NUMERICAL INVESTIGATION OF A WALL’S OPTIMUM MULTILAYER THERMAL INSULATION POSITION

Lazaros Mavromatidis1,2, Anna Bykalyuk1, Mohamed El Mankibi1, Pierre Michel1, and Mat Santamouris3
1Université de Lyon; ENTPE, CNRS FRE 3237, Building & Civil Engineering Department, Building Sciences Laboratory, 69518 Vaulx-en-Velin Cedex, France
2Architect-Engineer; Grant holder of the “Alexander S. Onassis” public benefit foundation
3National Kapodistrian University of Athens, Physics Department

ABSTRACT
This study aims to provide insights regarding the benefits of multilayer insulations in building applications placed in combination with two air gaps. For this purpose, a numerical approach was developed to determine the best multilayer insulation position. A combined radiation/conduction heat transfer numerical model was employed to predict the temperature distribution and heat transfer in multilayer insulation complexes comprised of insulating materials separated by multiple reflective foils. The radiation scheme was based on the two flux approximation, in order to model both optically thick and optically thin fibrous materials. The heat transfer equation was solved explicitly for a composite wall. Total nine different configurations were selected as initial state, where the theoretical thermal resistances were calculated and compared.

INTRODUCTION
Multilayer Thermal Insulation (MTI) products are nowadays subjected to a scientific and technological debate, which stems from the difficulty on defining the terms of characterization of their insulating capacity. This happens because their performance depends notably from the infrared radiative balance (long wavelengths) between reflective layers and outer surfaces. These materials are consisted of a stack of thin insulating layers (moss, wool plant, etc.) and thin reflective layers (a few tens of microns in length).

Our research in the framework of a Phd thesis started using an experimental type guarded hot box to investigate the insulation performance of a typical MTI product. After this first phase (see also Mavromatidis et al, 2010a ; 2010b), analytical and numerical modelling of heat transfer phenomena in the MTI was performed. The physical analysis of the thermal performance of some typical wall configurations including MTI products was decided to be approached numerically because of the concrete and economic advantages of this choice. Once a numerical model is validated according to existing experimental measurements and after verifying the accuracy of the simulated results, it could be used in order to study problems that may require complex and expensive experimental devices (such is the case of our research). Also, because of the high-speed computing systems available today, quick lesion is provided by reducing significantly the time required for the study.

Even if there are established models for other types of constructions, the development of a new model was realised in order to overcome difficulties that raised by the particularity of the experimental setup and the nature of the experimental data that should be introduced initially as inputs in conjunction with the boundary conditions. The created model is independent of the configuration’s particularity, is easy to use and can be employed in order to simulate any insulation complex (both multilayer and bubble insulations, while we can put any number of air gaps in any position) and a number of different boundary conditions.

In addition, it can be easily modified and developed for conditions that may not have been taken into account in the current phase of the holistic study on MTI. Certainly in this paper, the definition of the boundary conditions, the resolution of governing equations, the definition of the variables, the discretization method and the choice of nodes were based on the particularity of the guarded hot box experimental platform developed in the Building Sciences Laboratory in the ENTPE of Lyon.

After testing many different configurations the numerical solution presented in this paper was selected because it offers the best accuracy in the calculation of variables compared to the collected experimental data.

EXPERIMENTAL STUDY
Measurements in a guarded calibrated hot box
The principle of a guarded hot box consists to place the sample of which the overall thermal resistance is desired to be measured (3) between a hot chamber with a heating element (2) surrounded by a guarded heat ring (1) and a cold chamber (4) equipped with a cooling thermodynamic system (Figure 1). A temperature difference greater than or equal to 20°C is imposed between the two chambers of the device. In the hot chamber a sinusoidal type temperature profile was imposed while in the cold chamber the temperature profile seems to be constant. Temperature measurements allow to determine the
thermal resistance of the tested sample, using the
relation \( R = \frac{S \Delta T}{\Phi} \).

The guarded hot box of the Building Sciences Laboratory in the ENTPE of Lyon (Figure 2) measured the overall heat transfer through a large non-homogeneous wall composed of wood, air and MTI product. The experimental platform has determined the overall heat transfer through the structure by measuring the thermal resistance, which was resulted by the heat flow across the vertical wall structure. In addition to the options mentioned above and according to the CEN / TC 12 WG pr EN 16012:2010 standard’s revision, any type of MTI can be measured using the experimental procedure of the guarded hot box. Furthermore, according to this latest revision of the international standard, the guarded hot box experimental platform can be used for all thicknesses and the nominal thermal resistance of these products. The calibration of the guarded hot box has been done using different samples of Effisol (the calibration results are presented elsewhere; Mavromatidis et al., 2010a).

Sample description

The multi-layer sample investigated in this study for validation of the numerical heat transfer model and for further parametric investigation regarding the optimum MTI position in a composite wall, consisted of reflective foils separated by layers of polyester wadding and polyester fibrous insulation. It had seven silver-coated foils with a measured emissivity of 0.2 for the exterior foil and 0.15 for the interior foils respectively. At both sides of the MTI product, there were two 2 cm thick air gaps (Figure 4). The fibrous insulation spacers were made of polyester wading and polyester. Equivalent thermal conductivity and volumetric heat capacity of MTI’s composites were measured using the patented Hot Disk TPS 1500 Thermal Constants Analyzer Test System that had an average disc resolution of 3.65% during the different experimental sessions (the detailed measuring procedure is presented elsewhere; see Mavromatidis et al., 2010b).

Chain control of the guarded hot box

According to the standard “EN ISO 8990”, 120 PT 100 sensors with a resolution of 0.1°C “standard IEC B class” (see Figure 3) were installed on the sample’s surface and in different combinations (Jung et al., 2008). Two heating elements were installed in the measuring chamber (80 W) and the box cover (250 W). Four fans were also installed in the casing cover to homogenize the air temperatures. In order to break through the problems that arise during the AC measurement of heating power, a DC source (0-220V) driven by a voltage 0-5V was integrated into the device. The measurement of heating power injected into the measuring chamber was determined through a power converter (0-150 W) with an accuracy of 0.25% on the full scale (Figure 3).
Both steady-state and transient tests were conducted in the guarded hot box for studying the thermal behaviour of multi-layer insulations and for validating the computational heat transfer model that is used in this study. The case of the insulating complex is built using 1 cm thick wood panel with a measured emissivity of 0.8 and a measured volumetric specific heat of 524.9 kJm⁻³K⁻¹. The total thickness of the sample construction (including the wood) was 3.6 cm plus the thicknesses of the two air gaps (Figure 5). The total thickness of the sample construction (including the wood) was 7.6 cm.

**NUMERICAL ANALYSIS**

In this study, we focused on modelling both optically thick and optically thin materials. Therefore, the equation of energy conservation (Ozisik, 1973) is transformed (Ning Du et al 2008; Daryabeigi 2002) using the two-flux approximation (Milne - Eddington approximation) in:

$$\frac{\partial T}{\partial t} = \frac{\partial}{\partial x} \left( k \frac{\partial T}{\partial x} \right)$$

To resolve this problem, two boundary conditions and an initial condition are needed. A linear temperature profile through the thickness of the wall was taken at \( t = 0 \) which is the initial condition of the problem \( T(x, 0) = T(0) \) (Figure 5). To overcome the problem of the temperature results’ comparisons on the two limits of the insulating complex, we introduced the pseudonodes method at the input and output limits of the numeric field (Georgantopoulou et al., 2007). In particular, two additional grid points (both within the cold and the hot guarded chamber) were added (Figure 6).

As a result, if we separate the field into \( N-2 \) control volumes, which means that the number of grid points in the insulating complex is equal to \( N-2 \) and the number of interfaces of the control volumes is equal to \( N-1 \), the total number of grid points is \( (N-2)+2=N \). Denote for the temperature field \( T_j(t) \), \( j=1,2,\ldots,N \), while for the flow field of heat radiation, which is calculated at the interfaces of control volumes, it is in effect \( \Phi_j=F(B_j), j=1,2,\ldots,N-1 \) (with \( B_j \) is symbolized the \( j \) interface, while similar notations are in effect for \( F \)). Thus, because of the nature of the problem, the morphology of the physical domain and the available experimental data, the initial and boundary conditions are as follows:

**Initial Condition**

$$T(x, 0) = T_0(x)$$  \hspace{1cm} (2)

**Boundary Conditions**

$$T(0, t) = T_{p_0}(t)$$
$$T(L, t) = T_{p_{10}}(t)$$

In the above relations with \( 0 \) is denoted the first boundary (Figure 6), while with \( L \) is denoted the second boundary (also the total thickness) of the insulation complex (Figure 6). \( T_j(t) \) are the values of the air temperature in the hot chamber of the guarded hot box (symbolized with \( T_h \)) and \( T_j(t) \) are the values of air temperature in the cold chamber of the hot box (symbolized with \( T_c \)). However, considering convective boundary conditions the temperatures at the hot and cold limits of the insulation complex are calculated as follows:

$$h_0 \left( T_h(t) - T_c(0) \right) = -k \frac{\partial T}{\partial x} \bigg|_{x=0}$$
$$h_1 \left( T_c(L) - T_c(t) \right) = -k \frac{\partial T}{\partial x} \bigg|_{x=L}$$

The heat transfer coefficients \( (h_0 \) and \( h_1 \) take into account both convection and radiation heat transfer between the boundaries of the insulation complex and the air with temperatures \( T_h(t) \) and \( T_c(t) \). For model validation, simulated and measured temperature values at the external boundaries of the wall sample \( T_{v_h}(t) \) and \( T_{v_c}(t) \) have been compared (Mavromatidis et al., 2010b).

The main advantage of using the two-flux approximation is that, even if the typical fibrous insulation materials commonly used on building applications are usually optically thick, the model presented here is not limited to optically thin or thick materials (Daryabeigi, 2002). For this reason, the two-flux approximation is considered a suitable technique regarding fibrous insulation spacers used in the extremely thin MTI componsants, which does not fall into the category of optically thick materials.

This method was recently used, among others, by Daryabeigi (2002), Ning Du et al (2008), Dan Bai and Fan Xu-Ji (2007) and Zhang et al (2008) to calculate the radiative heat transfer in fibrous insulation, while the same method was used by Daryabeigi (2001) and Zhao et al (2009) to calculate the heat transfer radiation in fibrous insulation subjected to aerodynamic heating conditions. In our
model we used this technique for the first time in order to simulate wall configurations including such a big number of different layers (in total 25 layers including the reflective foils).

Thus, according to Ning Du et al (2008) the attenuation of the radiant flux is given by:

\[
\frac{\partial F^+}{\partial x} = -\beta F^+ + \beta \sigma T^+(x) \\
\frac{\partial F^-}{\partial x} = +\beta F^- - \beta \sigma T^-(x)
\]

(5)

A sketch of the multilayer system composed of N sections (radiation blocks) is shown in Figure 7. Each section is formed from the insulating materials within the MTI product and the two adjacent reflective sheets, while each section is further subdivided into optically thin layers (control volumes). The reflective foils work as radiant barriers and that is why they impose boundary conditions for the radiation scheme in the inner part of the multilayer product.

Thus, appropriate radiative boundary conditions for the forward and backward radiation fluxes are obtained from the balance sheets in the front \((x = 0)\) and the rear \((x = l)\) of each section:

\[
F^+(0) = \varepsilon_1 \sigma T_{\text{in}}^2 + (1-\varepsilon_1) F^-(0)
\]

\[
F^-(l) = \varepsilon_2 \sigma T_{\text{out}}^2 + (1-\varepsilon_2) F^+(l)
\]

(6)

where the subscripts 1, 2 refer to the boundary surfaces at \(x=0\) and \(x=l\), respectively.

\[
\text{Figure 7: Division of the MTI product in N radiation sections [source: Mavromatidis et al., 2010b].}
\]

The integration of equations (5) over a control volume \([B_{j-1}, B_j]\) led to the following equation for the radiative fluxes that develop forward and backward respectively:

\[
(F^+)_{j-1} = \left[\frac{2 - \beta \Delta x_j}{2 + \beta \Delta x_j}\right] (F^+)_{j-1} + \left[\frac{2 \beta \sigma \Delta x_j}{2 + \beta \Delta x_j}\right] T^+(x_j)
\]

\[
(F^-)_{j-1} = \left[\frac{2 - \beta \Delta x_j}{2 + \beta \Delta x_j}\right] (F^-)_{j-1} + \left[\frac{2 \beta \sigma \Delta x_j}{2 + \beta \Delta x_j}\right] T^-(x_j)
\]

(7)

At the beginning of the procedure, the initial \(F(0)\) value is chosen arbitrarily, and using the equation (6) \(F(0)\) is calculated. Then the equation (7) is applied to calculate all the \(F\) values. Once the \(F(l)\) is obtained, the \(F(l)\) is calculated using equation (6). Then, equation (7) is applied to calculate all the \(F\) values. Once the iteration period is complete, the \(F(0)\) is updated. If the difference between the calculated and the initial values of \(F(0)\) is lower than \(10^{-5}\), the iteration procedure is finished. Otherwise, the updated \(F(0)\) value is used to start a new period of iteration. The temporal distribution of temperature in the wall complex, using the control volume technique (Patankar, 1980) is:

\[
T^+ = \frac{\left[\frac{k_y}{\delta x} T_{\text{in}}^+ + k_y}{(\delta x)_j} T^{+}_j\right] \frac{\Delta t}{\rho c_{\Delta x} d x} + \frac{1}{(F^+ - F^-)} (F^-) - (F^+) + T^+
\]

(8)

where the index \(j\) represents the values that correspond to the \(j^{th}\) grid point which is the center of the \(j^{th}\) control volume and the exponent \(n\) is the current time step. The control volume formulation makes the overall balance of heat fluxes possible for any number of grid points, but care is needed, in calculated fluxes at the control volume interfaces (Patankar, 1980). Obviously, the heat flux that leaves one control volume through a particular face must be identical to the flux that enters the next control volume through the same face. Thus, an interface flux is considered independent, and not as belonging to a certain control volume. For better representation regarding the heat flux at the interfaces the interface conductivity was determined (details are presented elsewhere, Mavromatidis et al., 2010b). Denote that in order to obtain Eq. 8 the temperature profile assumption that was used is the piecewise-linear profile. For this profile, the slope \(dT/dx\) is defined at the control-volume faces and linear interpolation is used between the grid points.

The space between the nodes is uniform in all regions of the fibrous layers and it is limited by two reflective layers or a reflective layer and a solid surface of another material. However, the space between the different nodes may vary between the different layers (in this study \(\Delta x=0.5\text{mm or } 1.0\text{mm}\) depending on the nature and the geometry of each insulating layer. Finally, the time step is 0.125 seconds, and satisfies the stability criterion for the explicit scheme that is \(\Delta t < \rho c (\Delta x)^2 / 2k\).

**Model evaluation**

The model evaluation was based on the point to point comparison between the simulated temperature values at both boundaries of the insulation complex and the experimental measurements (the validation results are shown elsewhere; see also Mavromatidis et al, 2010b). This evaluation work was carried out for several months in order to validate the numerical model in various configurations and boundary conditions.

**Factorial simulation plan**

After having successfully evaluated the numerical model in accordance with the guarded hot box’s experimental data, a simulation plan has been
established in order to discover the best MTI’s position in the wall complex. For this purpose suffice it to identify the appropriate thicknesses of the two air gaps in order to obtain the maximum thermal resistance for the whole insulating complex.

Furthermore, after this simulation phase we could be able to obtain a mathematical expression of the influence of the air gaps’ thicknesses on the equivalent thermal resistance of the insulating complex. Considering that the two variables that we need to investigate in this case study are the nominal thicknesses of each air gap, a 2² complete factorial design (two factors at two levels) was used to evaluate the influence of two different levels for each variable on the relevant insulating performance of the wall complex.

According to Montgomery (1996) such a two-level factorial design requires a minimum number of tests for each variable. Given the fact that the expected responses (the Thermal Resistance of the insulating complex) do not vary in a linear manner with the selected variables (the thicknesses of the air gaps) and considering that all control factors are quantitative, in order to enable the quantification of the prediction of the responses, a central composite plan was selected, where the response could be modelled in a quadratic manner (Romero-Villafranca et al., 2007). Departing from basic factorial simulation plans, we rejected the most simplified ones because we obtained high residual values during the statistical validation procedure. This statistical observation was expected after knowing from the physical nature of the problem the key role of the air gaps in the thermal resistance of the wall configuration. For this reason, in order to include the sensible influence of the air gaps on the nominal thermal resistance of the complex we converted the general equation introduced among others by Box and Jones (1990), Lucas (1994) or Myers et al., (1997) into:

\[ R(Y / x) = b_0 + \sum_{i=1}^{2} b_i x_i + \sum_{i=1}^{2} b_i x_i^2 + \sum_{i,j=1}^{2} b_{ij} x_i x_j \]  

where \( x_i (i=1,2) \) is the level of the \( i^{th} \) control factor (in our case the thickness of each air gap). Furthermore, in order to eliminate the noises we limited the use of the models to an area bound by coded values corresponding to \(-a\) to \(+a\) limits (Lucas, 1994). The parameters were carefully selected to carry out composite factorial design, where the effect of each factor is evaluated at three different levels (two for the investigation and one for the validation of the statistical model) in codified values of \(-1, 0, +1\), selecting for extreme nominal thicknesses realistic applicable values on building applications (minimum extreme thickness: 1 cm / maximum extreme thickness: 5 cm). All the other parameters that influence the thermal resistance, such as emissivities thermal capacities densities etc, were fixed for all different simulation scenarios equals to the measured values, because the purpose of this parametric study is the investigation of the air gap’s influence on a wall’s thermal resistance in order to identify the optimum MTI position. Appropriate commercial software was used for the statistical analysis of the results (NEMRODW).

**SIMULATION RESULTS AND DISCUSSION**

The simulated and statistical calculated resistances of 9 different combinations of air gaps’ thicknesses to derive the factorial design model along with the standard deviation are summarized in Tables 1 and 2 respectively. The simulated thermal resistances on steady state were calculated employing the relation: 

\[ R = \frac{S}{\Delta T} \]  

**Table 1**

Matrix of simulations where level -1 represents an 1 cm air gap, level 0 represents a 3 cm air gap and level 1 represents a 5 cm air gap.

<table>
<thead>
<tr>
<th>Simulation N°</th>
<th>( x_1 ) (thickness of the air gap in the hot side)</th>
<th>( x_2 ) (thickness of the air gap in the cold side)</th>
<th>R (Thermal Resistance) Km²W⁻¹</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>-1</td>
<td>-1</td>
<td>1.606</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>-1</td>
<td>1.876</td>
</tr>
<tr>
<td>3</td>
<td>-1</td>
<td>1</td>
<td>1.897</td>
</tr>
<tr>
<td>4</td>
<td>1</td>
<td>1</td>
<td>2.220</td>
</tr>
<tr>
<td>5</td>
<td>-1</td>
<td>0</td>
<td>1.928</td>
</tr>
<tr>
<td>6</td>
<td>1</td>
<td>0</td>
<td>2.244</td>
</tr>
<tr>
<td>7</td>
<td>0</td>
<td>-1</td>
<td>1.894</td>
</tr>
<tr>
<td>8</td>
<td>0</td>
<td>1</td>
<td>2.262</td>
</tr>
<tr>
<td>9</td>
<td>0</td>
<td>0</td>
<td>2.282</td>
</tr>
</tbody>
</table>

**Table 2**

Summarized results of the statistical model and standard deviation.

<table>
<thead>
<tr>
<th>Response Average Standard Deviation Min Max</th>
<th>R</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.02319</td>
<td>0.23685</td>
</tr>
</tbody>
</table>

The results of the derived statistical model prepared in this study, and the residuals are given in Table 3 (Figure 8).

**Table 3**

Table of residuals: response Y1: R (Resistance).

<table>
<thead>
<tr>
<th>Simulation N°</th>
<th>R simulation</th>
<th>R calculated statistically</th>
<th>Residual</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.606</td>
<td>1.592384</td>
<td>0.013161</td>
</tr>
<tr>
<td>2</td>
<td>1.876</td>
<td>1.868884</td>
<td>0.007116</td>
</tr>
<tr>
<td>3</td>
<td>1.897</td>
<td>1.900218</td>
<td>-0.003218</td>
</tr>
<tr>
<td>4</td>
<td>2.220</td>
<td>2.229718</td>
<td>-0.009118</td>
</tr>
<tr>
<td>5</td>
<td>1.928</td>
<td>1.938398</td>
<td>-0.010398</td>
</tr>
<tr>
<td>6</td>
<td>2.244</td>
<td>2.241398</td>
<td>0.002862</td>
</tr>
<tr>
<td>7</td>
<td>1.894</td>
<td>1.914731</td>
<td>-0.020731</td>
</tr>
<tr>
<td>8</td>
<td>2.362</td>
<td>2.240648</td>
<td>0.129368</td>
</tr>
<tr>
<td>9</td>
<td>2.282</td>
<td>2.275994</td>
<td>0.007796</td>
</tr>
</tbody>
</table>

They were realized only 9 simulations (Figures 9 to 17), to obtain the statistical model, while the iso-curves could be obtained from the first six tests. In addition, reducing the number of simulations allows
us to generate an adequate statistical model that can give predictions via a framed estimation procedure. The accuracy of the statistical model was determined by comparing statistically predicted to simulated values obtained with different air gaps’ thicknesses prepared at the limits and the center of the simulation domain.

Figure 8: Statistical distribution of the statistical model’s responses in the simulation domain.

Figure 9: [Simulation No 1] / 1 cm air gap on the hot side and 1 cm air gap on the cold.

Figure 10: [Simulation No 2] / 5 cm air gap on the hot side and 1 cm air gap on the cold.

Figure 11: [Simulation No 3] / 1 cm air gap on the hot side and 5 cm air gap on the cold.

Figure 12: [Simulation No 4] / 5 cm air gap on the hot side and 5 cm air gap on the cold.

As it was resulted from the different simulation scenarios the maximum thermal resistance was calculated when the MTI product is put between two equal 3 cm air gaps as it is shown in the Figure 17.

Figure 13: [Simulation No 5] / 1 cm air gap on the hot side and 3 cm air gap on the cold.

Figure 14: [Simulation No 6] / 5 cm air gap on the hot side and 3 cm air gap on the cold.

Figure 15: [Simulation No 7] / 3 cm air gap on the hot side and 1 cm air gap on the cold.

Figure 16: [Simulation No 8] / 3 cm air gap on the hot side and 5 cm air gap on the cold.

Figure 17: [Simulation No 9] / 3 cm air gap on the hot side and 3 cm air gap on the cold.
Furthermore we expressed mathematically (Figures 18, 19) the overall thermal resistance of a composite wall including the MTI product with the thermo-physical characteristics mentioned in previous chapter as a function of the thicknesses of the two air gaps according to the following relation:

\[ R = 2.273994 + 0.151500 x_1 + 0.167167 x_2 - 0.184097 x_1 x_2 - 0.192097 x_2 x_2 + 0.013250 x_1 x_2 \]

**Figure 18:** 2D Graphic design - Variation of the variance function - in the simulation plan \( x_1 = d_1 \), \( x_2 = d_2 \).

**Figure 19:** 3D Graphic design - Variation of the variance function - in the simulation plan \( x_1 = d_1 \), \( x_2 = d_2 \).

The application of such statistical models to assist in the selection of the appropriate air gaps’ thicknesses in order to obtain the maximum thermal resistance of the composite wall is illustrated with the following isocurves (Figures 20, 21).

**Figure 20:** 2D Graphic design - Isocurves of the Variation of the response \( R \) (resistance) - in the simulation plan \( x_1 = d_1 \), \( x_2 = d_2 \).

**Figure 21:** 3D Graphic design - Isocurves of the Variation of the response \( R \) (resistance) - in the simulation plan \( x_1 = d_1 \), \( x_2 = d_2 \).

**CONCLUSIONS**

This work was based on the physical description of the heat transfer phenomena that guided us to conceive a numerical model. In the same time, a statistical model employing factorial design approach was created in order to express mathematically the influence of the air gaps’ thicknesses on the overall thermal resistance of a composite wall. As it was resulted from the simulations, when the 2 air gap’s thickness are both equal to 3 cm the resistance of the complex reaches its maximum. Overcoming this thickness on the warm or on the cold side has as result the reduction of the resistance because the convective conductivity of the air is increased inside the air gap. That also explains why the combination 3 cm–5 cm (simulation No 8) presents lower resistance than the combination 3cm-3cm (simulation No 9) but higher resistance than 5cm-3cm (simulation No 6). This happens because in simulation No 8 the higher than the critical (5 cm) air gap thickness was placed in the neighborhood of the coldest part of the complex.

**NOMENCLATURE**

\( R \) – thermal resistance \([m^2 K W^{-1}]\)

\( S \) – surface of the sample \([m^2]\)

\( T \) – temperature \([K]\)

\( t \) – time \([sec]\)

\( \rho_c \) – volumetric heat capacity \([J/(m^3 K)]\)

\( k \) – thermal conductivity \([W/(K m)]\)

\( x \) – the spatial coordinate through the thickness of the insulation complex \([m]\)

\( q_r \) – the total radiant flux in \( W/m^2 \) where \( q_r = F^+ - F^- \)
\( F^+, F^- \) the incident radiant fluxes developing to the hot and the cold side of the material respectively [W/m²]

\( l \) length of each section [m]

\( f \) solid fraction volume

\( R \) radius of the fiber [m]

\( h_{oe}, h_i \) equivalent heat transfer coefficients including convection and radiation heat transfer on the boundaries [W/(K m²)]

\[ \text{Greek symbols} \]

\( \Phi \) measured flow through the complex [W]

\( \Delta x \) distances between the interfaces of the finite volume [m]

\( \Delta T \) temperature difference on sample’s surface [K]

\( \delta x \) distances between the grid points [m]

\( \varepsilon \) emissivity

\( \beta \) absorption coefficient [m⁻¹]

\( \sigma \) Stefan-Boltzmann’s constant

ACKNOWLEDGEMENT

The first author would like to thank the “Alexander S. Onassis” public benefit foundation for the financial support during his PhD studies.

REFERENCES


LPRAI Company, NEMRODW software for design of experiments, 40 boulevard Icard, 13010 Marseille, France 2007.


