

# Numerical Modeling of Heat Transfer through Multilayer Walls of a Building, in Stationary and Non-stationary Regimes

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**Abstract** - In order to perform the calculations regarding the energy efficiency of a building, one of the stages is to determine the thermal resistance of the building elements. This quantity depends on the temperature difference and on the heat flow that crosses them. A possible approach to this calculation is the numerical one, where, depending on the existing or imposed by the regulations environmental conditions, these quantities can be accurately determined, whether the materials of these elements are homogeneous or inhomogeneous, linear or nonlinear. In this paper, the calculation method based on the numerical calculation using the finite volumes method is presented. As an example, the study focuses on the analysis of a multilayer exterior wall. The temperature conditions to which it is subjected correspond to the normal operating conditions during the winter, as they are provided by the regulations in force in Romania. The stationary case was firstly analyzed, considering the constant ambient temperatures inside and outside the building. Subsequently, a non-stationary thermal field analysis was performed, taking into account the variation in temperature over the course of a day, during the winter period. The algebraic equations resulting from the discretization of the problem equations were implemented in a program created in MATLAB. The obtained results were compared with those offered by a specialized calculation program for the thermal fields, which uses the finite element method, QuickField Professional. The calculation of errors proved a good correspondence between the obtained values.

**Cuvinte cheie:** pereți multistrat, transfer termic, metoda volumelor finite, MATLAB, performanța energetică a clădirii.

**Keywords:** multilayer walls, heat transfer, finite volume method, MATLAB, energy efficiency of the building.

## I. INTRODUCTION

The European Union has called for solutions to reduce energy consumption by 2020 [1]. One of the domains where this is necessary is related to the energy requirement for heating the buildings. In order to reduce this energy consumption, it can be acted in two directions: on the one hand, it is desired to increase the efficiency of the installations for heating the buildings and houses, irrespective of whether they are electrical installations or that use solid or gaseous fuels; on the other hand, the aim is to limit the loss of energy and heat.

Regarding the limitation of heat losses at the level of residential and industrial buildings, at national level a series of regulations have been formulated, too. Thus, the Romanian Ministry of Regional Development issued in

2017 an order that complements the order of the Minister of Transport, Construction and Tourism of 2007 specifying the minimum energy performance requirements to be taken into account when designing new buildings, as well as renovating existing buildings [2].

For residential buildings, the minimum energy requirements refer to the value of the corrected thermal resistance of each construction element that is part of the envelope of the building, but also to the global coefficient of thermal insulation and to the specific primary energy consumption from non-renewable sources for heating the building as a whole [2].

Referring further to the corrected thermal resistance of a construction element, it depends on the thermal resistance of each homogeneous or non-homogeneous layer in the composition of a construction element, on the surface thermal resistances, and on the reduction coefficient of the unidirectional thermal resistance [2-5]. The minimum corrected thermal resistance values are provided by norms [2,3].

By analogy with the electrical resistance, the thermal resistance can be determined as a ratio between the temperature difference and the heat flux [6]. The temperature variation and the heat flux can be determined by calculation. For flat or cylindrical body shapes, simple or more complex shapes, and for homogeneous or non-homogeneous materials, the recommended calculation method is the numerical one [7-12]. The bibliography in the field recommends the finite element method or the finite volume method [6,11,12].

In this context, this paper presents a numerical procedure for determining the temperature variations at the level of a multilayer wall. The numerical calculation is based on the finite volume method and it is detailed for both the stationary and the non-stationary regimes (for a variable outdoor temperature during a day). The obtained results are compared with those obtained numerically using a specialized calculation program for thermal fields, that uses the finite element method, QuickField Professional. The comparisons made and the calculation of errors proved the correctness of the created program. Later this program can be adapted for nonlinear behavior depending on the temperature of the layers of component materials, for a more realistic analysis and a more accurate calculation.

## II. GENERAL PRESENTATION OF THE PROBLEM

Most of the construction elements have a flat structure; one of the dimensions (thickness) is much smaller than the

other two (width and length). This consideration allows to consider a heat diffusion only in the direction of the minimum size, and the heat flux is plane-parallel [6] (fig.1).

At the exterior walls of a building, due to the temperature difference between the two – indoor and outdoor – environments, the heat transfer phenomenon appears. To ensure a thermal insulation structure, these are, in most cases, non-homogeneous flat walls, made of different materials [3]. To have thermal insulation, the wall as a whole must provide a high thermal resistance. Mainly their structure includes (fig. 1): plaster (1, 4), polystyrene insulation (2) and brick (3).

The heat flux is directly influenced by the thermal resistances of the layers of material it crosses. The value of the heat flow characterizes the level of the heat losses in the building during the cold season. Then the outside temperature is lower than the inside temperature. In addition, the temperature of the wall inside is lower than the temperature of the room, and the temperature of the wall outside is slightly higher than the temperature of the external environment; this can be explained by the existence of the convection phenomenon.

In order to analyze the temperature distribution at the level of an exterior wall, an analysis based on numerical calculation will be performed, that uses the finite volume method. Based on the above observations, the analysis will be one-dimensional (in the direction of axis  $0x$ ). The wall thickness  $d$ , resulting from summing the dimensions  $d_1, d_2, d_3$  and  $d_4$  of the considered component layers (fig. 1), was discretized using a uniform mesh (fig. 2). For each material in the multilayer wall, a uniform structure is considered, so that in this study the thermal conductivity  $\lambda$  has a constant value in each material.

If inside the wall the mode of heat transfer is made by conduction (it is considered that the wall does not include air layers between materials), outwards and inwards the phenomenon of convection appears. In these areas the heat flux depends on the air temperature.

For the air temperature inside the building (indoor temperature), a calculation value for the heated rooms recommended by the norms will be considered,  $T_{ind} = 20^{\circ}\text{C}$  [2,3].

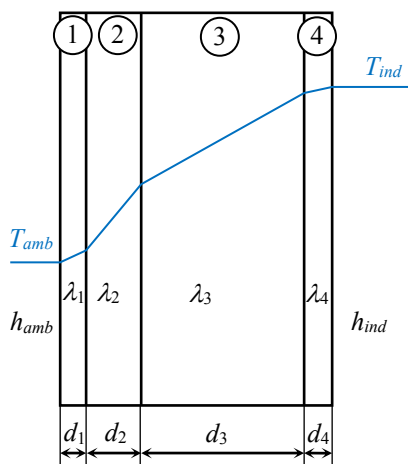


Fig. 1. Wall structure and its temperature distribution.

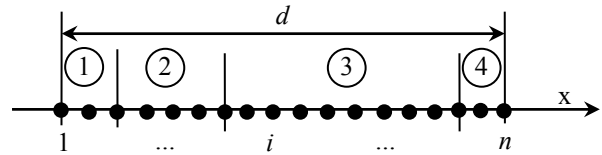


Fig. 2. Uniform mesh.

For the outdoor (ambient) temperature, the norms provide for the consideration of one of the four climatic zones related to Romania. For Dolj county, located in zone 2, the value of this temperature is given by the multiannual minimum temperature recorded in the area, i.e.  $T_{amb} = -15^{\circ}\text{C}$  [2,3].

Regarding the superficial heat transfer coefficient, the norms foresee that it is  $h_{ind} = 8\text{W/m}^2\text{K}$  for the interior and respectively  $h_{amb} = 24\text{W/m}^2\text{K}$  for the exterior.

#### A. Calculation of Temperature Distribution in Stationary Regime

The equation of thermal conduction in stationary regime, for a one-dimensional problem, is [10-12]:

$$\frac{d}{dx} \left( \lambda \frac{dT}{dx} \right) + S = 0, \quad (1)$$

where  $\lambda$  is the thermal conductivity of the material and  $S$  is the source term (the power generated per volume unit).

By integrating this equation on a control volume (finite volume) around an internal node  $i$  of the mesh (fig. 3), we obtain [12]:

$$\int_{i-1/2}^{i+1/2} \frac{d}{dx} \left( \lambda \frac{dT}{dx} \right) dx + \int_{i-1/2}^{i+1/2} S \cdot dx = 0, \quad (2)$$

respectively

$$\lambda_{i+1/2} \left( \frac{dT}{dx} \right)_{i+1/2} - \lambda_{i-1/2} \left( \frac{dT}{dx} \right)_{i-1/2} + S_{mean,i} (x_{i+1/2} - x_{i-1/2}) = 0 \quad (3)$$

where  $S_{mean,i}$  is the mean value of the source term on a control volume [12]. Each first order derivative is approximated by finite centered differences:

$$\lambda_{i+1/2} \frac{T_{i+1} - T_i}{x_{i+1} - x_i} - \lambda_{i-1/2} \frac{T_i - T_{i-1}}{x_i - x_{i-1}} + S_{mean,i} (x_{i+1/2} - x_{i-1/2}) = 0 \quad (4)$$

The relation (4) can be brought to the form:

$$-a_{i-1}T_{i-1} + a_iT_i - a_{i+1}T_{i+1} = b, \quad (5)$$

where the notations represent:

$$a_{i-1} = \frac{\lambda_{i-1/2}}{x_i - x_{i-1}}, \quad a_{i+1} = \frac{\lambda_{i+1/2}}{x_{i+1} - x_i}, \quad a_i = a_{i-1} + a_{i+1},$$

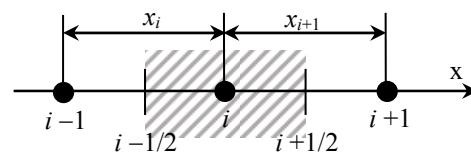


Fig. 3. Control volume around an internal node  $i$  of the mesh.

$$b = S_{mean,i}(x_{i+1/2} - x_{i-1/2}),$$

$$\lambda_{i-1/2} = \frac{2 \cdot \lambda_{i-1} \cdot \lambda_i}{\lambda_{i-1} + \lambda_i}, \quad \lambda_{i+1/2} = \frac{2 \cdot \lambda_i \cdot \lambda_{i+1}}{\lambda_i + \lambda_{i+1}}.$$

For the limit nodes 1 and  $n$  boundary conditions of convection type are set, having the general form [12]:

$$h_{air}(T_{air} - T_{node}) = -\lambda \cdot \frac{\partial T}{\partial x} \Big|_{node}. \quad (6)$$

For the node 1 the discretized equation becomes:

$$h_{amb}(T_{amb} - T_1) = -\lambda_{i+1/2} \cdot \frac{T_2 - T_1}{x_2 - x_1}. \quad (7)$$

This equation can be brought to the form:

$$a_1 T_1 - a_2 T_2 = b, \quad (8)$$

where the notations are:

$$a_2 = \frac{\lambda_{i+1/2}}{x_2 - x_1} = \frac{2 \cdot \lambda_2 \cdot \lambda_1}{\lambda_2 + \lambda_1} \cdot \frac{1}{x_2 - x_1},$$

$$a_1 = a_2 + h_{amb}, \quad b = h_{amb} \cdot T_{amb}.$$

For the node  $n$  the discretized equation becomes:

$$\lambda_{n-1/2} \cdot \frac{T_n - T_{n-1}}{x_n - x_{n-1}} = h_{ind}(T_{ind} - T_n). \quad (9)$$

This equation can be brought to the form:

$$-a_{n-1} T_{n-1} + a_n T_n = b, \quad (10)$$

where the notations are:

$$a_{n-1} = \frac{\lambda_{n-1/2}}{x_n - x_{n-1}} = \frac{2 \cdot \lambda_{n-1} \cdot \lambda_n}{\lambda_{n-1} + \lambda_n} \cdot \frac{1}{x_n - x_{n-1}},$$

$$a_n = a_{n-1} + h_{ind}, \quad b = h_{ind} \cdot T_{ind}.$$

The equations (5), (8), (10) form a system of equations which, implemented in a calculation program in Matlab, offers the temperature distribution in the external wall in stationary regime.

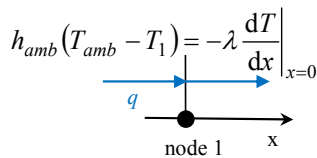


Fig. 4. Boundary condition of convection type for the node 1.

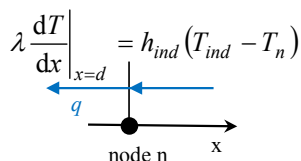


Fig. 5. Boundary condition of convection type for the node  $n$ .

## B. Calculation of Temperature Distribution in Non-stationary Regime

The equation of thermal conduction in non-stationary regime, for a one-dimensional problem, is [12]:

$$\rho \cdot c \cdot \frac{\partial T}{\partial t} = \frac{\partial}{\partial x} \left( \lambda \frac{\partial T}{\partial x} \right) + S, \quad (11)$$

where  $\rho$  is the mass density of the material and  $c$  is the specific heat at constant pressure.

By integrating this equation on a control volume (finite volume) around an internal node  $i$  of the mesh (fig. 3), and on the time interval  $[t, t + \Delta t]$ , we obtain [12]:

$$\int_t^{t+\Delta t} \int_V \rho \cdot c \cdot \frac{\partial T}{\partial t} dV dt = \int_t^{t+\Delta t} \int_V \frac{\partial}{\partial x} \left( \lambda \frac{\partial T}{\partial x} \right) dV dt + \int_t^{t+\Delta t} \int_V S \cdot dV \cdot dt. \quad (12)$$

The relation (12) becomes:

$$\begin{aligned} & \rho_i \cdot c_i \cdot (T_i^{k+1} - T_i^k) \cdot A \cdot (x_{i+1/2} - x_{i-1/2}) = \\ & = \int_t^{t+\Delta t} \left( \lambda_{i+1/2} \frac{T_{i+1} - T_i}{x_{i+1} - x_i} - \lambda_{i-1/2} \frac{T_i - T_{i-1}}{x_i - x_{i-1}} \right) \cdot A \cdot dt + \\ & + \int_t^{t+\Delta t} S_{mean,i} \cdot (x_{i+1/2} - x_{i-1/2}) \cdot A \cdot dt, \end{aligned} \quad (13)$$

where  $A$  is the area crossed by the heat flux.

For discretization, a total implicit scheme is chosen [12]. Thus,

$$\int_t^{t+\Delta t} T_p \cdot dt = T_p \cdot \Delta t, \quad (14)$$

and (13) becomes:

$$\begin{aligned} & \rho_i \cdot c_i \cdot (T_i^{k+1} - T_i^k) \cdot (x_{i+1/2} - x_{i-1/2}) = \\ & = \left( \lambda_{i+1/2} \frac{T_{i+1}^{k+1} - T_i^{k+1}}{x_{i+1} - x_i} - \lambda_{i-1/2} \frac{T_i^{k+1} - T_{i-1}^{k+1}}{x_i - x_{i-1}} \right) \cdot \Delta t + \\ & + S_{mean,i} \cdot (x_{i+1/2} - x_{i-1/2}) \cdot \Delta t, \end{aligned} \quad (15)$$

where  $T_i^k$  is the temperature in node  $i$  at the time  $t$ , and  $T_i^{k+1}$  is the temperature in node  $i$  at the time  $t + \Delta t$ .

This equation can be brought to the form:

$$-a_{i-1} T_{i-1}^{k+1} + a_i T_i^{k+1} - a_{i+1} T_{i+1}^{k+1} = b, \quad (16)$$

in which  $a_{i-1}$  and  $a_{i+1}$  have the same meaning as in (5), and supplementary the following notations were used:

$$a_i = a_{i-1} + a_{i+1} + \frac{\rho_i \cdot c_i}{\Delta t} \cdot (x_{i+1/2} - x_{i-1/2}), \quad \text{and}$$

$$b = \frac{\rho_i \cdot c_i}{\Delta t} \cdot (x_{i+1/2} - x_{i-1/2}) \cdot T_i^k + S_{mean,i} \cdot (x_{i+1/2} - x_{i-1/2}).$$

For the limit nodes 1 and  $n$  the discretized equation takes specific forms.

Thus for the node 1 the discretized equation is:

$$\begin{aligned} & \rho_1 \cdot c_1 \cdot (T_1^{k+1} - T_1^k) \cdot (x_{1+1/2} - x_1) = \\ & = \int_t^{t+\Delta t} \left( \lambda_{1+1/2} \frac{T_2 - T_1}{x_2 - x_1} + h_{amb}(T_{amb} - T_1) \right) \cdot dt + \\ & + \int_t^{t+\Delta t} S_{mean,1} \cdot (x_{1+1/2} - x_1) \cdot dt. \end{aligned} \quad (17)$$

It can be brought to the form:

$$a_1 T_1^{k+1} - a_2 T_2^{k+1} = b, \quad (18)$$

where  $a_2$  have the same formula as in (8), and

$$a_1 = a_2 + h_{amb} + \frac{\rho_1 \cdot c_1}{\Delta t} \cdot (x_{1+1/2} - x_1),$$

$$b = \frac{\rho_1 \cdot c_1}{\Delta t} \cdot (x_{1+1/2} - x_1) \cdot T_1^k + S_{mean,1} \cdot (x_{1+1/2} - x_1) + h_{amb} \cdot T_{amb}^k.$$

For the node  $n$  the discretized equation is:

$$\begin{aligned} & \rho_n \cdot c_n \cdot (T_n^{k+1} - T_n^k) \cdot (x_n - x_{n-1/2}) = \\ & = \int_t^{t+\Delta t} \left( h_{ind}(T_{ind} - T_n) - \lambda_{n-1/2} \frac{T_n - T_{n-1}}{x_n - x_{n-1}} \right) \cdot dt + \\ & + \int_t^{t+\Delta t} S_{mean,n} \cdot (x_n - x_{n-1/2}) \cdot dt. \end{aligned} \quad (19)$$

It can be brought to the form:

$$-a_{n-1} T_{n-1}^{k+1} + a_n T_n^{k+1} = b, \quad (20)$$

where  $a_{n-1}$  have the same formula as in (10), and

$$a_n = a_{n-1} + h_{ind} + \frac{\rho_n \cdot c_n}{\Delta t} \cdot (x_n - x_{n-1/2}),$$

$$b = \frac{\rho_n \cdot c_n}{\Delta t} \cdot (x_n - x_{n-1/2}) \cdot T_n^k + S_{mean,n} \cdot (x_n - x_{n-1/2}) + h_{ind} \cdot T_{ind}^k$$

The equations (16), (18), (20) form a system of equations which, implemented in a calculation program in Matlab, offers the temperature distribution in the external wall in non-stationary regime.

### III. RESULTS

For the analysis, an exterior wall of a building having the structure shown in fig. 1 was considered. The materials of this multilayer wall, the thickness of each layer and their properties are presented in table I. The building is located in Dolj county.

#### A. Temperature Distribution in Stationary Regime

Based on the system of equations presented above and considering the source term equal to zero (the wall does not generate heat), a calculation program was developed in Matlab that provided the temperature values at different points in the multilayer wall and that represented graphically the temperature distribution related to the analyzed wall. The mesh was considered uniform with 101 nodes.

Thus in fig. 6 thermal conductivity and temperature distribution in the wall in stationary regime are presented (with continuous lines).

TABLE I.  
THE MATERIALS IN THE WALL STRUCTURE AND THEIR PROPERTIES

Material	Plaster	Polystyrene	Brick
Thickness $d$ [m]	0.02	0.1	0.25
Thermal conductivity $\lambda$ [W/m·K]	0.9	0.04	0.7
Mass density $\rho$ [kg/m <sup>3</sup> ]	1700	20	1700
Specific heat $c$ [J/kg·K]	840	1460	870

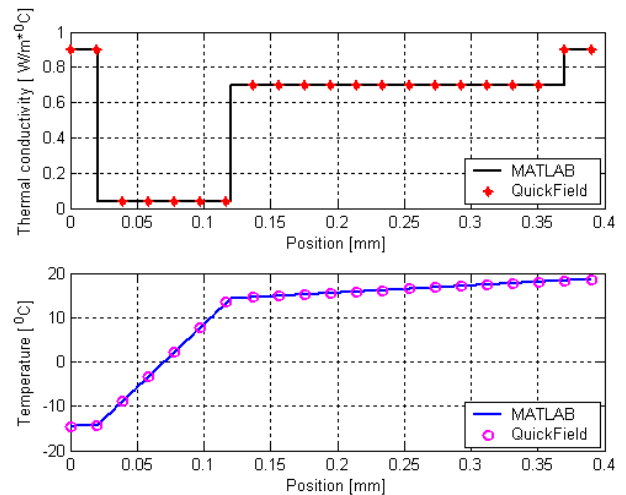


Fig. 6. Thermal conductivity (up) and temperature distribution (down) in the wall in stationary regime.

It is observed that, because the materials were considered homogeneous, isotropic and linear, the value of the thermal conductivity was constant in each component material area and therefore the temperature distribution is linear at the level of each layer. In the thermo-insulating layer of polystyrene, where the thermal conductivity has the lowest value, the temperature variation is the most noticeable (the highest gradient).

To verify the accuracy of the results obtained by modeling the thermal field using the finite volume method, the same problem was solved using a specialized software that uses the finite element method, QuickField Professional [13]. The results obtained in QuickField are shown in fig. 6 by dots. It is observed that the temperature variations obtained by the two methods overlap very well.

Taking advantage of the additional facilities of the QuickField program, in figures 7 and 8 the temperature distribution and the heat flux through the wall in stationary regime are represented.

Fig. 7 shows graphically, suggestively, the temperature distribution in the exterior wall during the cold season.

The analysis of fig. 8 indicates, as expected, a constant heat flux through the wall. It also indicates its orientation, from the inside of the building to the outside, suggesting the existence of heat losses.

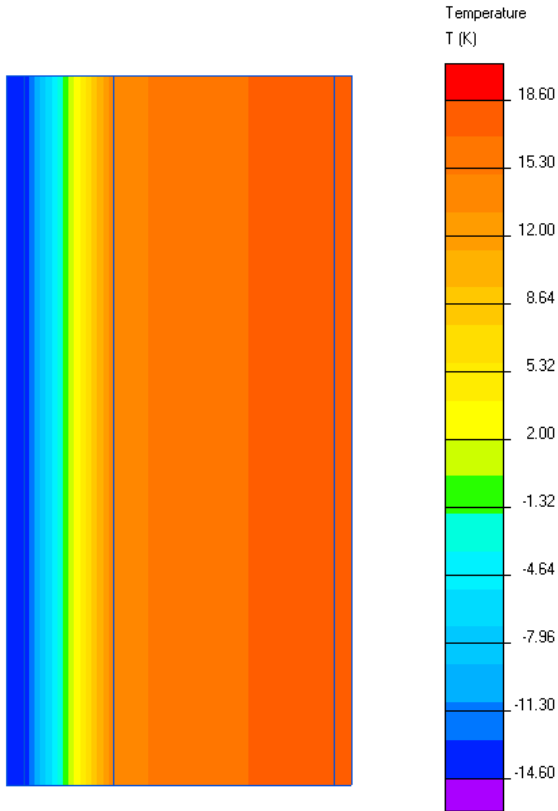


Fig. 7. Temperature distribution in the wall in stationary regime (QuickField).

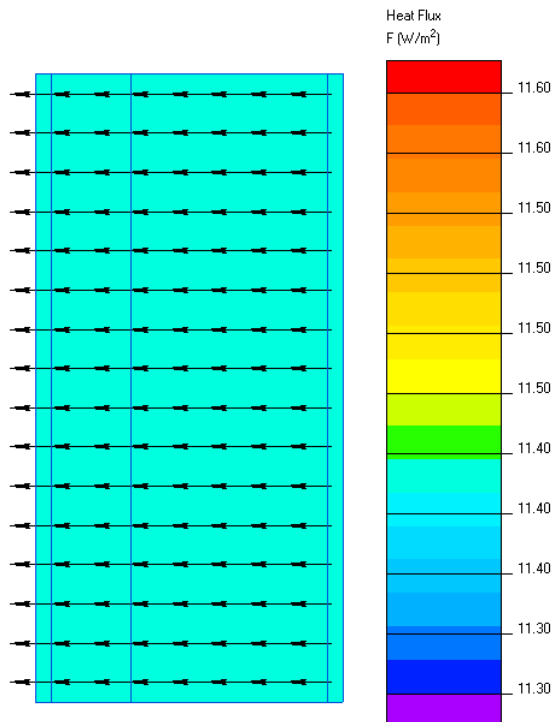


Fig. 8. Heat flux in the wall in stationary regime (QuickField).

### B. Temperature Distribution in Non-stationary Regime

Based on the system of equations presented above for the non-stationary regime and considering the source term equal to zero, a calculation program was developed in Matlab. It provides the temperature values at different points in the multilayer wall and at different moments of time, representing graphically the temperature distribution related to the analyzed wall.

As non-stationary regime, we considered a daily variation of the ambient temperature  $T_{amb}$  in cold season having an approximately sinusoidal shape [11]. It can be seen in fig. 9.

The initial values considered for the temperature of each node are those corresponding to the stationary regime. The temperature variation in the exterior wall will be analyzed for a period of 24 hours. The time step  $\Delta t$  was chosen of 60 seconds.

The results obtained using the proposed algorithm implemented in Matlab are presented in figures 10 and 11 (with continuous lines).

Fig.10 shows the temperature distribution in the wall in non-stationary regime, for different moments in time during a day:  $t = 0$ ,  $t = 2$  hours,  $t = 5$  hours,  $t = 12$  hours,  $t = 15$  hours and  $t = 20$  hours. It is noted that the largest temperature variations during a day are recorded in polystyrene, which indirectly influences the temperature of the outer plaster, too. In the brick and in the interior plaster, the temperature variations are imperceptible. In conclusion the insulation of polystyrene fulfills its role.

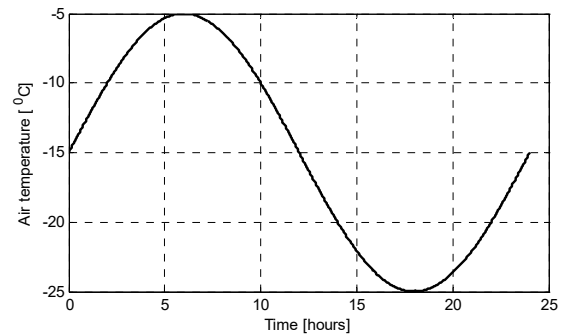


Fig. 9. Air ambient temperature  $T_{amb}$  variation during a day in the cold season.

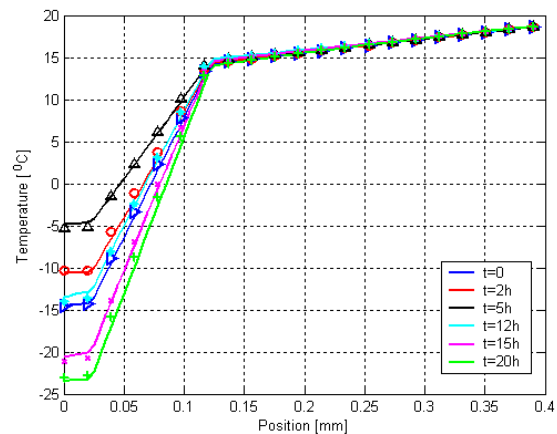


Fig. 10. Temperature distribution in the wall in non-stationary regime, for different moments in time.

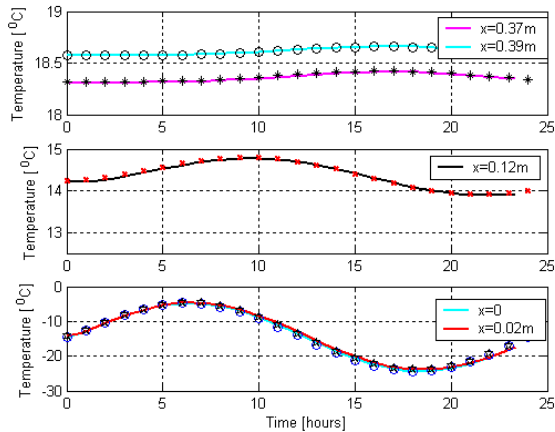


Fig. 11. Temperature-time variations in different points in the wall in non-stationary regime.

In fig. 11 the temperature variations in time in different points in the wall are presented. We chose the points related to the boundary nodes ( $x = 0$  and  $x = 0.39$  m), as well as points on the separation surfaces between materials ( $x = 0.02$  m,  $x = 0.12$  m and  $x = 0.37$  m).

It is observed that, at all points, the temperature varies sinusoidal during a day, with higher amplitudes in the outer plaster and in the polystyrene, and with very small amplitudes in the analysis points from the interior part of the building wall. Further, there is a change in the phase of these sinusoids, those related to the points from the interior part of the building wall being delayed compared to those from the exterior part. This observation is consistent with the results presented in the literature [11].

To verify the accuracy of the results obtained by modeling the thermal field using the finite volume method, the

same problem was solved using the specialized software QuickField Professional [13]. The results obtained in QuickField are shown in figs. 10 and 11 by dots. One can observe that the temperature variations obtained by the two methods overlap very well.

Taking advantage of the facilities of the QuickField program, in fig. 12 the temperature distribution in the wall in non-stationary regime is represented. The representations were made for three moments in time:  $t = 2$  hours,  $t = 5$  hours,  $t = 12$  hours. We can observe the change in temperature values especially in the external part of the wall, being influenced by the variation of the exterior (ambient) temperature during the day.

Fig.13 represents the temperature variations in time in different points in the wall obtained in QuickField Professional. We chose the same points as in the Matlab analysis (fig. 11). It is observed, once again, the similarity of the results obtained in the two analyzes, that use finite volume method and finite element method respectively.

For a more suggestive analysis of the results, a calculation of errors was made, both in stationary and non-stationary regime. The relative error was calculated between the results obtained in Matlab and those offered by QuickField Professional. Representative samples were taken from the entire thickness of the wall, at different moments. The comparative results are presented in Table II. There is a tendency to have increase values of the errors as time increases. This is explained by the iterative calculation process that make the calculation errors to accumulate. However, the highest obtained relative error does not far exceed the value of 3%, which indicates a very good correspondence between the obtained results in Matlab and QuickField. The errors can be reduced by considering a more refined discretization mesh (with a greater number of nodes).

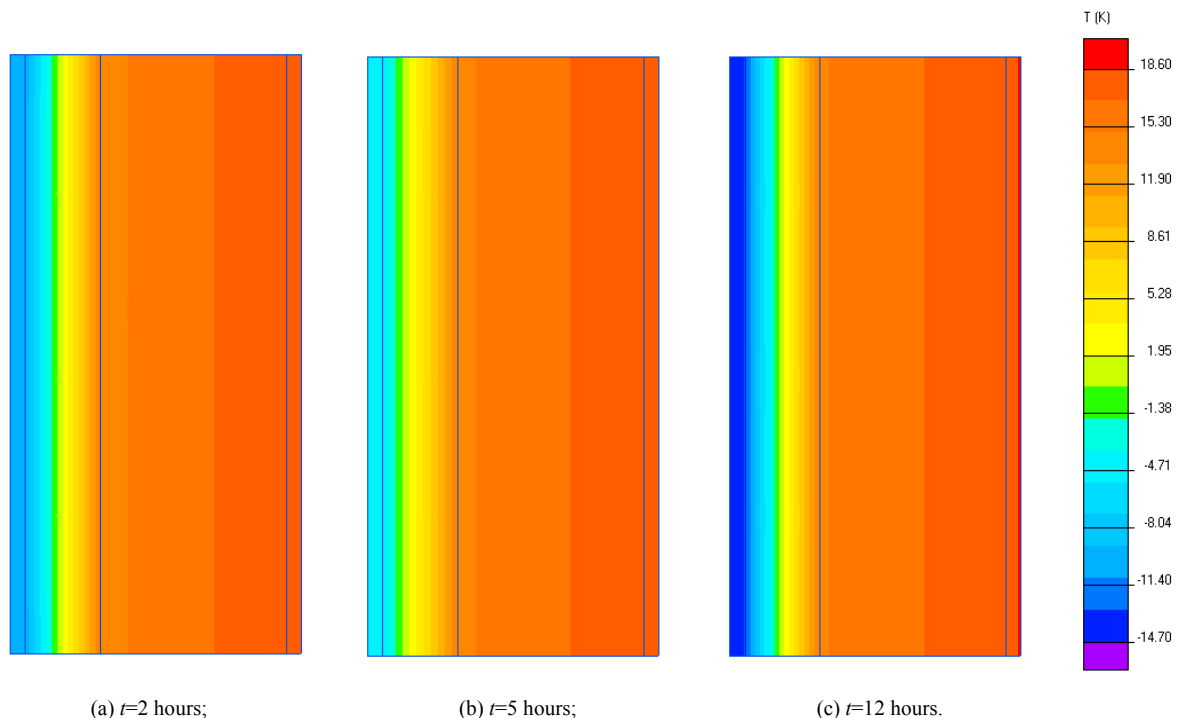


Fig. 12. Temperature distribution in the wall in non-stationary regime (QuickField).



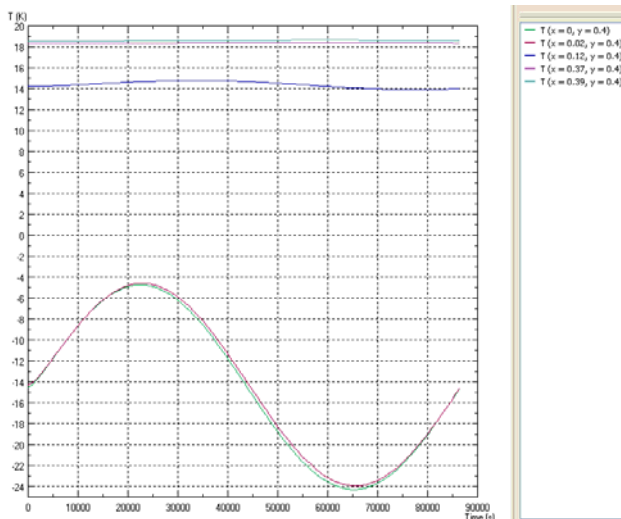


Fig. 13. Temperature variations in time in different points in the wall in non-stationary regime (QuickField).

TABLE II.  
ERROR CALCULATION, FOR STATIONARY AND NON-STATIONARY REGIMES

Point position $x[m]$	Relative error [%]			
	Stationary	Non-stationary		
		After 60 seconds	After 2 hours	After 12 hours
0	0	-0.007	-1.201	-1.392
0.019	0	-0.010	-1.217	-1.325
0.039	0.769	0.275	-0.966	-1.028
0.078	-3.002	-0.970	0.351	0.627
0.117	-0.508	-0.146	1.070	1.116
0.156	-0.006	-0.052	1.377	1.420
0.195	-0.005	-0.041	1.441	1.476
0.234	-0.005	-0.030	1.505	1.532
0.273	-0.005	-0.021	1.570	1.589
0.312	-0.005	-0.012	1.635	1.648
0.351	-0.005	-0.004	1.700	1.708
0.390	0	-0.016	1.754	1.760

#### IV. CONCLUSION

This paper focuses on obtaining the temperature distribution in a wall, as part of the calculation to determine the energy efficiency of a building. A numerical calculation procedure based on the finite volume method was presented. The obtained formulas can be easily implemented later in any computing program. Moreover it shows how the problem can be solved using a specialized software for the thermal field. The analysis was done for both the stationary regime and the non-stationary regime, with the mention that it is recommended that the initial data of the non-stationary regime be those of the stationary problem, solved previously. The elaborated programs will be able to be developed later to take into account the temperature

dependence of some material properties, and thus to obtain results as close as possible to the practice.

#### ACKNOWLEDGMENT

**Source of research funding in this article:** Research program of the Electrical Engineering Department financed by the University of Craiova.

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First author – 60%

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Received on July 19, 2019

Editorial Approval on November 15, 2019

#### REFERENCES

- [1] M. Embaye, R.K. Al-Dadah, and S. Mahmoud, "Effect of flow pulsation on energy consumption of a radiator in a centrally heated building", in *International Journal of Low-Carbon Technologies* Advance Access, pp.1-11, September 2014.
- [2] The Order no. 2641/2017 regarding the modification and completion of technical regulation "Calculation Methodology of the Energy Efficiency of the Buildings", Romania, 2017.
- [3] F. Mateescu, *Thermal Insulation of Houses (Izolarea termica a locuintelor)*, Bucharest: M.A.S.T., 2007.
- [4] U. Berardi, "The impact of temperature dependency of the building insulation thermal conductivity in the Canadian climate", *11<sup>th</sup> Nordic Symposium on Building Physics*, 11-14 June 2017, Trondheim, Norway, *Energy Procedia*, no. 132, pp.237-242, 2017.
- [5] C. Schumacher, L. Ricketts, G. Finch, and J. Straube, "The effect of temperature on insulation performance: considerations for optimizing wall and roof designs", *31st RCI International Convention and Trade Show*, March 10-15, 2016, Orlando, FL, pp.165-173.
- [6] V. Dolga, *Fundamentals of Mechanical Engineering (Fundamente de inginerie mecanica)*, course, Politehnica University Timisoara, 2013.
- [7] M.I. Neaca, "The resistive tubular heater - Mathematical and simulation model", *Annals of the University of Craiova, Electrical Engineering series*, no. 42, issue 2, pp.71-80, December 2018.
- [8] M.I. Neaca, A.M. Neaca, "Estimation of the heat cumulated inside a wall with cylindrical symmetry", *Annals of the University of Craiova, Electrical Engineering series*, no.39, p.189-193, 2015.
- [9] C.F. Ocoleanu, G. Manolea, I. Popa, "Experimental validation of numerical results corresponding to catenary contact wire temperature", *Annals of the University of Craiova, Electrical Engineering series*, no. 35, p.152-157, 2011.
- [10] L. Gavrilă, *Transfer Phenomena (Fenomene de transfer)*, vol. I-II, Bacau : Alma Mater, 2000.
- [11] F. Iordache, F. Baltaretu, B. Caracaleanu, *Modeling and Simulation of Dynamic Processes of Heat Transfer (Modelarea si simularea proceselor dinamice de transfer termic)*, Bucharest: MatrixRom, 2003.
- [12] I. Popa, *Numerical Modeling of Heat Transfer - Finite Volume Method (Modelisation numerique du transfert thermique-Methode des volumes finis)*, Craiova: Editura Universitaria, 2002.
- [13] [www.quickfield.com/free\\_doc.htm](http://www.quickfield.com/free_doc.htm).