THERMAL BEHAVIOR OF PREMISES EQUIPPED WITH DIFFERENT ALVEOLAR STRUCTURES

by

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This paper presents a numerical study of local thermal behavior. Vertical walls are equipped with alveolar structure and/or simple glazing in east, south, and west frontages. Local temperature is assumed to be variable with time or imposed at set point temperature. Results principally show that the simple glazing number has a sensitive effect on convection heat transfer and interior air temperature. They also show that the diode effect is more sensitive in winter. The effect of alveolar structure and simple glazing on the power heating in case with set point temperature is also brought out.

Key words: *solar energy, heat transfer, glazing-alveolar structure*

Introduction

Given the lack of conventional energy resources (coal, oil, gas, timber, *etc.*) and the various oil crises, countries are turning to the development of new renewable sources of energy. The building field is one of the energy consuming sectors on which we focused in creating a new thermal economical system to save energy.

In this context, we develop a numerical model to study a local thermal behavior. Each one of the vertical walls that are, respectively, pointed to the south, east, and west contains a simple glazing and/or an alveolar structure in a parallelogram arrangement which causes a variable thermal flux according to the flow direction.

Many researchers are interested in this kind of aniso-tropical thermal behavior of the alveolar structure. Among them are those involved in studying rectangular cavities. Their faces are tilted or not active against the gravity field; others are rather interested in inclined cavities.

Taking into account the aero-thermal phenomena in the alveolar structure, Guittierez [1] experimentally determined temperature and speed fields in the alveolar structure according to different angles of inclination and report of shape (*L/H*). He proposed correlations connecting Nusselt and Grashof numbers of $\mathrm{Nu} = a\mathrm{Gr}^b$ type for rectangular cavity ($\alpha = 0$) and insulating cavity ($\alpha < 0$). These correlations were established for ideal cases of thermal boundary conditions: perfectly conductive or insulated lamellas. Grashof number was considered varying between 10^4 and 10^6. These experimental results showed that the alveolar structure can on one hand reduce heat losses by anti-convective effect and on the other hand technical problems that pose insulation in the field of housing.

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Seki *et al.* [2] experimentally studied heat transfer by natural convection in a cavity of size ($H = 72$ mm, $L = 50$ mm) performed with fluid (air, distilled water, refined oil). The inclination angle takes the values: 0, ±0.25, ±45, ±60, and ±70 degree. The high and low passive walls (lamellas) are constituted by an insulating material. They proposed correlations connecting Nusselt, Prandtl, and Grashof numbers: $\mathrm{Nu} = 0.218F(\alpha)\mathrm{Pr}^{0.024}\mathrm{Ra}^{1/4}$ with $F(\alpha) = 1 + 0.546\alpha - 0.781\alpha^2$. This correlation is more accurate for small than for large angles when the fluid is air.

Bairi [3] experimentally studied the natural convection in the closed alveolar structure when the active walls remain vertical. He brought out correlations of Nusselt number type according to the Grashof number: $\mathrm{Nu} = F(\alpha)\mathrm{Gr}^n$, where n depends on the nature of the flow for different configurations by varying the angle of inclination in the cavities, the report of shape and the temperature difference (ΔT) between both warm and cold vertical walls. He presented in this study the evolution of the convective heat transfer coefficient at the wall for all studied cases, and correlations Nu-Gr for ($10^6 < \mathrm{Gr} < 10^9$) were brought out. He also showed the influence of the thermal boundary conditions at the level of the passive walls (lamellas) on the convective heat transfer.

Alkhoja [4] studied and compared numerically and experimentally the daily and annual thermal behavior of several types of walls exposed to the solar radiation and pointed southward, namely:

– a classic wall with insulation,
– a concrete wall with simple or double glazing, and
– a concrete wall with glazing and an alveolar structure, equipped with glass lamellas inclined with regard to the horizontal plan.

Results showed that the wall provided with a multi-alveolar structure has more efficiency than a wall with simple glazing and in certain cases equivalent to a wall with double glazing.

Zugari and Vullierme [5] specified that a simple glazing equipped with inclined lamellas structure will have during one day (the incidence of radiation varying constantly) an overall efficiency upper to that of the simple glazing or double glazing. The energy saving is often sensitive in the case of a selective cover. Results show that a distance window-sensor of the order of 10 cm, placed lamellas all the 20 cm with an inclination of 60° certainly constitutes an optimal structure for an average daily behavior (fig. 1). The obtained results drove the authors to pursue this study when the simple vertical glazing is inclined by an angle β to receive the maximum solar flux. In fact, this inclination can modify the anti-convective effect of this structure. They then studied the influence of this angle on heat transfer and on the speed of fluid.

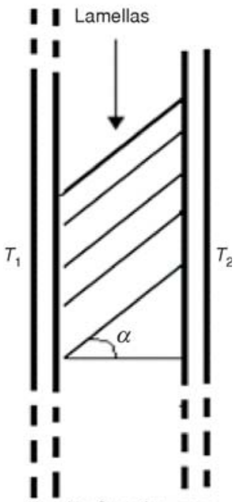

Figure 1. A schematic of multi alveolar structure

Alkhoja [4] and Zugari, and Vullierme [5] have shown that the performance of simple glazing equipped with inclined lamellas was improved in comparison with a conventional simple glazing.

Vullierne and Boukadida [6] realized a first numerical study and showed that the coefficient of thermal diode (*Ed*), defined as being the ratio between the global heat coefficient (including the convection and the radiation) in the spending and insulating senses, increases appreciably with the angle of inclination α.

A space of the vertical walls and the report of shape have a lot of influence. By leaning on this study and by considering the problem of a practical realization, the retained alveolar structure is formed with lamellas tilted to 60° with regard to the horizontal plan and has a report shape equal to 1 and a space between the vertical faces of 20 cm. By continuing the study, Boukadida and Vullierme [7] experimentally determined the global density of heat transfer flux including convection and radiation (F_a) in the crossing and insulating senses through the realized alveolar structure. These measures allowed to bring out laws for different distributions of low or high emissivity coatings of the inside faces of alveolar. These lows are defined by the following correlation: $F_a = \sigma(\Delta T)^{1.25}$ where σ is a constant which depends of emissivity, transfer direction, and the angle of inclination.

In order to show the effect of the anti-convective structure, Ben Amor *et al*. [8] studied heat transfer in a room by using this alveolar structure. The aim of this work was to study the external temperature, solar flux, and wall nature effects on the building thermal behavior using a structure with a diode thermal effect. The structure is conceptualized to be used for a cooling or a heating application. Numerical simulations allowed us to compare the thermal behavior of a building equipped with this structure on its east, south, and west frontages to that of standing or conventional building with large or low inertia. Heat transfer equation is numerically resolved by the numerical nodal method in case of one dimension. The simplified used model is subdivided into 38 nodes. Simulations were made for a cooling application in a deserted zone where the thermal amplitude between day and night is sensitive. Results showed the effect of conducted and insulated wall layers thickness and the external solar flux on the local thermal behavior. They also showed that the average interior temperature of place equipped with this structure is slightly lower than one having high or low thermal inertia.

Boukendil and Abdelbaki [9] numerically studied 2-D coupled heat transfer by conduction, convection, and radiation through a hollow brick formed by an air layer separating two alveolar walls. The outside vertical surface of the structure is submitted to a sinusoidal thermal excitation $T_e(t) = T_m + a\sin(\omega t)$, and the inside vertical surface is maintained at a constant temperature (T_i). The top and bottom horizontal sides of the structure are assumed to be adiabatic. Equations governing natural convection in different cavities of the system, radiation heat exchanges between the internal surfaces of cavities and heat conduction in the surrounding walls are solved using the control volumes method and the SIMPLE algorithm. The impact of the sinusoidal thermal excitation $T_e(t)$ on both global heat flux through the structure and maximal stream function ψ_{max} was predicted for different values of the amplitude a and the period T. The effect of the emissivity ε of the internal surfaces of cavities on the heat transfer and the air flow is also discussed.

The main results of this study can be summarized as follows.

- The heat flows $\phi_e(t)$ and $\phi_i(t)$ which passes through the honeycomb structure from the outer and inner faces and the stream function $\Psi(t)$ are also periodic.
- The mean values of the stream function $\Psi(t)$ depend on the excitation temperature amplitude a, in contrary the mean value of heat flow $\phi_i(t)$ is independent of a and is significantly equal to that of steady state. The amplitudes of $\phi_i(t)$ and $\Psi(t)$ depend on the amplitude a of the excitation temperature T_e (t).
- The emissivity ε of the internal surfaces of cavities greatly affects both the overall heat flow through the structure and the stream function. Indeed, the radiation contributes to increase the heat flow and tends to reduce the gap of surface temperatures and consequently participates indirectly to decrease the maximum value of the stream function Ψ_{max}.

Position of the problem

Based on the above and on previous work, we are interested in studying the thermal behavior of the premises; the walls are equipped with alveolar structure. To increase the energy storage inside the premises a single glazing is used on the sunniest sides (east, south, and west). Each wall is exposed to variable solar flux.

Description of premise walls

Figure 2 depicts the kind of different walls in the building. An alveolar structure or an insulator layer can take place as an intermediate layer. Table 1 contains the description of each layer for the different walls.

Table 1. Description of each wall

Layers thickness in traditional building		Layers thickness in building with alveolar structure and glazing
Outside layers	• *Insulating coat:* – 2 cm in: North, ceiling – 4 cm in: south east and west sides – 1cm: in floor	• *Insulating coat*: – 2 cm in: North, ceiling – 1cm in: floor • Simple glazing: – 4 mm in: south, east, and west sides
Intermediate layers	• Tin: 1 mm, • Glass wool: – 5 cm:North and ceiling • Wood and glass wool: – 2 cm: south, east and west sides – Concrete: – 6 cm: floor – Glass wool: – 5 cm: floor	– Tin:1mm – Glass wool: – 5cm: North and ceiling – Wood and slats: – 2cm: south, east, and west sides – Concrete : – 6cm: Floor – Glass wool – 5cm: floor
Inside layers	• *Wood*: – 1cm: north, ceiling – 2 cm: south, east and west sides • Concrete: – 6 cm:floor	• *Wood* – 1 cm:North, ceiling – 2 cm: south, east, and west sides • Concrete: – 6 cm: floor

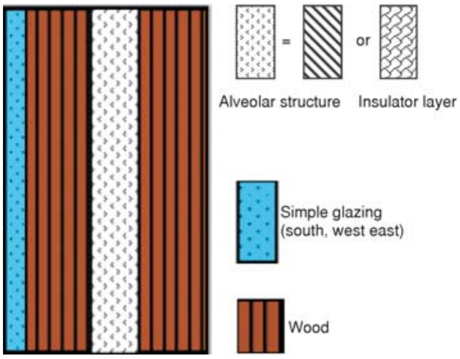

Figure 2. Different wall elements

The working assumptions

- Heat transfer by conduction is unidirectional.
- Air is considered a perfectly transparent gas.
- The thermo-physical properties of materials are constant.
- Heat capacity of the glass is low given to low volume.
- The temperature of air inside the room is uniform.
- The energetic participation of the occupant is negligible.

Formulation of the problem

The thermal balance of element i is:

$$(mc)_i \frac{\mathrm{d}T_i}{\mathrm{d}t} = \sum_{j=1,n} C_{i,j}(T_j - T_i) + \sum_{j=1,n} K_{i,j}(T_j^4 - T_j^4) + P_i \tag{1}$$

Thee previous system composed of *n* equations of type $F_i(\mathrm{T}_1, \mathrm{T}_2, \mathrm{T}_3 \ldots T_n) = 0$ is not linear because the presence of F_i^4 terms. It is assumed that there is an approximate solution which is written as: $F_i(T_1 + \Delta T_1, T_2 + \Delta T_2, \ldots T_2, T_3, \ldots T_n + \Delta T_n) = 0$.

Developing in Taylor series limited to the first order, we get:

$$F_i(T_1, T_2, \ldots T_n) + \Delta T_1 \left(\frac{\partial F_i}{\partial T_1}\right)_{T_1, T_2, \ldots T_n} + \ldots + \Delta T_n \left(\frac{\partial F_i}{\partial T_n}\right)_{T_1, T_2, \ldots T_n} = 0$$

Linear system in ΔT_i is obtained and the implicit method is used to calculate the different temperatures. An iterative calculation is then performed at each time step (one hour) until the solution near the convergence criterion (0.1 °C). At each iteration, convective and radiative heat transfer coefficients which depend on temperature are recalculated. Solicitations correspond to periodic thermal conditions over 24 hours, the calculation is repeated a sufficient number of times to obtain a periodic steady state. For two successive iterations, the convergence criterion over 24 hours is 0.05 °C.

Based on the multi-reflectance on surface Si, the radiosity notion and by using the balance equation for the element *j*, the radiative coefficient $K_{i,j}$ is expressed by:

$$K_{i,j} = \frac{\varepsilon_i \varepsilon_j A_j \sigma}{1 - \varepsilon_j} X_{i,j} \quad \text{with} \quad X_{i,j} = \frac{r_j E_{i,j}}{\varepsilon_i A_j \sigma T_i^4}$$

where $E_{i,j}$ is the fraction of E_i that reaches the surface *j* indirectly or by reflection from other surfaces.

Thermal boundary conditions

External conditions

The convective heat transfer coefficient reflecting the exchange between the outer wall and ambient air is assumed to be uniform, we have taken the values 11 W/m^{2}K for each vertical face and 14 W/m^{2}K for the horizontal face. The solar flux including direct and diffuse radiations is calculated hourly, the 15th day which is considered as a typical day of the month.

Internal conditions

Temperature

In the first case, the temperature varies according to time, in the second case, it is fixed as:

19 °C, during the day time (from 7:00 to 21:00)
16 °C, during the nocturnal time (22:00 to 6:00)

Global heat transfer coefficient in the rectangular cavity

The global heat transfer coefficient in the rectangular cavity is expressed by:

$$h_\mathrm{g} = h_\mathrm{c} + h_\mathrm{r} \tag{2}$$

where:

- h_c is the convective heat transfer coefficient expressed by the following correlation derived from [5]:

$$h_\mathrm{c} = 0.046 \sqrt[3]{\left(9.81 \frac{T_\mathrm{c} - T_\mathrm{F}}{T_\mathrm{m} \nu^2}\right)} \lambda + 1 \tag{3}$$

– h_r is the radiative coefficient between the glass and the absorber defined by:

$$h_\mathrm{r} = \sigma\varepsilon_\mathrm{e}(T_C + T_F)(T_C^2 + T_F^2) \tag{4}$$

– ε_e is the equivalent emissivity of vertical walls (absorber-glazing) which is expressed by:

$$\varepsilon_\mathrm{e} = \frac{1}{\dfrac{1}{\varepsilon_\mathrm{s}} + \dfrac{1}{\varepsilon_\mathrm{abs}} - 1} \tag{5}$$

ε_s and ε_abs: are respectively glass and absorber emissivity.

Global heat transfer coefficient inside alveolar

We have opted for the correlation including convection and radiation, determined experimentally by Boukadida *et al.* [8]:

$$h_\mathrm{t} = \gamma(\Delta T)^{0.25} \tag{6}$$

where γ is a coefficient which depends on the heat direction, the angle of inclination, and faces emissivity of the lamellas (low or high emissivity). It is obtained for an angle of 60° and takes the value of 2.950 in the spending direction and 1.388 in the insulating direction.

Diode effect coefficient Ed

It is defined as the ratio between the time average of convective heat transfer coefficient during the day time (spending direction) and the nocturnal period (insulating direction):

$$Ed = \frac{h_\mathrm{ts}}{h_\mathrm{ti}} \tag{7}$$

Coefficient of heat transfer between interior faces and indoor air of the premises

In order to characterize the convective heat transfer between interior faces and indoor air, we used the classic average correlation:

$$\mathrm{Nu} = A(\mathrm{Gr}\ \mathrm{Pr})^\mathrm{B} \tag{8}$$

with $A = 0.11$, $B = 0.33$ for (faces: north, south, east, west), $A = 0.27$, $B = 0.25$ for roof, $A = 0.14$, $B = 0.33$ for floor.

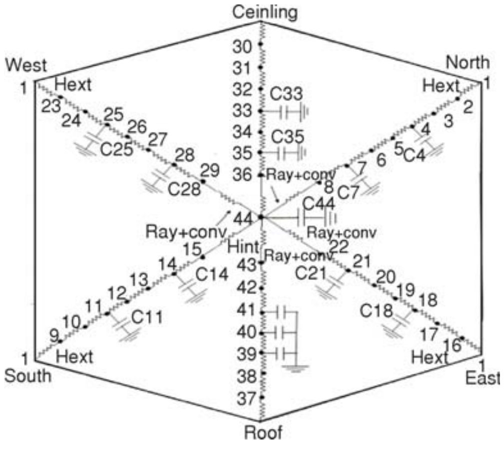

Figure 3. Schematic plan of various thermo-electric models

The Grashof and Nusselt numbers are respectively defined by:

$$\mathrm{Gr} = \frac{\beta g \Delta T L^3}{T_m \gamma_m^2}, \quad \mathrm{Nu} = h\frac{L}{\lambda_m} \tag{9}$$

Numerical methods

The numerical method used is the nodal method, the system is divided into several elements, each one is represented by a node placed at its center and affected by the average temperature and specific heat capacity. To limit the number of nodes, we used the method of fictitious node to transcribe the exchange surface. The model is divided into 44 nodes. Each wall contains 7 nodes (4 nodes for the outer wall and 3 nodes for the inner wall). The medium between the two walls (insulating or rectangular cavity or alveolar structure). By using the thermo-electric analogy a schematic circuit is presented in fig. 3.

Results and interpretations

It should be remembered that for all simulations, each month is represented by its fifteenth day considered as a typical day.

Meteorological temperature and solar flux in Tozeur

Figures 4 and 5, respectively, show time evolution of metrological temperature and solar flux in Tozeur (south of Tunisia). The maximum values of temperature difference between the daytime and night periods are more important during the summer (fig. 4). The annual evolution of the global density of incident solar flux received by vertical wall, illustrated in fig. 5 shows that on the south side, the solar flux is more important in winter and minimum in summer. This is mostly due to the fact that the angle of incidence (angle between the normal to the vertical wall and the solar beam) is much larger in summer than in winter. The time evolution of the global density of incident solar flux received by a vertical wall (fig. 6) shows that on the south side and in the roof, the flux is symmetrical with regard to midday. The south side has an important role in January, fig. 6(a), since the maximum flux estimated at 750 W/m^2 that is reached and conversely minimum in July, fig. 6(b), but in the roof, the flux is minimum in January, fig. 6(a), and maximum in July, fig. 6(b).

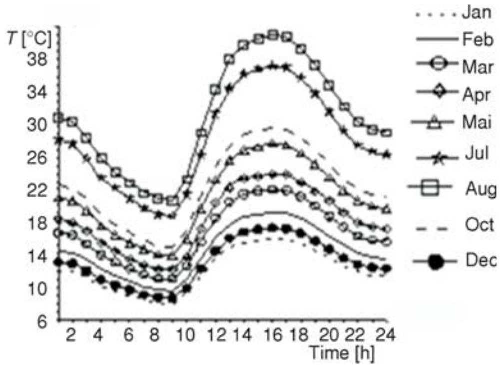

Figure 4. Time evolution of meteorological temperature

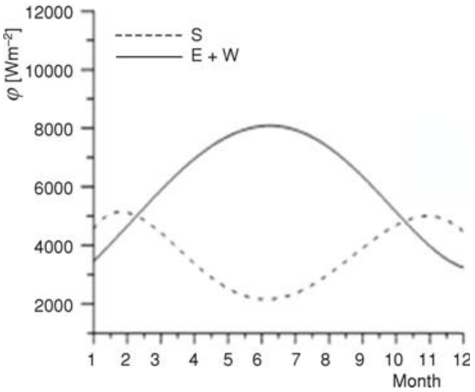

Figure 5. Annual evolution of the global density of incident solar flux

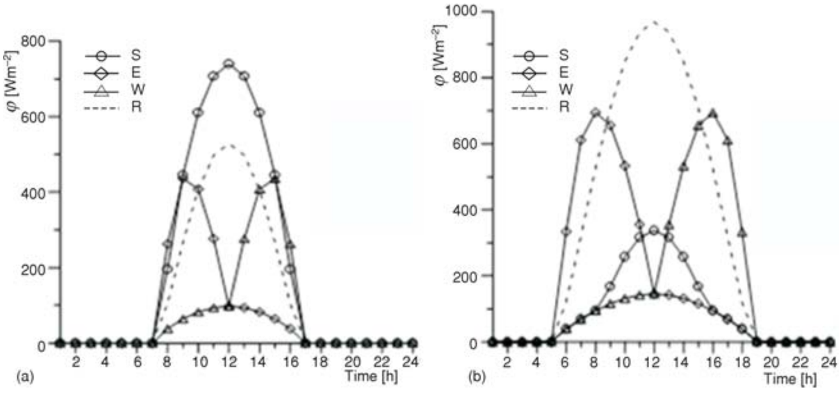

Figure 6. (a) Time evolution of the global density of incident solar flux in January; (b) time evolution of the global density of incident solar flux in July

Time evolution of interior air temperature in case without a set point temperature

The local is not equipped with alveolar structure

For the month of January, fig. 7 illustrates the time evolution of convective heat transfer coefficient in each sensor. As it is shown, each profile sensitively depends on time variation of the global density of the received solar flux in each face. During night time, the convective coefficient of heat transfer (h_c) is of the order of 5.5W/m^{2}K but in daytime it can reach 8.6 W/m^{2}K (south face). Figure 8 illustrates the effect of glazing number of interior air temperature of the premises which shows there is a small crossing between curves. This crossing is principally due to the density solar flux that is very remarkable in the south side. The glazing number increases as the interior air temperature rises.

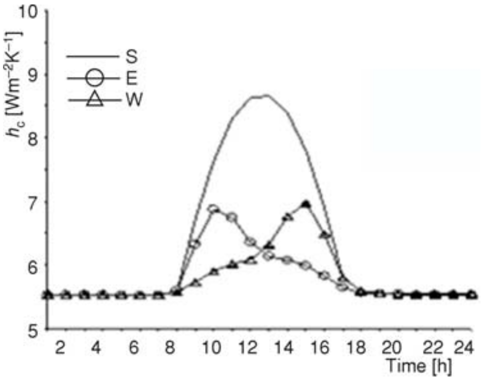

Figure 7. The time evolution of convective heat transfer coefficient in each glazing

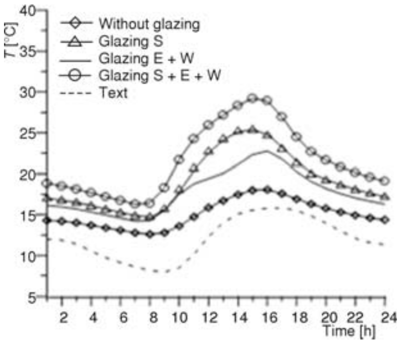

Figure 8. Effect of glazing number on air temperature of the local

The premises are equipped with alveolar structure and simple glazing

Figures 9 and 10, respectively, show the annual evolution of the thermal diode effect coefficient (with glazing: south + east + west or without glazing).We notice that there is a small

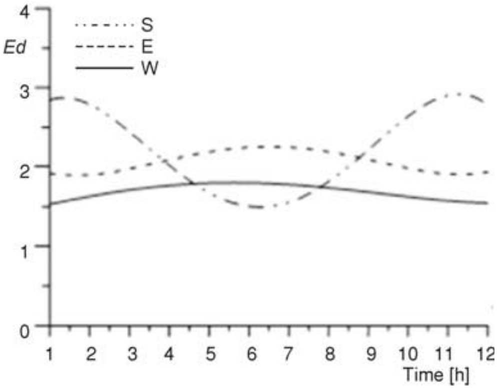

Figure 9. Annual evolution of the thermal diode in case of three glazing

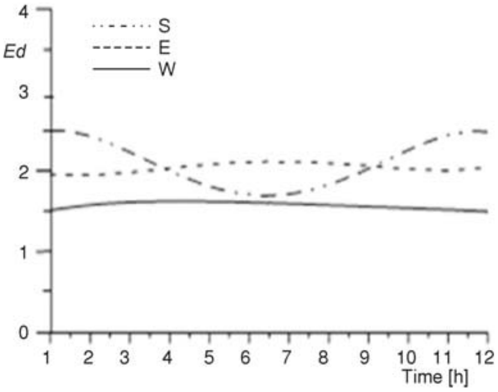

Figure 10. Annual evolution of the coefficient thermal diode coefficient in case without glazing

variation in case of east and west faces during the summer. In this case, although the contribution of sunshine is very intense we can distinct that the alveolar structure inhibits the passage of flux during the warm season. The effect of simple glazing is interesting for the south side during the cold season. In fig. 11, the effects of the alveolar structure on the interior local temperature show that the maximum value of temperature is presented in case with alveolar structure. This rise is mainly due to thermal diode phenomena.

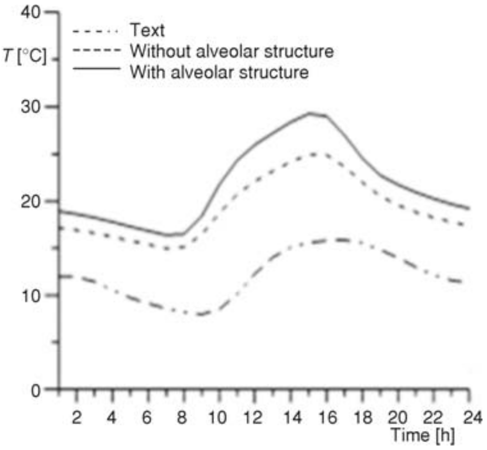

Figure 11. Effect of alveolar structure on interior air temperature of the premises

Time evolution power in case of set point temperature

Figures 12 and 13 show the annual evolution of the heating power in case of traditional premises, premises with glazing (E, S, W) and premises with alveolar structure (E, S, W).The alveolar structure and the simple glazing are not advantageous during the summer for an application of type housing environment or administrative use but can be used in the case of an application in which we have to save energy.

As it is depicted in fig. 13 using glazing in east, south, and west frontages causes sensitive energy saving in winter and spring seasons.

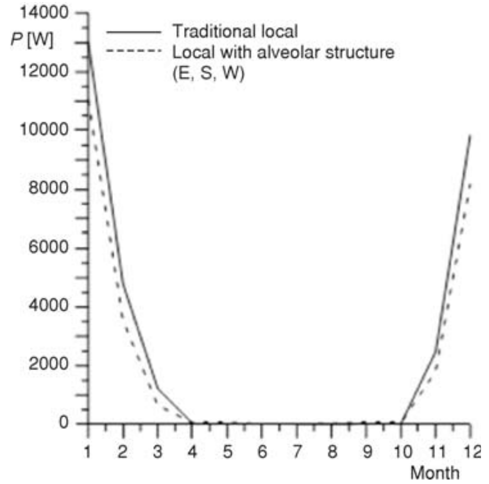

Figure 12. Effect of alveolar structure on heating power without glazing structure

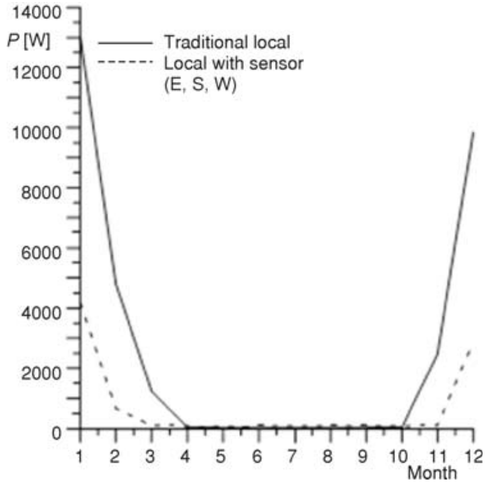

Figure 13. Effect of 3 glazing (E, S, W) on the heating power without alveolar

Conclusions

The objective of the numerical study is to compare instationary state, the monthly and annual thermal behavior of local ($S = 30$ m^2 and $V = 300$ m^3) equipped with alveolar structure and/or simple glazing.

The main results are as folow

- In case of simulation with or without set point temperature, the alveolar stucture allows to save energy in winter and spring seasons.
- In case of simulation without set point temperature, the interior air temperature is fairly higher in the presence of the alveolar structure, it increases according to the simple glazing number used on faces.
- In the presence of the simple glazing energy saving by the premise is more important, this case is interesting for the cold season.

Nomenclature

A_j – square of surface j
$C_{i,j}$ – conductive and/or convective coefficient between nodes i and j [$\mathrm{WK^{-1}}$]
E – East
g – acceleration of gravity, [$\mathrm{ms^{-2}}$]
H – height of the vertical walls of the cavity, [m]
h_c – convective heat transfer coefficient, [$\mathrm{Wm^{-2}K^{-1}}$]
h_r – radiative heat coefficient, [$\mathrm{Wm^{-2}K^{-1}}$]
j – emissivity of surface j
$K_{i,j}$ – radiative coupling coefficient between nodes i and j [$\mathrm{WK^{-4}}$]
L – horizontal distance between the walls of the cavity, [m]
$(mc)_i$ – heat capacity, [$\mathrm{JK^{-1}}$]
P_i – solar flux absorbed at the time (t) by node i [W]
R – Roof
S – South
T_C – hot wall temperature, [°C]
T_F – cold wall temperature, [°C]
T_i – temperature at the time (t) by node i [K]
T_m – average temperature $T_\mathrm{m} = (T_\mathrm{C} + T_\mathrm{F})/2$, [°C]
W – West

Greek symbols

α – angle of inclination
λ – thermal conductivity of air, [$\mathrm{Wm^{-1}K^{-1}}$]
ν – kinematic viscosity of air, [$\mathrm{m^2s^{-1}}$]
σ – Stefan-Boltzman constant ($5.6 \cdot 10^{-8}$ $\mathrm{Wm^{-2}K^{-4}}$]
β – dilatation coefficient, [$\mathrm{K^{-1}}$]

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