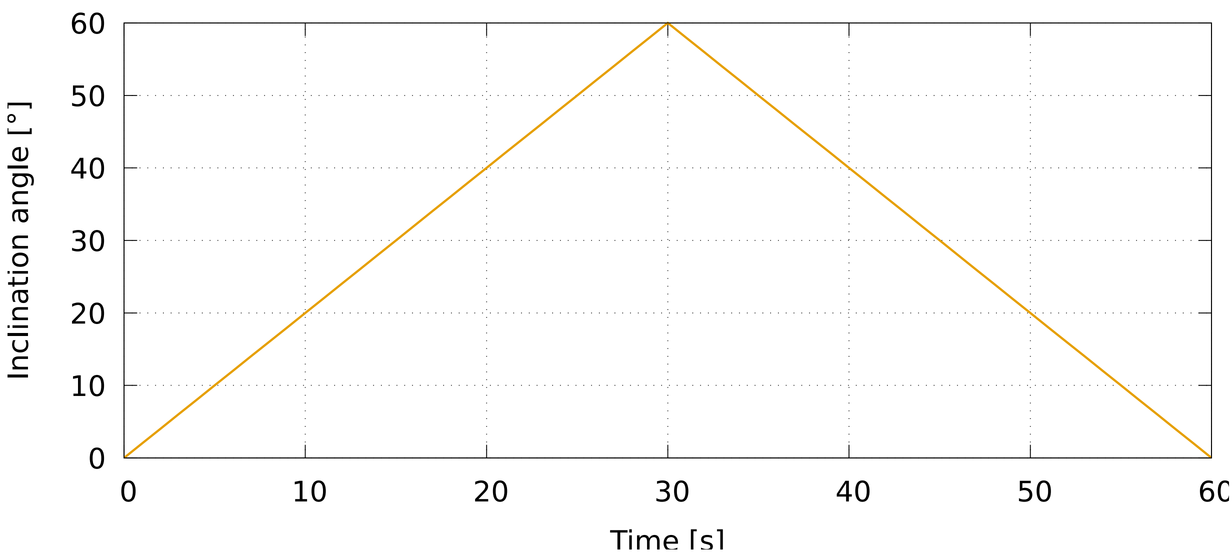

Figure 3. Outlet angle for ACS in swing mode as function of time.

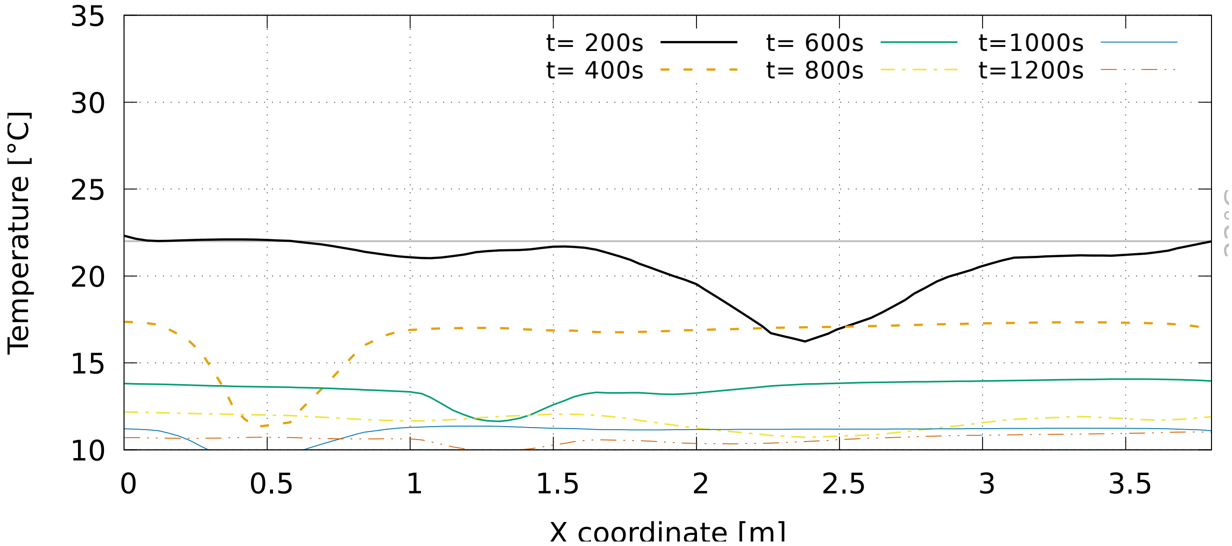

Figure 4. Temperature profile on a line over person's head (x direction).

axis of the airflow originating from the ACS.

There is also a homogeneity in the velocity distribution in the environment, according to Figure 5 and with a suitable speed for the axis considered, respecting the limits suggested by ASHRAE of 0.2 m / s. It should also be noted that the contact speeds will decrease with time, as the angle repeats. It is important to detach that after the thermal comfort is achivied, the flow from ACS could be reduced.

Figure 6 shows the temperature profile inside the room. For better visualization two cut planes were used for times of 200s and 400s. Analyzing the isotherms, one can clearly see the changes in temperature with the change in inclination of air conditioning deflector.

4. CONCLUSIONS

The results shown the consistent behavior of the boundary condition transient for represent test. It should be noted that, for the case of variable angle in air conditioning systems operating on swing mode, the variable boundary condition worked properly. The results show when the air conditioning system has the flow directed towards its proximities, the air flow speed achieves values major then recommended ones. A reduction in this value is possible by limiting the value of the insufflation angle to values less than 60°. In this case, however, new tests need to be performed. Another possibility is the reduction of the air flow rate when the thermal comfort condition is reached. Despite the success in implementing

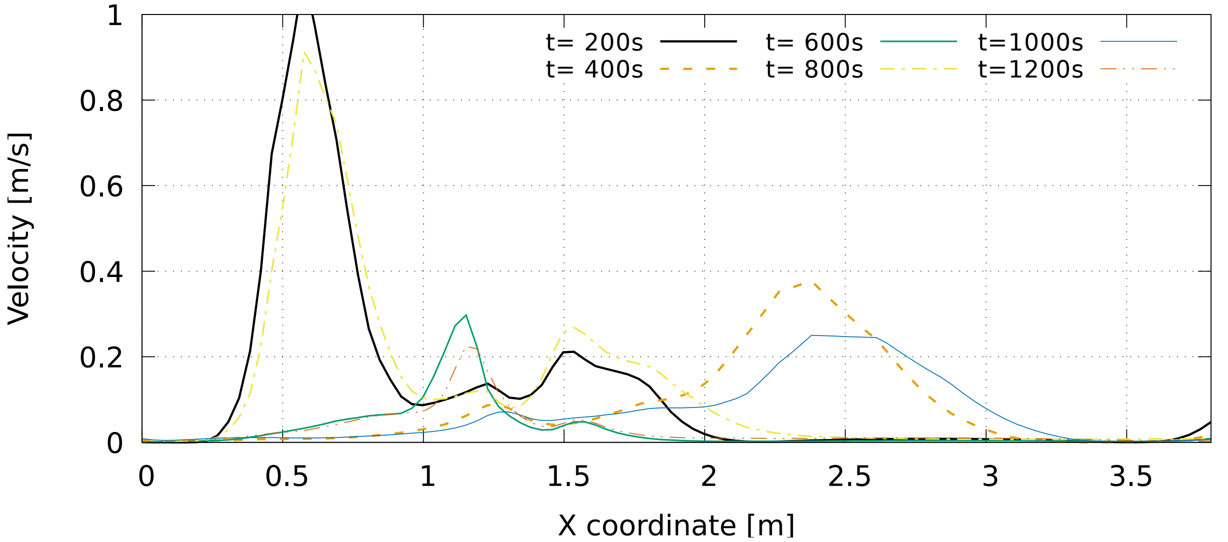

Figure 5. Velocity profile on a line over person's head (x direction).

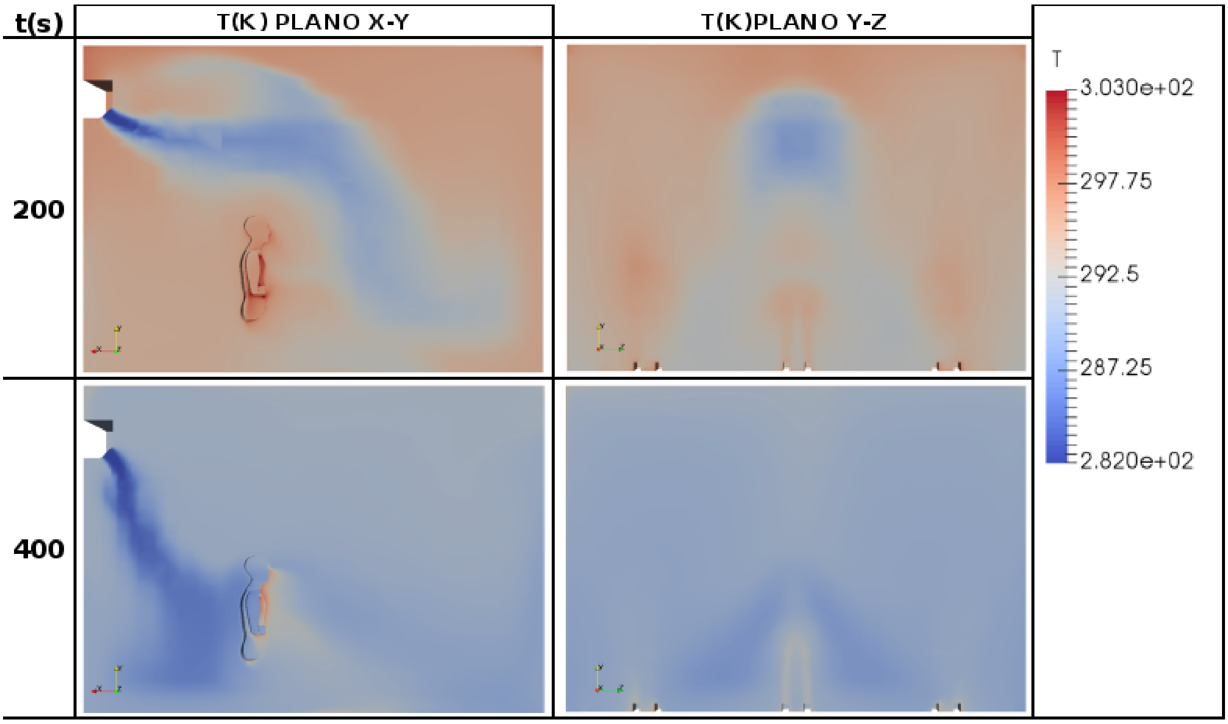

Figure 6. Temperature distribution at the room for different times and planes.

the boundary conditions variables in this case, new tests using some other kind of different transient boundary conditions also need be done.

5. ACKNOWLEDGEMENTS

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