THE NUMERICAL SIMULATION OF THE CULTURE MEDIUM FOR THE PLEUROTUS GENUS MUSHROOMS

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Abstract – **The ultimate goal of this article is the realization of an automaton system to monitor and control the microclimate of the mushroom culture. In this respect, we have considered it necessary to study the technological culturing of the Pleurotus genus mushrooms and to numerically simulate the culture medium for the horticultural products in order to identify the best solutions concerning the control of the culture medium, and to remove the carbon dioxide resulting from the fructifying process of the mushrooms.**

Keywords: *air refreshing, quadripole, airing, conduction and convection constants.*

1. INTRODUCTION

The recycling of the air inside the workshop is the effect of letting a fresh draught of air waft in, through the air thick with noxious emissions (especially CO_2), which heads to raising the air quality in the areas cultivated with mushrooms. The fresh air (clean or cold) at the mushroom growing room's entrance causes the movement of the stale air as well, which is also known in the act of ventilation as *carrying* air. Ventilation is possible whenever there appears a pressure difference between the air in the room (larger room or rooms adjacent to the mushroom growing room) and the air outside the room because of the existing heat source, which makes the temperature inside the room higher then outside, respectively the specific weight of the air inside is lower than that of the air outside. Under these circumstances, ventilation can be realized either by mechanical devices (ventilators), or in a natural way [8], [13].

2. THE PROCESS OF MUSHROOM CULTIVATION

2.1. Pleurotus Genus – general characteristics

These culture mushrooms include more than one spectres: Pleurotus ostreatus, Pleurotus florida, Pleurotus cornucopiae, Pleurotus sajor-caju, etc., which taxonomical may by included in the FUNGI kingdom, BASIDIOMYCOTA phylum, BASIDYOMICOTINA subphylum, BASIDIOMYCETIDAE class, AGARICOMYCETIDAE subclass, ARARICALES order, PLEUROTACEAE family, PLEUROTUS genus.

2.2 The Phenophases of growing

The incubation of the mycelium can be defined as the period from the moment of sowing to the appearance of the first fructifications, and this phenomenon manifests itself starting with the first day after sowing. Mycelium maturing or the fructification induction represents the period between the end of the incubation and the apparition of the first fructification, and it lasts between 5 and 10 days, depending on the species. The fructification begins as soon as the fructification primordials appear, and it takes place during several "waves of harvesting", depending on the species. As far as Pleurotus species is concerned, the period from the incubation time to the beginning of the harvesting is from 15 to 35 days. The phenophases in their chronological order from the appearance of the primordial are [3], [4]:
- the primordial phase lasts from 5 to 7 days, and it includes the fructification induction;
- the cornet phase (youth phase) manifests itself in 1 to 2 days since the youth cornet phase, depending on the temperature of the culture plot;
- the convex margin phase – the convex margin (oriented downwards) manifests itself from 1 to 2 days since the cornet youth phase, depending on the temperature of the culture plot;
- the horizontal margin phase (straight) appears from 1 to 2 days since the convex margin phase, and it marks the beginning of the harvesting period;
- the concave margin phase – the concave margin (oriented upwards) appears very quickly from ½ to 1 day since the straight margin phase;
- the turned up margin phase appears very quickly from ½ to 1 day since the anterior phase, and it is a physiological phenomenon which manifests itself in the turning up of the mushroom top;
- the cornet old age phase manifests itself from ½ to 1 day since the turned margin phase, and it develops very quickly.

2.3 Temperature, humidity and ventilation requirements

Pleurotus ostreatus. The incubation temperature is of 22-24 ^{o}C and takes place during 20 days, followed by a period of 10 day thermal shock when temperature falls down with 8-10 ^{o}C as compared to the

incubation phase. The fructification temperature is of 10 to 16 ^oC, with a limit of 18 ^oC above which the mushrooms will not fructify any more. The relative humidity in the fructification zone is of 80-85 %. The illumination intensity ranges from 100 to 150 luces – a fluorescent tube placed at a distance of 8-10 metres.
Pleurotus florida. The incubation temperature ranges between 22-24 ^oC in winter and 18-20 ^oC in summer, and it is reached in 15-20 days. The fructification temperature is of 22-24 ^oC. The relative humidity in the fructification zone is of 80-85 %. The pretensions concerning the airing are greater in comparison with Pleurotus ostreatus, as an airing of 150 m^3 air/hour/ton of cellulosed blend is required.
Pleurotus comucopiae. The incubation temperature is of 22-24 °C in winter and 18-20 °C in summer, and it is reached in 15-20 days. The fructification temperature is of 22-23 °C. It requires an airing of 150 m^3 air/hour/ton of cellulosed blend. The relative humidity in the fructification zone is of 90-95%.
Pleurotus sajor-caju. The incubation temperature of 22-24 ^oC must be provided for 12-16 days. The fructification temperature is above 25 ^oC, fructifying being possible at temperatures lower than 17-20 ^oC as well. An important airing is necessary, respectively 200 m^3 of air/hour/ton of cellulosed blend. The relative humidity in the fructification zone is of 70-75%.

2.4. The Culture Plots for Pleurotus Mushrooms

The culture plots used in the intensive system presume a solid concrete, brick or stone building characterized by good tightness in order to ensure and control the specific microclimate conditions throughout the whole year. The intensive culture presumes larger surfaces as compared to the classic ones in order to benefit from facilities, in accordance with the settled type of culture, and from installations that could permit the automated, even computerised control of the specific microclimate, thus cutting down on the production expenses to a greater extent [5]. Irrespective of the system and type of mushroom culture, the rooms should meet some obligatory requirements [14]:

- they should be naturally illuminated trough windows or artificially illuminated by placing, especially fluorescent tubes, or/and bulbs with mercury vapors (cold light);
- they should offer the possibility of maintaining a temperature as constant as possible, in winter of 14-20 ^oC, and in summer the temperature should not be higher than 30 ^oC;
- they should have water source or running installations that could have a flow of 60-80 l/m^2/culturecycle;
- the airing should be done by free ventilation (doors, windows) for small spaces, and by controlled ventilation (ventilators) for surfaces that are larger than 10 m^2/culture, providing, in the period of fructification, the change of air on an 8-10 time proportion/hour, to eliminate the carbon dioxide and the mushroom spores disseminated in the air.

3. ELABORATING THE MATHEMATICAL PATTERN FOR THE STUDY OF THE REFRESHING PHENOMENON

3.1. The identification of the air refreshing phenomenon in the mushroom growing room

Let us consider that there is a certain amount of pollutant emission in the room (fig. 3.). Putting forward the hypothesis that this emission is constant in time, as well as the concentration of the same noxiousness in the let in air, and the ventilation is uniform everywhere inside the room and there are no other emissions (of temperature or humidity), the differential equation of mass balance sheet is [1], [2]:

$$\left[D \cdot c_r + D_{1m} - D \cdot c(t)\right] dt = V dc \qquad (1)$$

where: D is the volume flow of displaced air, respectively let in and out of the room, in $m^3 \cdot s^{-1}$; D_{1m} is the pollutant mass flow (noxious substance – CO_2) emitted in the room, in $kg \cdot s^{-1}$; c_r is the concentration of the pollutant in the air inside the room, variable in time, in $kg \cdot m^{-3}$; t is the time, in s; V is the volume of the room, in m^3.
Because $d\left[D \cdot c_r + D_{1m} - D \cdot c(t)\right] = -D \cdot dc$, the equation (1) may be written under the form:

$$\frac{dt}{V} = -\frac{1}{D} \frac{d\left[D \cdot c_r + D_{1m} - D \cdot c(t)\right]}{D \cdot c_r + D_{1m} - D \cdot c(t)} .$$

Integrating the equation, we may obtain successively

$$\frac{1}{V}\int_0^t dt = \frac{1}{D}\int_{c_0}^{c} \frac{d\left(D \cdot c_r + D_{1m} - D \cdot c\right)}{D \cdot c_r + D_{1m} - D \cdot c} \Leftrightarrow$$

$$e^{\frac{D}{V}t} = \frac{D\left[c_0 - c_r\right] - D_{1m}}{D\left[c - c_r\right] - D_{1m}} , \qquad (2)$$

where: c_0 is the value of the pollutant concentration in the air inside the room, at the time t=0, and c is the value of concentration, at time t.
Writing the ratios as: $\frac{V}{D} = T_0$ - the hour number of air changes and $\frac{D_{1m}}{V} = d_s$ (ds, the specific flow of the emitted noxious substance), the equation (2) becomes

$$e^{\frac{t}{T_0}} = \frac{c_0 - c_r - d_s T_0}{c - c_r - d_s T_0}. \qquad (3)$$

From the equation (3) the final concentration of the pollutant in the air inside the mushroom growing room results

$$c = \left(c_r + d_s T_0\right)\left(1 - e^{-\frac{t}{T_0}}\right) + c_0 e^{-\frac{t}{T_0}}. \qquad (4)$$

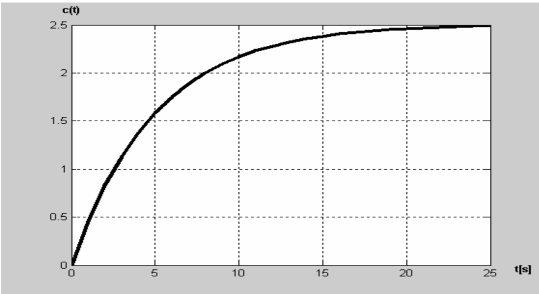

Figure 1 The variation in time of the carbon dioxide in the air, in a certain zone of the room for T_0=5s, d_s=0,0015 kg/m^{3}s, c_{adm}=3kg/m^3, when in the displaced air, there is some noxious substance.

Subsequently there comes a presentation of a method of identifying the concentration value in a place inside the room, when the air comes in as the effect of the superposition of a circulation by convection (artificial) and of a diffusion and of its recirculation, which operates as a reaction circuit [7], [13].
The resulting pattern helps the understanding of the way the recirculation phenomenon takes place, and it can be the basis of a future adaptive adjustment of a concentration value.

3.2. Patterning the recirculation of the fluid by convection and diffusion

Let us consider that the room is constituted of a volume where the fluid flows by convection, in series with a volume where the flow is a perfect blend (by diffusion), having the time constant T_0. In fig. 2. time t_{rc} is defined as the minimum period of time it takes a fluid particle, heaving P as point of departure, to reach again P point, by recirculation. In what follows, the answer in the M stationary point will be determined, as a result of applying in I point an impulse including m mass of experimental fluid (indicator fluid). The answer will be calculated in time intervals:
$\left[t_{tr}, t_{tr} + t_{rc}\right], \left[t_{tr} + t_{rc}, t_{tr} + 2t_{rc}\right]....$

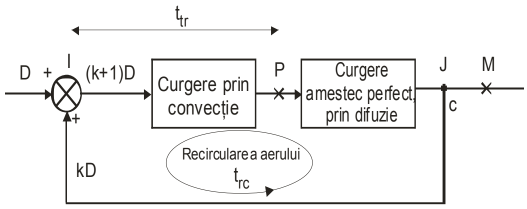

Figure 2 Explaining the fluid recirculation

The numerical model (MATLAB/ Simulink) [6], [15] of the air recirculation phenomenon by artificial convection and diffusion (taking into consideration the way the process of recirculation takes place) is illustrated in fig. 3. , where the Laplace transformed was applied to the exponential that defines the volume of flow by diffusion.

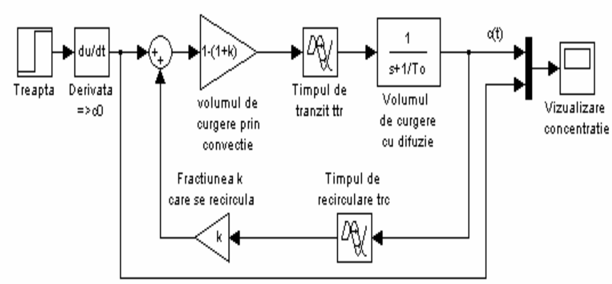

Figure 3 Block structural scheme, in Simulink, of the recirculation phenomenon

The approximate aspect of the concentration curve $c_{(t)}$, illustrating the response to the impulse (in a point on the recirculation circuit), between T_0 and t_r+2t_{rc} (after the three recirculations of the pollutant concentration is represented in fig. 4.

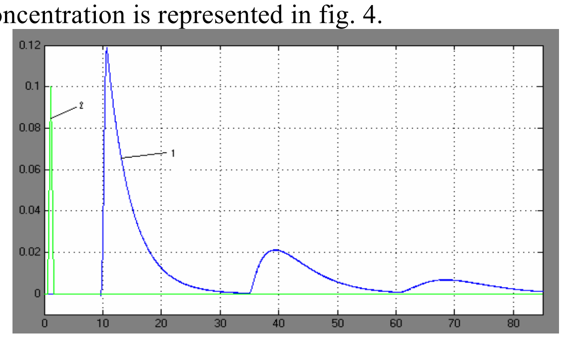

Figure 4 The aspect of the recycled instantaneous concentration response (curve 1), when applying an impulse (c_0 = 1, k = 0.75, curve 2 – scale 1:10) when the concentration of the pollutant is null in the flow volume – for t_{tr} = 10s, t_{rc} = 25s, T_0 = 5s.

With the help of this modeling, one can notice that it is possible to characterize the circulation phenomenon of the air in a certain place in the air conditioned room, by a curve typical of the response to the impulse. On the other hand, one may notice that the time interval that separates two successive peaks of the response is approximately constant (≈ 30s). A necessary condition for the time interval to be constant is that the concentration should be null between the two successive peaks of the response impulse.

In fig. 5. it is shown the graphical aspect of the response to the impulse, if the concentration of the pollutant is not null in the flow volume, that is, that at a certain moment recirculation can be felt. One may observe that for two successive recirculations, one may obtain low values of the pollutant concentration of the air in the room by means of:

- Decreasing the time interval between a sequence of 2 peaks (fig. 5);
- Leveling the general form of the response, but the presence of the peaks will be difficult to support [10].

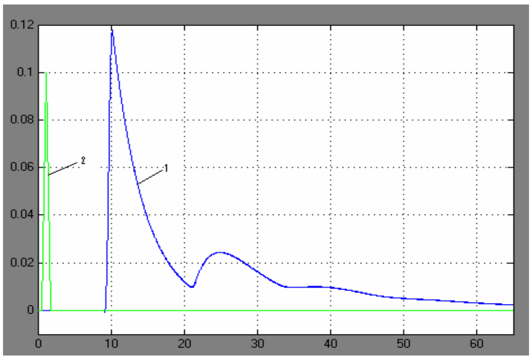

Figure 5. The graphical aspect of the instantaneous recycled concentration (curve 1), when applying an impulse ($c_0 = 1$, $R = 0.75$, curve 2 – scale 1:10) the moment the pollutant concentration is not null – for $t_{rc} = 75$.

4. MODELLING OF THE MICROCLIMATE IN CASE OF THE EXISTENCE OF TERMIC EMISSION

4.1. The model of the inside as a quadripole

When patterning the room, there occur difficulties in determining the income of exterior heat, because of the unsteady regime of the heat transmission. The recirculation of the air in a room with thermal emissions may be considered to be similar to a pipe through which the same air flow circulates (warm inside it and cold outside it). Presupposing that there is some heat emission in the room, for an elementary volume of fluid, moving monodimensionally in the x direction (alongside the interior pipe), Newton's second low of dynamics and the continuity equation are written [12], [13]:

$$\frac{\partial H}{\partial x}+\frac{1}{gS}\frac{\partial D}{\partial t}+\frac{k_f D|D|}{2gZS^2}=0 \quad \frac{\partial D}{\partial x}+\frac{gS}{v^2}\frac{\partial H}{\partial t}=0 \tag{5}$$

where: v is the speed of the air in $m\cdot s^{-1}$; D is the volume flow, expressed in m^3/s; t is the time in s; Z is the pipe diameter, in m; S is the area of the cross section of the pipe, expressed in m^2; H is the total pressure (static and dynamic), expressed in m, given by the relation $H=\frac{p}{\rho g}$, where: p is the total pressure expressed in $N\cdot m^{-2}$. The friction coefficient K_f varies with the flow regime (therefore, with Reynolds' Re number), and it can be calculated:

- for laminar flow ($R_e < 2000$), using Poiseuille's formula [1], [2];
- for turbulent flow ($R_e > 3000$), using Colebroak-White's formula [2]. Be H_0 and D_0 the absolute values of the total pressure and of the flow in a section of the outside pipe, and

$h=\frac{H}{H_0}$, $d=\frac{D}{D_0}$, the relative coordinates, then we obtain:

$$\frac{\partial h}{\partial x}+\frac{D_0}{H_0 gS}\frac{\partial d}{\partial t}+\frac{k_f D_0^2}{2gZS^2H_0}d|d|=0,$$

$$\frac{\partial d}{\partial x}+\frac{H_0 gS}{D_0 v^2}\frac{\partial h}{\partial t}=0, \tag{6}$$

and after that, making the discrete results of the equations:

$$\frac{\partial h}{\partial x}\to\frac{\Delta h}{\Delta x},\quad \frac{\partial h}{\partial t}\to\frac{dh}{dt},\quad \frac{\partial d}{\partial x}\to\frac{\Delta d}{\Delta x},$$

$\frac{\partial d}{\partial t}\to\frac{d(d)}{dt}$, we may obtain :

$$\Delta h=-\frac{D_0\Delta x}{gH_0S}\frac{d(d)}{dt}-\frac{k_f\Delta x D_0^2}{2gZS^2H_0}d|d|,$$

$$\Delta d=-\frac{H_0 gS\Delta x}{D_0 v^2}\frac{dh}{dt} \tag{7}$$

Be the ration

$\frac{D_0\Delta x}{gH_0S}=L$, $\frac{k_f D_0^2\Delta x}{2gZS^2H_0}=R$, $\frac{H_0 gS\Delta x}{D_0 v^2}=C$,

equations (7) for a discrete flowing fluid volume become:

$$\Delta h=-L\frac{d(d)}{dt}-Rd|d|,\quad \Delta d=-C\frac{dh}{dt}, \tag{8}$$

and they represent the basic equations of the T shaped (quadripole) electric circuit in fig. 6. The block structural scheme of the room ventilation, in the case of heat emissions, illustrated in fig. 7, corresponds to the electric circuit in fig. 6.

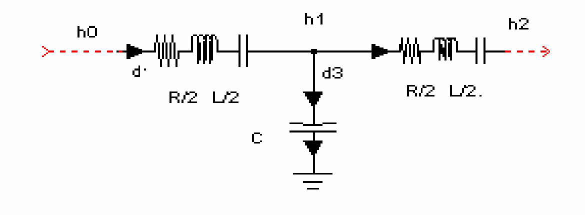

Figure 6 The scheme of the electric circuit analogous to the recycling of a volume of fluid

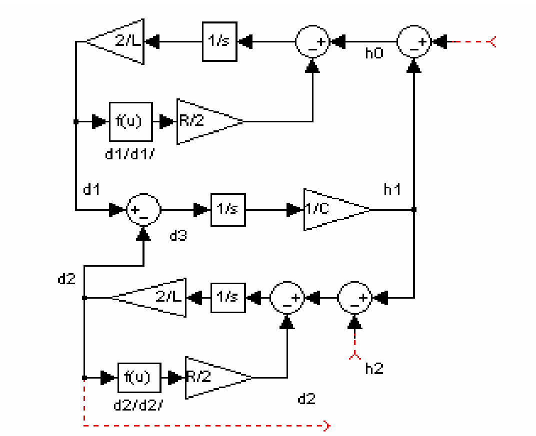

Figure 7 The block structural scheme for recycling air in Simulink

The T shaped electrical circuit connection to a source that develops a variable voltage (fig. 8.) is analogous to the room ventilation. The efficiency of the ventilation can be established by the constant $T_c = \frac{H_0 S}{D_0}$, where, S_c represents the cross section area that was traversed by the discrete volume of the fluid.

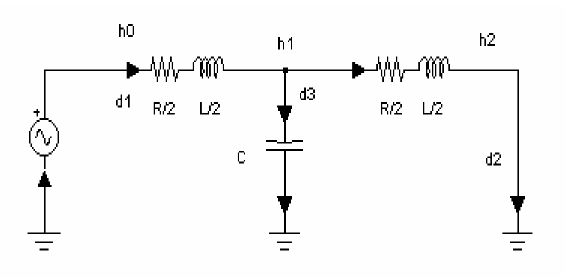

Figure 8 The electrical circuit analogous to the ventilation of the room by natural convection

As in the air – conditioning technique, in rooms with or without heat emissions, the recalculating processes are considered isobaric (because of the fact that the pressure variations as compared to the atmospherically pressure are insignificant), there has been no stress on the illustration of some results of the inside pattern as quadripole. Nevertheless, in the instance of turning on the air ventilation there have been obtained results (9), which highlighted pressure variations (with a negative impact on the equipment or on occupied area) inside the room. This phenomenon is diminished if the ventilators are turned on slowly with the help of an adjustable speed (inverter). The recirculation of the air by means of a VE ventilator (patterned as a source of constant pressure), controlled by an inverter INV which receives information from a pressure transducer TP, placed in the room, is analogous to the behavior of the electrical circuit shown in fig.9

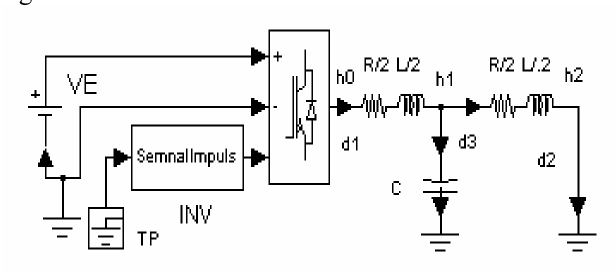

Figure 9 The electrical circuit analogous to the ventilation of the room by means of a controlled ventilator

4.2. Modeling the thermal transfer in the culture room and hall from the outside or the adjacent rooms

For a room that has a one layer structure of a small surface and lower windows panes (in the case of the mushroom growing room – there is none), Qcd representing the amount of heat that goes through the wall of the room, and presupposing that the heat transfer will occur by conduction from the outside to the inside, then the thermal equivalent resistence of the building Rt is obtained on the bases of the stationary regime of thermal transfer (Iourier's low).

$$Q_{cd} = \frac{\theta_e - \theta_i}{R_f} + \frac{\theta_e - \theta_i}{R_z} = \frac{\theta_e - \theta_i}{R_t} \text{ [W]} \qquad (9)$$

where: θ_e, θ_I are the temperatures of the outside and inside air; R_f and R_z represent the thermal resistances of the windows (if they exit), respectively of the room's walls, given by the relations $R_f = \frac{z_f}{\lambda_f S_f}$, $R_z = \frac{z_z}{\lambda_z S_z}$, where: Z_f, Z_z are the breadths of the windows, respectively the breadths of the walls of the room, in m; λ_f, λ_z are the thermal conductivities of the materials, in $Wm^{-10} \cdot C^{-1}$; S_f, S_z are the arias of the interior sections of the windows, respectively the walls, normal in the direction of the thermal flow, expressed in m². Expressing the C_t capacity of the air, as a product between the mass m of the air existing in the room and the specific heat of the air inside c_{si}, then the equation of the mass balance sheet, in the hypothesis of the existence of the thermal incomes from the adjacent rooms Q_1, of the thermal emissions produced by evaporation Q_2, or the air heating Q_3, or resulting from other sources (illuminated Q_4, electrically operated Q_5, the warm solid or liquid surfaces Q_6), is [11]:

$$C_t\frac{d\theta_i}{dt}+Q_i+Q_{cd}=0,\ R_tC_t\frac{d\theta_i}{dt}+\theta_e-\theta_i+R_tQ_i=0, \quad (10)$$

where: the proportion $\frac{d\theta_i}{dt}$, is the temperature gradient following the direction of the heat flux, and Q_i is the sum of the amounts of heat emitted per time unit in the room, expressed under the form $Q_i = Q_1+Q_2+\dots$.

If there are no heat emissions ($Q_i = 0$) and the temperature of the exterior air q_e is being constant (supposing that the time interval is approximately 1 hour), integrated in the time interval is $[t_0, t_1]$, corresponding to the temperatures θ_{i0} and θ_I then at both ends of the time interval:

$$\frac{1}{T_\theta}\int_{t_0}^{t}dt=\int_{\theta_{i0}}^{\theta_i}\frac{d\theta}{\theta_i-\theta_e} \Rightarrow \frac{t-t_0}{T_\theta}=ln\left(\frac{\theta_i-\theta_e}{\theta_{i0}-\theta_e}\right), \quad (11)$$

will have as final solution:

$$\theta_i=\theta_e+(\theta_{i0}-\theta_e)exp\left(\frac{t-t_0}{T_\theta}\right)= =\theta_e\left(1-exp\left(\frac{t-t_0}{T_\theta}\right)\right)+\theta_{i0}\,exp\left(\frac{t-t_0}{T_\theta}\right),$$

where: $R_tC_t = T_\theta$, represents the thermal constant of the system. In a longer interval of time (one day), supposing that the solar radiation is not taken into consideration and that the air outside and the environment radiations are smaller as compared to the heat exchange between the air and the building elements, the temperature outside is expressed analytically under the form of a Fourier series [1]:

$$\theta_e=\theta_{em}+\sum_{n=1}^{\infty}\theta_{en}\cos\frac{2\pi n}{24}(t-t_n), \quad (12)$$

where: θem is the daily average temperature of the air outside, in °C; θen is the amplitude of the umpteenth harmonic of the exterior air temperature, in °C, T is the hour of the day; tn is the hour when the maximum of the umpteenth harmonic of the exterior air temperature occurs. Taking into consideration that the first harmonic has the greatest share, one can admit that:

$$\theta_e=\theta_{em}+\theta_{e1}\cos\frac{2\pi}{24}(t-t_n), \quad (13)$$

where: θ_{e1} is the temperature of the first harmonic, in °C. If besides the thermal transfer by conduction, there occurs a heat transfer by convection as well (natural or artificial) Q_{cv}, then the equation of the mass balance sheet in the room will be:

- for natural convection or partial air-conditioning systems (ventilation):

$$C_t\frac{d\theta_i}{dt}+Q_i+\frac{\theta_e-\theta_i}{R_t}=D_m\,\mathbf{c}_{se}(\theta_e-\theta_i) \quad (14)$$

- for total air-conditioning systems (heating, humidification and ventilation):

$$C_t\frac{d\theta_i}{dt}+Q_i+\frac{\theta_e-\theta_i}{R_t}=D_m\mathbf{c}_{se}(\theta_{ff}-\theta_{if})_{inst}, \quad (15)$$

where: D_m is the mass fluid flow circulated in the room (air) or in the installation (water), in kg/s; θ_e, θ_{ff} and θ_{if}, are the initial temperature of the exterior air, the final temperature of the fluid (water) in the heating equipment, respectively the initial temperature of the fluid in the equipment, in °C; C_{sc} is the specific heat of the fluid circulated in the room (air) or in the heating equipment (water), in $j\cdot kg^{-10}\cdot C^{-A}$.

If there are no heat emissions $Q_1 = 0$, and considering the exterior air temperature θ_e as constant, the equation (38) integrated in the time interval (t_0, t), corresponding to temperatures θ_{i0} and θ_I:

$$\frac{1}{T_\theta}\int_{t_0}^{t}dt=-\int_{\theta_{i0}}^{\theta_i}\frac{d\theta_i}{(\theta_i-\theta_e)(D_mR_tc_{se}-1)}$$

$$\Leftrightarrow \frac{1}{D_mR_tc_{se}-1}ln\left(\frac{\theta_i-\theta_e}{\theta_{i0}-\theta_e}\right) \quad (16)$$

will have the final solution:

$$\theta_i=\theta_e\left(1-exp\left(-\frac{(D_mR_tc_{se}-1)(t-t_0)}{T_\theta}\right)\right)+ +\theta_{i0}\,exp\left(-\frac{(D_mR_tc_{se}-1)(t-t_0)}{T_\theta}\right) \quad (17)$$

In the majority of the practical applications, the amount of heat is expressed in W, relating the quantity expressed in J, per time unity. For heat quantity resulting from the natural convection Dmcs (qe-qi), expressed in J, relation [13] is used:

$$nV\rho_e c_{se}(\theta_e - \theta_i), \qquad (18)$$

expressed in W, where: V is the volume of the room (in m3), pe is the density of the exterior (in kg m-3); n is the schedule number of air changes per hour (in s-1). For a certain room, the temperature variation has been graphically represented, in the instance of the heat transfer by artificial conduction and convection (fig. 10.). Writing as Δp the total pressure produced by the pump, respectively ventilator, and ε as the ratios:

$$\frac{D\Delta p}{D_m (c_{se})_{apa/aer} (\Delta\theta)_{inst}} = \frac{\Delta p}{\rho_{apa/aer} (c_{se})_{apa/aer} \Delta\theta_{inst}}$$

the useful energy required in the time unity by the pump, or ventilator, to makeup for the pressure loss on the installation or in the room, will be compared to the amount of heat or cold circulated in the installation or room, per time unity.

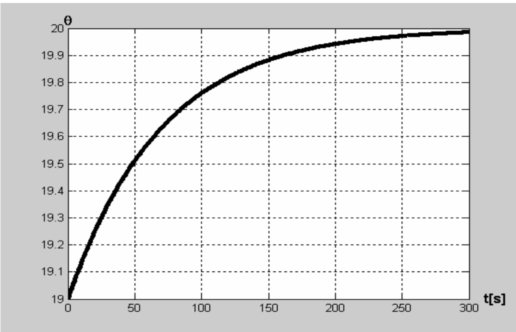

Figure 10 The 1 °C increase in temperature in the room in the instance of the heat transfer by artificial conduction and convection

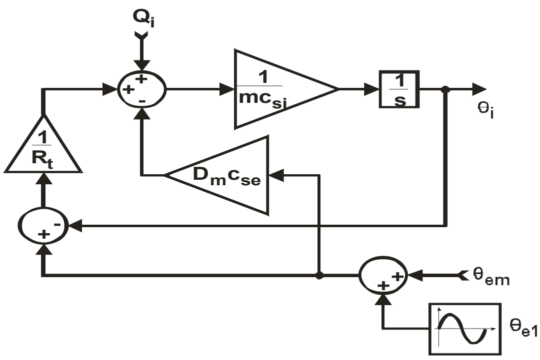

Figure 11 The block structural scheme of the heat transfer phenomenon in the room by natural or artificial conduction and convection (air conditioning system-only air)

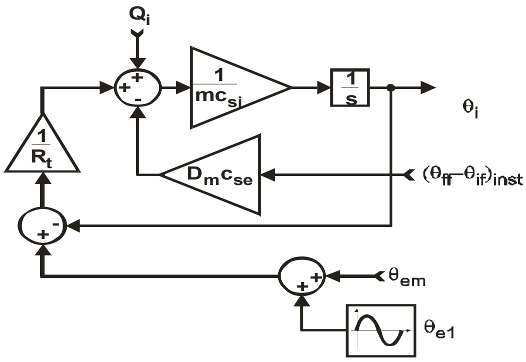

Figure 12 The heat transfer phenomenon by artificial conduction and convection (air-water conditioning system): a) the block structural scheme; b) the entrance and exit structural block scheme

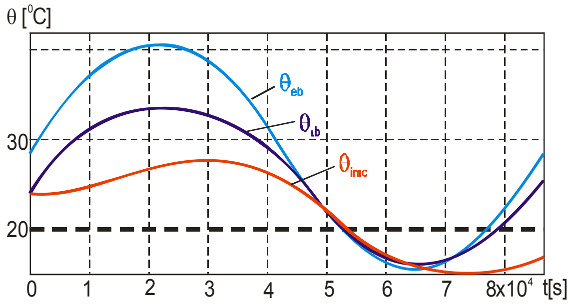

Figure 13 The temperature variation in the room for 24 hours, for concrete (λ = 1.74 W/M °C), and respectively material composite (λ = 0.21 W/M °C), for θ_{em} = 28 °C.

For the air-water conditioning systems, used in small dimension rooms, the local pressure leakages as approximately 100 m, a pump being needed to produce a 300 Pa pressure for each m, and running at 50% of the maximum efficiency.

For the totally air conditioning systems (ventilation installations), for covering the pressure leakages in the room, the ventilator must develop a 1 Pa pressure for each m, running at maximum speed. Replacing numerically the densities and the heat values specific to water (4,2 kJ.kg.^{0}C)and air (1,2 kJ.kg.^{0}C), as well as the typical values for the difference of temperature for the two situations

$$\Delta\theta_{inst} = (\theta_{ff} - \theta_{if})_{inst} = 80^0\text{-}70^0\text{=}10^0\text{C} \quad \text{and}$$

$$\Delta\theta_{inst} = (\theta_e - \theta_i)_{inst} = 8\,^0\text{C},$$

we obtain ε=0,71 W.kW^{-1}, the pump running at 50% of its maximum speed, or ε=1.4 W.kW^{-1}, the pump running at maximum speed, respectivelly ε=0.1 W.kW^{-1}, the ventilator running at maximum speed.

5. CONCLUSIONS

The air refreshing phenomenon modeling helps us understand its way of working. Particularly, the response to the application of an impulse type signal is characterized by a series of decreasing amplitude peaks. The time interval between 2 peaks, for a fixed point in the room, may be considered approximately constant. But the characterization of recirculation depends on the point in the room where the air quality estimation is required (the air in the polluted zone). Therefore, one can conclude that in order to correctly monitor the evolution of the polluting factors in view of controlling the production process of the horticultural product, it is important to place a larger number of transducers to obtain a piece of information as accurate as possible.

The air recirculation phenomenon modeling in the instance of the existence of heat emissions allows the determination of an airing constant, used when describing the efficiency of a ventilated room. Therefore, this constant will be taken into account when choosing the elements to be used in the structure of the automation system of the cultivating process. The patterning of the phenomenon of thermal transfer in the production rooms has highlighted the efficiency of using the air-water conditioning system (installations of heating, humidification and ventilation) as compared to the totally air conditioning system (installations of heating and ventilation). The useful power transmitted to the fluid, consuming the same amount of energy, is 14 times bigger in the instance of using the air-water conditioning system, as compared to the totally air-conditioning system. It has also demonstrated that the use of the composite materials instead of the usual building materials is more advantageous from the point of view of creating the microclimate conditions, thus making possible – between some limits of the exterior temperature – a passive conditioning (justifying, in terms of the exterior climate conditions, the fact that a temperature adjustment has not been used, and consequently, neither a climate conditioning equipment, which implies cutting down on the energy consume). In conclusion, when designing the mushroom growing spaces, there must be done an economic analysis regarding, on the one hand, the investments related to the building materials on the one hand, and the costs of the equipment which make the climate secure, on the other hand. The thermo dynamical models of the room do not detect anything else but the heat transfer by conduction and convection.

A modeling aiming at detecting the heat transfer by thermal radiation is also necessary, and the transitory regime that occurs when the heat is transmitted by the building elements of the room becomes necessary as well. In this respect, it is recommended that the rooms should be patterned as a quadripole.

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