NUMERICAL SIMULATION OF FURNACES TRANSIENT REGIMES FOR GRAIN DRYING

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Abstract This work aims at providinga *mathematical model of a furnace with forced flow of combustion products is developed. For this, a few variables were considered, such as: the combustion on grate (chemical equilibrium); the combustion products flow along the channels with pressure losses and heat transfer (by radiation and convection); the heating of the walls; the entrance of the secondary air; the displacement of the work point displacement by the characteristic of the ventilator; and the furnace transient regime operation. The coupled model includes several equations types, such as, the ODE's, and the PDE's non linear algebraic. The algorithm uses the different numeric methods with complex connection among them, when one method is applied inside another. The numeric simulations of transient regime reveal the causes of the furnace abnormal operation. This work also identified the: sketched safety's operational regimes; and determined the characteristics of the influence of the fuel consumption oscillations on the furnace. The procedure of abnormal phenomena deviation.*

Keywords*: furnace, combustion, heat transfer, transient regime, modeling.*

1. Introduction

The furnaces with forced flows, fed by vegetable fuels (firewood, cane pulp, rice peel, etc.) are thoroughly used by agribusiness in several countries as a source of “hot air”. Some of their advantages are: a comparatively low cost; use of renewable fuels; and a relatively short time of system implantation.

There are several types of such equipment like combustors, furnaces and gasifiers that use biomass as fuel (Heinz et al., 2001; Bhattacharya et al., 1999) and a research about their energetic and an environmetric aspect is necessary. Since the processes of: combustion, heat transfer by convection and radiation, heating of the walls and gas flows, among others occur in the furnaces are very complex. For understanding those processes not only experimental researches but also numeric simulation results obtained by the mathematical models are necessary. Numerous works were published about modeling and solving problems that occur in this kind of equipment.

Stehlik et al. (1995) and Stehlik et al. (1996), developed an approach in which the mathematical model of a furnace is represented by the group of sub-models that allows evaluating the furnace characteristics of different outlines. The main sub-models in this approach are of combustion and heat transfer both by radiation and by convection.

Lans et al. (2000) present the mathematical model and an experimental installation to determine the characteristics of straw combustion, including the front velocity of burning. The model is two-dimensional and describes the processes of crossed flows of straw and air. The volatile phase pirolize and char combustion are considered.

Valarelli et al. (1993), propose a mathematical model of a grain drying furnace with indirect fire where combustion, heat transfer (between combustion products and air for drying), and pressure losses are considered. The model is guided to furnace design and is simplified.

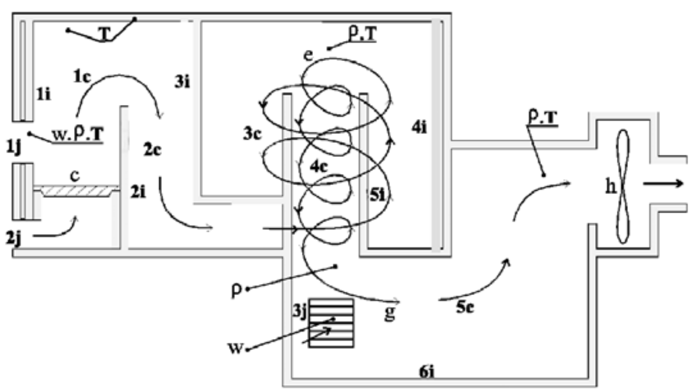


Figure 1. Scheme of industrial furnace: 1j, 2j, 3j - windows; c - grate; 1c, 2c, 3c, 4c, 5c – channels; 1i, 2i, 3i, 4i, 5i, 6i - walls; g - mixture zone; e - flows vortexes in cyclones; h – ventilator; P, T, W –points of measurement: pressure, temperature and velocity of flow, respectively.

In the present work the mathematical model of a furnace often used in the Brazilian agribusiness is proposed. Figure 1 presents a typical scheme of an industrial furnace adapted for experimental measurements. Their main components are: window 1j (fuel feed); window 2j (primary air entrance to maintain the combustion in the grate); grate c (where the combustion happens); channel 1c, 2c (flow without vortex); cyclones 3c, 4c (vortex flow); window 3j (controlled injection of secondary air to maintain the prescribed temperature of "hot air" (T_s); g (area of mixture of combustion products with the secondary air); 5c (channel with "hot air"); h (ventilator to force the flow through the furnace and dryer). The cyclones are used to separate the gas from solid particles that are produced during the combustion. The walls (1i,..., 6i) are thick and can be multi-sheet a long time (8 hours or more) for the establishment of the stationary regime.

It is observed that this furnace demands the creation of a peculiar mathematical model due to several particularities, such as :

a) window 3j has variable area making possible to control the temperature T_s;

b) the long transient regime caused by heating of the thick walls;

c) the incorporation of a ventilator with characteristics $G_v= f(\Delta P_v,\ W)$ that causes alteration of the flow rates for the channel 1c,..., 5c with the change of hydraulic losses in the furnace accentuating the transitory regime.

In the work by Krioukov et al., (1997) the mathematical model and numeric simulations of the stationary regimes of furnace are presented. The model is destined to predict the main operational characteristics ("hot air" outlet, furnace yield, total hydraulic losses in the channel, and the temperature alteration of the walls).

In the present work a modified model for transitory regimes is developed considering its importance due to the following properties:

- fuel consumption, being constant in the average, oscillates according to portion fuel feed;
- a transient regime (even that G_c, T_s, W are constant) is quite long occupying one third of the average work interval;
- during the passage to the stationary regime an emergency with abnormal effects can take place and force the furnace to stop.

2. Processes description

The presented furnace (Fig. (1)) works with almost constant values of fuel consumption (G_c), ventilator potency (W) and prescribed temperature of "hot air" (T_s). It is activated with the atmosphere temperature of the walls, adjusting some initial values: G_{pc} (flow of combustion products through the channel 1c,..., 4c); T_{sc} (cyclone exit temperature); $\Delta P_{1,4}$ (pressure difference between the atmosphere and the exit of channel 4c). To control T_s secondary air (G''_{ox}) is injected and the total flow rate ($G_s = G''_{ox} + G_{pc}$) passing the ventilator is adapted to its characteristic establishing the initial values ΔP_{v} e Q_{v} or a initial "work point". The temperatures of the walls (their distributions through the thicknesses and along the walls are not uniform) go up altering the values G_{pc} ,T_{sc} and $\Delta P_{1,4}$. Then the flow G''_{ox} through the window j3 is increased changing G_s and, the" work point" in the ventilator characteristic moves. Due to this transition, air flow (G'_{ox}) through the window 1 (Fig. (1)) can vary up to 3 times during 9 hours as long as the regime parameters (G_c, T_s, W) are established.

The main processes occurring in the furnace are: a) combustion in the grate area; b) heat transfer by radiation off this area to the walls; c) combustion products flow with the distributed hydraulic losses (along the channel) and the local losses (between channels); d) heat transfer by convection; e) transient heating of the walls (internal walls 2i and 5i are warmed on both sides); f) vortex flows in cyclones 3c and 4c that intensify the heat transfer;
g) mixture of the secondary air with the combustion products that leave channel 4c; and, h) displacement of the "work point" along the ventilator characteristic.

Therefore the mathematical model of the processes inside the furnace developed by Krioukov et al. (1997) is coupled, considering the following main phenomena: a) the combustion that is described by the chemical equilibrium model considering the possibility of the excess (α_{ox}>1) or air shortage (α_{ox}<1); b) radiation heat transfer from the grate area; c) the convection heat transfer to the walls; d) the hydraulic losses (distributed and local) in the channels; e) the transient heating of the walls; f) the cyclones influence in the heat transfer and in the hydraulic losses; g) the work point evolution by ventilator characteristic; h) the multi-sheet structure of the walls; and i) the secondary air injection.

3. Mathematical model

In this work, the mathematical model presented in Krioukov et al., (1997) was modified considering the heat transfer by radiation to two channels and the temperature dependence of heat conductivity $\lambda = f(T_p.)$. The simplified hypotheses are: a) the composition of the gas media is constant in channels 1c,..., 4c and after mixture area to the furnace exit; b) close to the grate the combustion process is concluded; c) the flow in all the channels are one-dimensional; d) each channel is considered as a tube with effective diameter and length; e) solid particles do not influence on the flow characteristics; f) Archimedes force is negligible; g) the radiation heat flux from the grate is divided in two parts: one is distributed uniformly by the channel 1c surface; the other one (Q_{12}) passes to channel 2c and is also distributed uniformly; h) the vortex flow parameter in channels 3c and 4c is constant; and, i) the surface temperature of each wall is uniform altering during the time.

In the mathematical model: a) the enthalpy (I_c) and fuel empiric formula (b_{ic}); b) the data on thermodynamic properties of the individual species of gas; c) geometric parameters of channels and walls; d) the properties of their materials: λ_p, c_p, ρ_p; e) the values T_s, T_{p0} (the initial wall temperature), T_a (atmosphere temperature); f) the heat transfer coefficient between the walls and atmosphere (h_a); g) the consumption G_c; h) the ventilator potency and their characteristics are knows.

3.1. Combustion sub-model and T_{gr} determination

The combustion in furnaces can happen either for $\alpha_{ox} > 1$ as for $\alpha_{ox} < 1$ and the adiabatic combustion of the products temperature (T_{ad}) gets a maximum value for $\alpha_{ox} = 1$. The sub-model bases on the chemical equilibrium model (Alemassov, et al.1971) predicts: the enthalpy, the temperature T_{ad} and the thermodynamics and termophysics properties: c_p (specific heat), μ (average molecular mass), η (viscosity), λ (heat conductivity) which are necessary for the subsequent sub-models.

In the combustion process charcoal, which generates the great heat radiation flux (Q_{gr}) transferring it to the walls is formed (Stehlik, et al., 1996). Consequently the gaseous temperature on the grate (T_{gr}) becomes lower: $T_{gr} < T_{ad}$. At the same time the wall surface temperature (T_{p1}) can be higher than T_{gr}. In the previous model (Krioukov, et al., 1997) the value of Q_{gr} was also considered, increasing the heat transfer coefficient by convection (h) and maintaining $T_{gr} = T_{ad}$. This scheme does not work correctly if $Q_{gr} >> Q_c$ (heat flow by convection) witch that is observed in industrial furnaces (Holman, 1981). Therefore the combustion sub-model was modified considering the heat radiation transfers between grate and walls of the channels 1c and 2c (Fig. (2)).

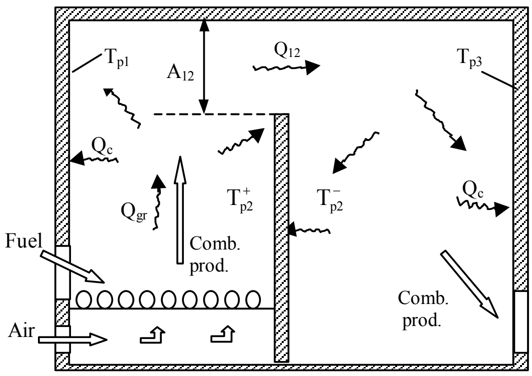


Figure 2. Heat radiation transfer scheme in channels 1c, 2c

Admitting that the flows Q_{gr} e Q_{12} are distributed uniformly on the wall surfaces, we obtain the energy equation on the grate zone:

$$I_{pc}(\alpha'_{ox}, T_{ad}) \cdot (1 + k_m^0 \cdot \alpha'_{ox}) \cdot G_c = I_{pc}(\alpha'_{ox}, T_{gr}) \cdot (1 + k_m^0 \cdot \alpha'_{ox}) \cdot G_c + Q_{gr} + Q_{12} \tag{1}$$

where,

$$Q_{gr} = A_{gr} \cdot K_{rd} \cdot \sigma \cdot \varepsilon_{gr} \left[\left(\frac{T_{cv}}{100} \right)^4 - \left(\frac{T_{p1}}{100} \right)^4 \right] \tag{2}$$

$$Q_{1,2} = A_{12} \cdot \sigma \cdot \varepsilon_{1,2} \left[\left(\frac{T_{p1}}{100} \right)^4 - \left(\frac{T_{p2}}{100} \right)^4 \right] \tag{3}$$

where: $I_{pc}(\alpha'_{ox}, T_{ad})$ - is the enthalpy combustion products with temperature T_{ad}, and $I_{pc}(\alpha'_{ox}, T_{gr})$ - with temperature T_{gr}; α'_{ox} - excess air coefficient in the grate zone; A_{gr} – grate area; K_{rd} - identification coefficient; σ - Stefan-Boltzmann constant; ε_{gr} – coal emissivity; $A_{1,2}$ – passage area between channels 1 and 2; $\varepsilon_{1,2}$ – emissivity of the area $A_{1,2}$, ($\varepsilon_{1,2} = 1$, black body emissivity is acceped); T_{cv} – burned coal temperature; T_{p1}, T_{p2} – wall average temperatures in channels 1c, 2c; k_m^0 - mass stoichiometric ratio between air and fuel.

The burned coal temperature (T_{cv}) can be defined as equal to the adiabatic temperature when $\alpha_{ox} = 1$. In the simplified form considering that $I_{pc} = \overline{C}_p T$ we obtain:

$$\overline{C}_p T_{ad} G_{pc} - Q_{gr} - Q_{1,2} = \overline{C}_p T_{gr} G_{pc} \tag{4}$$

or

$$T_{gr} = T_{ad} - \frac{Q_{gr} + Q_{1,2}}{\overline{C}_p G_{pc}} \tag{5}$$

where: G_{pc} e $\overline{C}_p$ are, respectively, mass flow and average specific heat of the combustion products in channels 1c, 2c.

Also considering the additional energy losses the Eq. (5) is transformed:

$$T_{gr} = T_{ad} - \frac{Q_{gr} + Q_{1,2}}{\overline{C}_p G_{pc}} - \frac{\Delta H_b \varphi}{\overline{C}_p (\alpha'_{ox} k_m^0 + 1)} \tag{6}$$

where: ΔH_b – low heating value; φ - coefficient of energy losses in the grate ($\varphi \approx$ 0,02,...,0,05) (Stehlik, et al., 1995).

3.2. Heat transfer sub-model in the walls

This sub-model includes the heat transfer equations for each wall considering the dependence $\lambda_p = f(T_p)$:

$$\frac{\partial T_p}{\partial \tau} = \frac{1}{\rho_p c^p} \frac{\partial}{\partial y_p} \left(\lambda_p \frac{\partial T_p}{\partial y_p} \right) \tag{7}$$

with boundary condition:

$$h\left(T - T_p^g\right) + q_{rd,i} = -\lambda_p \left(\frac{\partial T_p}{\partial y_p} \right)_{gp} ; \tag{8}$$

$$h_a \left(T_p^a - T_a\right) = -\lambda_p \left(\frac{\partial T_p}{\partial y_p} \right)_{ap} ; \tag{9}$$

where: $q_{rd,1} = \frac{Q_{gr}}{B_1 l_{ef1}}$; $q_{rd,2} = \frac{Q_{gr}}{B_1 l_{ef1}}$; $q_{rd,3} = \frac{Q_{12}}{B_2 l_{ef2}}$; $q_{rd,i} = 0$; for i = 4, 5, 6; T_p^g - wall temperature on the hot side; B_k, $l_{ef,k}$ – perimeter and effective length of the k-th channel; y_p – current coordinate for the wall thickness; T_p^a - wall temperature on the cold side.

For walls 2i and 4i, (Fig. (1)) the boundary conditions (Eqs. (8,9)) has another form considering the contacts of the hot gases with both sides.

3.3. Flow sub-models into channels

This sub-model was deduced starting from gas one-dimensional transient equations, considering friction and heat transfer (Sisson, et al., 1988). The equations were simplified proposing that $\frac{\partial S}{\partial \tau} << u \frac{\partial S}{\partial x}$; S = T, P ρ. In result of evident transformations the final form of sub-model for k-th channel obtained was:

$$\frac{dT_k}{dx} = -\frac{h_k B_k \left(T_k - T_p(\tau)\right)}{G_k c_{p\,k}} = -f_{Tk} \tag{10}$$

$$\frac{dP_k}{dx} = \left(\frac{f_{Tk}}{T_k} - \frac{\xi_k}{2D_{ef,k}} \right) \Big/ \left(\frac{A_{ef,k}^2 P_k}{G_k^2 R T_k} - \frac{1}{P_k} \right) \tag{11}$$

where: $G_k = G_{pc}$ (k = 1...4); $G_k = G_s$ (k = 5); u – velocity; τ - current time; x – coordinate along channel; $A_{ef,k}$ – channel effective area; ξ_k - friction factor; ρ - gas density; $D_{ef,k}$ – effective diameter; h_k – heat transfer coefficients that are calculated by the formulas:

$$D_{ef,k} = \frac{4A_{ef,k}}{B_k}; \qquad h_k = \frac{Nu_k \lambda_k}{D_{ef,k}}; \tag{12}$$

The number of Nusselt is determined by the formula

$$Nu_k = 0{,}145\,\mathrm{Re}_k^{0,72}\,\mathrm{Pr}_k^{0,33}\left(\frac{L_T}{D_{ef,k}}\right)^{0,6}\exp\left(1-\frac{L_T}{D_{ef,k}}\right); \tag{13}$$

obtained by Gortishev & Olimpiev, (1999) for channels with irregular surfaces (L_T - brick thickness).
The friction factor, using Reynolds analogy, is determined by:

$$\xi_i = \frac{8 Nu_k\,\mathrm{Pr}_k^{2/3}}{\mathrm{Re}_k}; \tag{14}$$

The system represented by Eqs. (10, 11) is integrated for each channel. The parameters in the previous channel exit are the initial data in the posterior channel entrance (Fig. (1)). Some channels are limited by two walls, for instance: the channel 1c is limited by the walls 1i and 2i; the channel 3c – by the walls 4i and 5i. At the same time a wall can have contact with two flows, for example: wall 2i has contact with the flows of the channels 1c and 2c, wall 5i – with flows of channels 3c and 4c. The sub-model considers these structural particularities, providing a coordination matrix between the channels and the walls. In this sense, the sub-model is flexible and it can be easily adapted for furnaces with different structures.

Whirl flows (channels 3c, 4c) are described by Eqs. (10, 11) with correction of coefficients ξ_k, h_k. Fig. (3) presents cuts of two channels with necessary designations, where *a* and *b* represent the sizes of the rectangle that form the vortex in the cyclone. Coefficient h_k in cyclones is determined by the formula (Mukhachev & Shukin, 1972):

$$h_k = h_k^0\left(1 + 1{,}15\Phi^{1,06}\right) \tag{15}$$

where: h_k^0 is the heat transfer coefficient without the vortex.

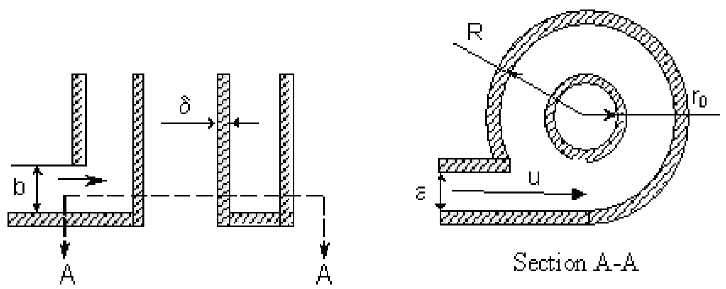


Figure 3. Scheme of furnace cyclone

The vortex parameter Φ is determined for both external and internal cyclone by the formulas:

$$\Phi_{ext} = \frac{2\pi R^2\left(1-X^3\right)}{3ab}; \tag{16}$$

$$\Phi_{int} = \frac{2\pi r_0 R^2\left(1+X+X^2\right)}{3ab\left(1+X\right)}; \tag{17}$$

where: $X = r_0 + \delta$.

The friction coefficient in cyclones (ξ_i) is corrected using by Reynolds analogy. The formulas for local hydraulic losses (because of abruptness between channels or flow turn) were traditional:

$$\Delta P_k^m = \xi_k^m \rho (u_k^2/2); \tag{18}$$

where: ξ is determined by the guide data (Lencastre, 1972). In the cyclone, the exit formula is applied:

$$\Delta P_k^m = \xi_k^m \rho \left(w_k^2 + u_k^2 \right) / 2 \qquad \text{for } (k = 4) \tag{19}$$

Considering the axial (u_k) and tangential (w_k) velocity, the coefficient ξ_4^m characterizes the pressure losses that are the result of kinetic energy dissipation, not only in the cyclone exit but also and due to mixture with secondary air. The coefficient ξ_4^m = 0,25 was determined by experimental measures between 3 points: the internal cyclone entrance; the exit of this cyclone; the ventilator entrance (Fig. (1)).

3.4. Sub-model of mixture with secondary air

The mixture is necessary to maintain the prescribed temperature (T_s) in the furnace exit (dryer entrance) independently from parameter values (*P, T, G)* in the cyclone exit. In this sub-model the G''_{ox} e G_s mass flows is calculated. Initially α_{ox}^s is determined by non-linear equation:

$$I_{pc}\left(\alpha'_{ox}, T_c\right)\left(1 + \alpha'_{ox} k_m^0\right) + I_{ox}\left(\alpha_{ox}^s - \alpha'_{ox}\right) = I_{pc}\left(\alpha_{ox}^s, T_s\right)\left(1 + \alpha_{ox}^s k_m^0\right) \tag{20}$$

and then the following the formulas are used:

$$G''_{ox} = \left(\alpha_{ox}^s - \alpha'_{ox}\right) k_m^0 G_c ; \qquad G_s = \left(\alpha_{ox}^s k_m^0 + 1\right) G_c ; \tag{21}$$

3.5. Submodel of the flow adaptation

This submodel is described by the equation:

$$\frac{G_s}{\rho_s Q_{nm}} = W \sum_{q=0}^{3} a_q \left(\theta\right)^q ; \quad \theta = \frac{\sum_k \Delta P_k + \sum_k \Delta P_k^m}{\Delta P_{nm} W} \tag{22}$$

where: Q_{nm} , ΔP_{nm} are, respectively, the volumetric flow and the pressure difference in the ventilator in nominal regime; a_q – approximation coefficients of the ventilator characteristics.

The distributed and local losses (ΔP_k , ΔP_k^m) are functions of many variables: T_k, u_k , $\alpha'_{ox}, \alpha_{ox}^s$, G_c, G'_{ox}, G''_{ox} , T_p etc. Therefore Eq. (22) including the characteristics of several furnace parts is complex and cannot be explicitly presented. To perform an iteration in this equation it is necessary to accomplish calculations for all the sub-models above described with an initial value G'_{ox} , i.e.: a) solve the equations of chemical equilibrium; b) integrate Eq. (7) for each wall; c) integrate Eqs. (10, 11) in each channel; d) determines the local losses; and, f) solve Eq. (20).

By Equation (22) the primary airflow value (G'_{ox}) is determined. The furnace yield is determined as the ratio of the "hot air" flows in the furnace exit:

$$\eta = G_s / G_s^{id} = \frac{\alpha_{ox}^s k_m^0 + 1}{\alpha_{ox}^{s,id} k_m^0 + 1} \tag{23}$$

where, G_s^{id} is a virtual mass flow when the heat transfer coefficients to the walls are null.
The value $\alpha_{ox}^{s,id}$ only depends on the fuel type and on T_s, being assured by a chemical equilibrium model. In the case of eucalyptus firewood with T_s = 400K, the $\alpha_{ox}^{s,id}$ is equal to 27,5.

The calculation algorithm is complex and includes a variety of numeric schemes applied simultaneously, such as know: a numeric method of ODE's resolution; Newton's method to solve the equilibrium chemical equations; Newton's method to solve the Eq. (22) that includes as fragments the Eqs. (6, 10, 11, 20) and the method of lines to solve the Eq. (7) for each wall.

The FORNIJ code was created in C++ language and it has a modular structure. Each module computes the processes in a fragment of furnace. As a result, the code flexibility that allows its adaptation for different furnace types is gotten. The parameters of an operational regime control can be altered during the execution which enlarges essentially the area of numeric simulations, for instance, to simulate the exit of abnormal regime.

4. Model identification and comparison with experimental data

The mathematical model was verified and identified by the tests in an industrial furnace, and the following variables were measured (Fig. (1)): temperature (with the precision of 2^{o}C), airflow velocity (with 0,1m/s precision), and pressure differences along the channels (with 1mm H_2O precision). The basic regime parameters were maintained:

$T_s = 400 \pm 2$K; $W = n/n_{nm} = 0{,}95 \pm 0{,}1$ (n_{nm} = 1800 rotation/min – ventilator nominal rotation); $G_c = 0{,}046 \div 0{,}048$kg/s (170kg/h) during 9 hours. The main geometric furnace characteristics are presented in Tab. (1),

Table 1 - Furnace geometric parameters.

Channel	1c	2c	3c	4c	5c	Grate
A_{ef} (m^2)	0,70	0,70	0,67	0,20	1,5	0,7
L_{ef} (m)	1,20	1,20	1,00	1,00	2	-

Wall	1i	2i	3i	4i	5i	6i
A_p (m^2)	3,51	1,89	3,10	4,84	1,44	9,20
δ_p (m)	0,30	0,10	0,30	0,20	0,10	0,10

where: A_p is the wall effective area; δ_p is the wall thickness.

The multi-sheet walls 1i and 3i were built with refractory brick (10cm), fiberglass (10cm), clay brick (10cm). Eucalyptus firewood was used for fuel with: the empiric formula $C_{4,02}H_{5,92}O_{2,80}N_{0,04}S_{0,02}$; humidity = 20%; enthalpy I_c = - 4123kJ/kg. Fig. (4) shows the comparison of the experimental and theoretical data: ΔP_s, $\Delta P_{1.3}$, G_s, G'_{pc}, T_{ci} .

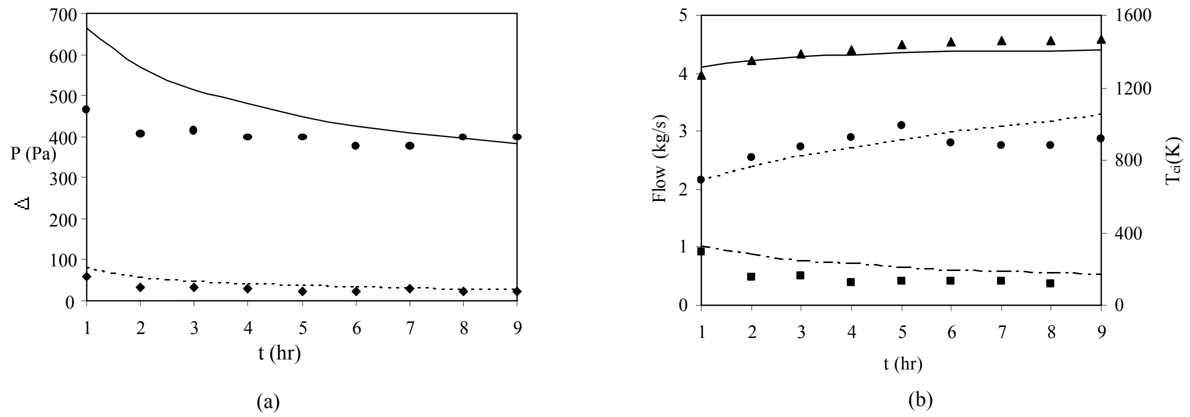


Figure 4. a) Pressure evolution: ΔP_s (──) e ΔP_{13} (- - -); ΔP_{13} – pressure differences between atmosphere and end of channel 3c; (•,♦) – experimental data.; b) Mass flow rates G_{pc} (–·–·–), G_s (—) and temperature T_{ci} (- - -) evolution

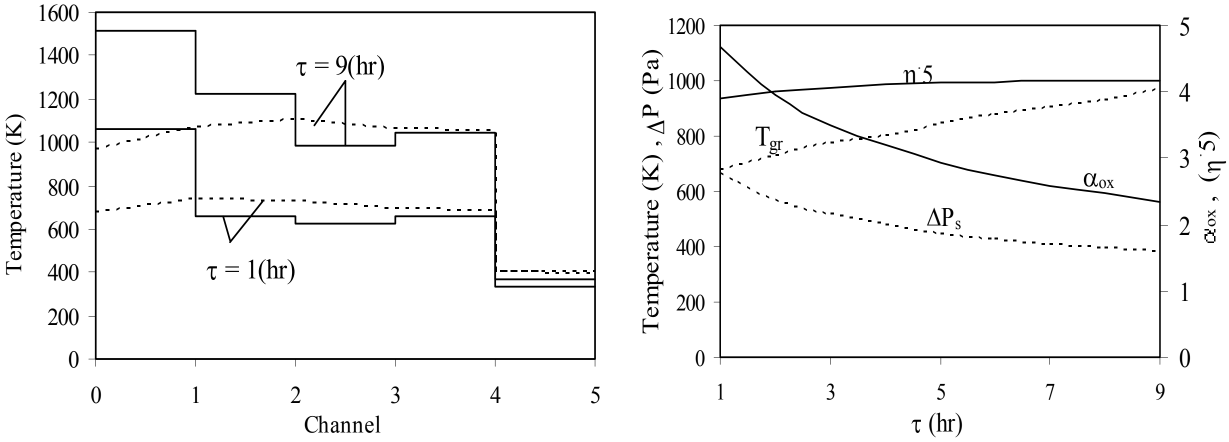
The heat radiation is the main part of the heat transfer from the grate area (Holman, 1981) and its prediction is not possible without using the experimental data, among which the radiation area and the charcoal temperature are essential. These data or other information about the radiation were not measured during the tests. Therefore, in formulas (2, 3) to calculate the heat radiation it was accepted that the irradiation area is equal to the grate area (A_{gr}), multiplied by the a $K_{rd} < 1$ coefficient that was determined by the mathematical model identification with the experimental dependence $T_{ci} = f(\tau)$ in the basic regime.

The difference between theoretical and experimental results is accounted for the simplicity of the sub-models of radiation and of local losses in the cyclone exit 4c. It was observed that the identification coefficient K_{rd} = 0,3 is almost invariable when regime parameters, wall materials and most of furnace sizes change. But for another fuel (with a heating value quite different from the heating value of the eucalyptus firewood) it is necessary to accomplish the new model identification using the new experimental data.

5. Simulation results

5.1. Normal regime

The important characteristics of the basic regime (Tab. (1)) are presented in Figs. (5a) and (5b). The distributions T_k and T_{pi} through the furnace channels (Fig. (5a)) show that in the process beginning the temperature T_{p1} is higher than T_1 because of the great heat flux by the radiation inside the first channel. A part of the heat radiation also enters in the second channel, but its potency is smaller, for that $T_2 > T_{p2}$. But in the end of the operation (τ = 9hs), it is observed that $T_2 < T_{p2}$. For these reasons, the temperatures T_1 e T_2 increase along the channels. And, on the contrary, in the channels 3, 4 and 5 $T_k > T_{pi}$ the flow temperatures stoop down. Between channels 4 and 5, the temperature T diminishes until T_s (Fig. (1)) because of the mixture with secondary air.



a) b)

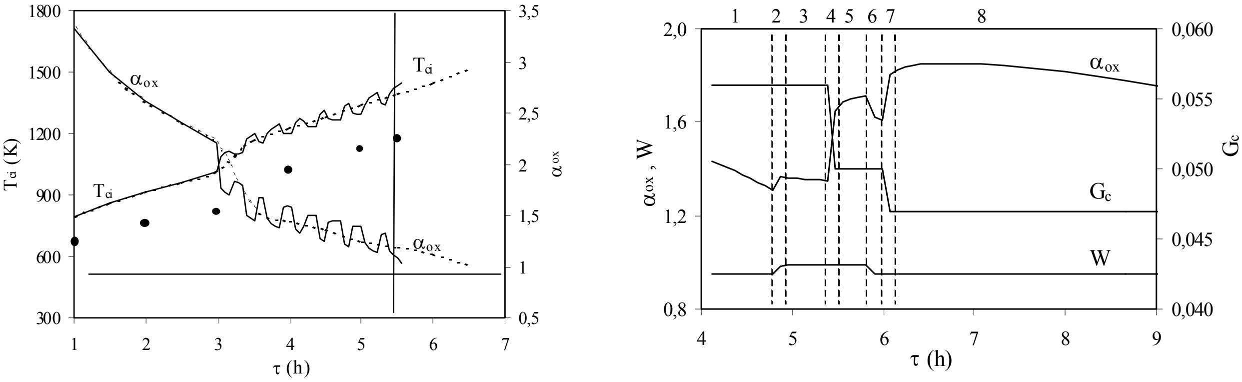
Figure 5. a) Distribution of T_k e T_{pi} along channels for τ = 1hr and τ = 9hr (T_{pi} — ; T_k – –); b) Evolution of furnace main characteristics

Later, the differences of ($T_{cv} - T_{p1}$) and ($T_{p1} - T_{p2}$) diminish what causes the decrease of heat radiation and contributes to increase the initial flow temperature. Then, in the mixture entrance area the flow G_{pc} has more energy and it is needed to inject more secondary air to maintain the temperature T_s in the furnace exit. Therefore, flow G_s is increasing. In this case, the work point moves by the ventilator characteristic decreasing ΔP_s. As a result, the flow inside the furnace is adapted to this alteration reducing the values G_{pc} and G'_{ox}. Then the coefficient $\alpha'_{ox} = G'_{ox} / \left(k_m^0 \, G_c\right)$ diminishes increasing even more the temperature (T_k) in channels 1-4, which is confirmed by Figs. (5a) and (5b). Through this main effect the behavior of other characteristics are explained. For instance, the yield (η) which is determined, by formula (23), increases during the time.

5.2. Abnormal regime

During the furnace tests with high G_c consumption (with other constant parameters) an abnormal effect was observed when, after some operation hours, flame appeared in the mixture area and the furnace was stopped to preserve the ventilator. Afterwards the operational regimes with this particularity were called "abnormal regimes". In Fig. (6a) the evolutions of α'_{ox} and the experimental and theoretical temperatures T_{ci} are shown. The test was accomplished with parameters T_s = 395K, W = 0,95 and, during the first 3 hours, G_c = 0,051kg/s was maintained, increasing it later until G_c = 0,056kg/s. The flames appeared at the end of the fifth operation hour.

Then the elaborated mathematical model is necessary not only to simulate the regular regimes but also to predict and to explain the abnormal regimes and, handling through the control parameters T_s, W, G, to avoid these regimes, in order not stop the work. Analyzing the numeric simulations the cause of the abnormal regime was discovered. It is obvious that with the G_c increase, the flow energy and G_s are intensified and that induces a displacement of the work point. Also the coefficient α'_{ox} decreases and in one moment it crosses value α'_{ox} = 1.



a) b)

Figure 6. a) Evolutions T_{ci}, α'_{ox} in abnormal regime (• experiment; ——, – – theory); b) Schedule off abnormal regime exit and the alteration of α'_{ox} in this procedure.

But in this case, an oxidizer lack in combustion product appears and species H_2, CO, CH_4 , etc are generated in notable concentrations. Then, in the mixture entrance area, these substances contact with the secondary air that induces the combustion renovation. In the numeric simulations (without oscillations) value α'_{ox} =1 was gotten after 7 hours of operation. But considering the oscillations the arrival time of α'_{ox} =1 diminishes (Fig. (6a)).

Based on the numeric simulations a procedure to avoid abnormal regimes can be recommended, if suddenly, the work processes is tended to this regime. Mainly it can be handled by three control parameters: G_c, W, T_s. in order not to prejudice the grain quality we can only alter the temperature T_s in the interval 400 ± 5K. The fuel consumption also cannot be quickly altered because, in practice, the relaxation time G_c is quite big ($\Delta\tau \approx$ 5...10min). Quickly it can only alter W in interval (± 20% of ventilator nominal regime).

To exemplify, we admited that during the simulation (G_c = 0,056kg/s, T_s = 395K, W = 0,95, Fig. (6a), line α'_{ox}) value $\alpha'_{ox} \approx$ 1,3 already was gotten and it is evidenced that soon from the furnace will create flames (area 1, Fig. (6b)) and it is needed to established the basic regime (G_c= 0,047kg/s, W = 0,95). Then we increased the ventilator potency W = 1 (area 2). In the result value α'_{ox} (until 1,3 to 1,35) increases. In the zone 3 α'_{ox} diminishes lightly. In area 4 we decrease the consumption G_c till 0,050kg/s that induces the abrupt α'_{ox} increase (avoiding the abnormal regime) and fast T_{gr} fall. In area 5 the G_c decrease influence is continued but with a smaller gradient. In area 6, the ventilator potency turns into its initial value resulting in an abrupt decrease of α'_{ox}. In area 7 a new decrease of G_c until 0,047kg/s takes place contributing to a considerable growth of α'_{ox}. In the last phase (Fig. (6b) area 8) the furnace characteristics develop to their stationary values. Then the proposed procedure allows transition from the abnormal regime to the basic regime without stopping the furnace. It is observed that during this procedure the wall temperature alterations were few because they are quite thick and accumulate great heat energy. Therefore, it can be concluded that the walls are a little sensitive to the fast alterations of the gaseous flow parameters.

6. Conclusions

The furnace mathematical model with forced flow using vegetable fuels was developed to simulate the evolution of the main furnace characteristics ("hot air" flow in the dryer, pressure differences in the channels and in the entire furnace, walls heating, etc.). Despite using some simplified hypotheses (combustion product flows are one-dimensional; approach of chemical equilibrium; heat flux by radiation is distributed evenly by surfaces in channels 1 and 2; etc) the model predicts the main characteristics accurately enough to be useful in the design, tests and exploration because it considers the main phenomena in the furnace (combustion, flows by the channels with the friction and heat transfer, wall heating, mixture with secondary air, properties of ventilator characteristic, etc) in their interconnections. The model is sensitive to the fuel type and its consumption, to the geometric characteristics of the channels and walls, to the wall materials, to the ventilator characteristic and potency, to the "hot air" prescribed temperature. Therefore, the numeric simulations coincide mainly with the available experimental data. The most important application of this model in the present work is to research transient regimes. In this aspect it can be emphasized the quite long time to get the stationary regime; the evolutions of the grate temperature; the evolution of the air excess coefficient; the gas temperature in the cyclone; the yield; the total hot air flow; the explanation of flames appearance in abnormal regimes and the elaboration of the procedure to avoid these regimes.

One of the model advantages is its modular structure that allows easily an easy adaptation of it to the different furnace schemes. The results obtained in the work are useful for furnaces in which vegetable fuels and ventilators are used to check the existence of abnormal regimes; the influence of oscillations; the great importance of heat transfer by radiation; the long time to enter in stationary phase.

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