

Study of Heat Transfer in a Fixed Bed Centimeter Furnace with Pottery Walls : The Case of Natural Laterite and Kaolinic Laterite

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Abstract: The valorization of municipal waste is a good option for energy production. Our work consists of a numerical study of the heat equations in a fixed-bed centimetric grate furnace modeled by a natural laterite inner wall and a kaolin laterite wall. Simulation results showed that the kiln filling rate at 100% is 5 kg of fuel. From 500° C as the starting point for heating, the temperature reaches a maximum of 1225° C and 1050° C respectively for natural and kaolin laterite at 30 minutes of operation. In addition, the internal temperature is optimal around 35 ± 5 hgs⁻¹ of internal air flow. A drop in internal temperature was observed after 35 minutes due to mass degradation and convection of the outlet gas. Natural laterite tends to retain heat inside the kiln more than kaolinite, which dissipates too quickly. These results show that, depending on how heat is converted into energy, the right materials and operations need to be chosen to optimize the process.

Keywords : Municipal Waste, Grate Furnace, Natural Laterite, Kaolin Laterite, Heat Equations.

1. INTRODUCTION

Municipal solid waste (MSW) is a product that needs to be used and transformed to produce green energy with no adverse effects on the environment. Energy recovery technologies such as pyrolysis, gasification, incineration and liquefaction are being developed to produce electricity and biofuels from municipal waste [1]. Incineration is one of the most common methods of recovering heat energy from waste in many countries [2]. This involves equipping them with a powerful incinerator capable of processing municipal waste in respect of ecological rules [3], [4]. The requirements imposed on combustion systems are continually changing over time, and becoming more and more strict [5]. Combustion system safety has always been essential, but the emphasis on heat transfer efficiency, temperature uniformity, equipment scaling, efficiency, controls and more recently, environmental emissions and combustion-generated noise has evolved over time [6]. These challenges have been successfully met in most applications by combining experience and good engineering practice with creative and innovative problem solving. The fundamentals of turbulent mixing, heat transfer and chemical kinetics are necessary to understand combustion. The combustion process involves a number of complex chemical reactions. It is a process of complete oxidation of pyrolysis gases in the presence of excess oxygen at temperatures between 500°C and 2000°C [7], [8]. The varying shapes and characteristics of the solids to be burned require different combustion devices. The most commonly used technologies for incinerating solid waste are grate furnaces, fluidized-bed furnaces and rotary kilns. The grate furnace is the most widely used incineration process for municipal waste [9]. However, it is important to save energy. This means optimizing heat and mass transfer in the incineration process. The aim of this approach is to increase incinerator performance and reduce energy consumption [10]. Heat transfer is a process by which energy is exchanged in the form of heat between bodies or environment at different temperatures. Heat can be transferred by conduction, convection or radiation [11]. Heat transfer is one of the most studied physical phenomena today. Improving mass and heat transfer remains a major challenge, both in industry and in research.

Several researchers have presented their work on this phenomenon. Among them, the work of N. Berour [12]. He has numerically investigated coupled heat transfer, combining radiation, conduction and convection for semi-transparent, non-gray environment heated to high temperatures. The results of the study by K. Saito and M. Ohta on heat transfer phenomena in the preheat zone of a cupola have shown that layer-by-layer loading leads to more efficient melting of the metal charge [13]. H. Sun et al have developed a mathematical model for a continuous-casting cupola, based on both mass and heat balance [14]. A numerical study of heat transfer in a multilayer wall with two or three layers has been carried out by Y. Tamene and al [15]. Boulkroune et al. studied the influence of

several parameters on heat transfer, such as the choice of material and the thickness of the first inner layer. The results showed that the various parameters have an influence on the internal furnace temperature, and the choice of the best configuration for good thermal insulation leads to greater energy savings [16]. Nougbléga and al. carried out a thermal and dynamic analysis of mixed convection fluid flows in the building-integrated chimney, using governing equations discretized by the finite difference method. The results are presented in terms of rationalizations, isotherms, velocities and heat transfer intensities [17].

The present study deals with heat transfer in the natural laterite and kaolin laterite walls of a centimetric fixed-bed furnace.

The main objective of this study is to compare and choose between natural laterite and kaolinic laterite walls, the one best suited for a centimetric fixed-bed kiln with pottery walls, intended for the incineration of municipal waste.

2. MATERIEL AND METHODS

The aim is to carry out thermal modeling in order to study the influence of the main operating parameters of a fixed-bed centimetric grate kiln with pottery walls. We will analyze the temperature evolution at the surface of the internal walls in natural laterite and kaolin laterite.

2.1 Description of Heat Transfer Modes

In a furnace, the inner walls of the combustion chamber exchange heat with the fuel by conduction, convection and radiation [18]. In a rotating incinerator, there are three modes of heat transfer: convection, conduction and radiation. Figure 1 shows the heat transfer modes in a grate incinerator.

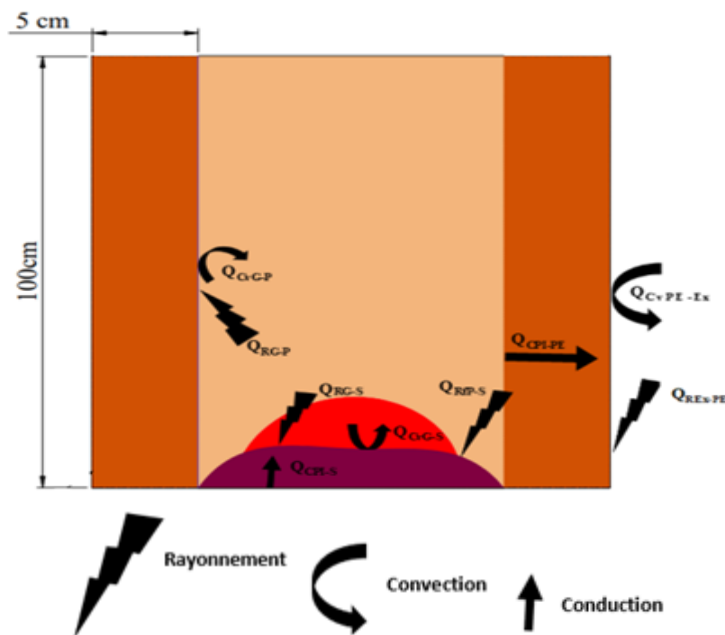


Figure 1. Heat transfer balance in a grate incinerator.

The upper part of the solid waste surface receives heat directly by radiation (Q_{RG-S}) and convection (Q_{CV-G-S}) from the gases above, while on the lower, solid waste-covered surface, heat transfer takes place by conduction (Q_{CPI-S}) from the wall to the solid waste. During combustion, the wall receives thermal energy via radiation (Q_{RG-P}) and convection (Q_{CVG-P}) from the burning hot gases. Part of this energy is stored in the incinerator walls. The other part, which is exclusively reflected (Q_{RIP-S}), is radiated from the wall to the top of the waste bed. However, radiation remains the dominant mode of transfer [19], [20].

The amount of solid waste in the kiln must take into account the kiln dimensions. Figure 2 shows how to fill the furnace.

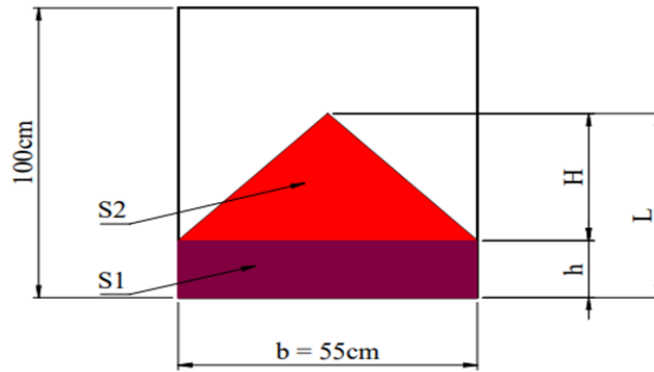


Figure 2. Cross-section of combustion chamber.

The filling rate of the furnace is given by the following relationship.

$$\tau = 2h/L \quad (1)$$

Where h is the height of the solid bed, L is the mean length of the furnace.

The mass of solid waste is then calculated using the following equation.

$$m = \rho \frac{bL^2}{4} (3\tau - 1) \quad (2)$$

Where m is the mass of solid waste, ρ is the density of the waste, L is the length of the combustion chamber, h is the height of the solid bed, b is the width of the furnace, τ is the filling rate.

2.2. Thermal Energy Distribution Inside The Furnace

During combustion, the upper part of the solid waste surface receives heat directly by radiation and convection from the gases above, while on the lower, solid waste-covered surface, heat transfer takes place by conduction from the wall to the solid waste. The waste radiation form factor is calculated by the following relationship.

$$F_C = length_i^2 + 2(width_i + height_i)^2 - 2 \frac{(width_i + height_i)(length_i^2 + (width_i + height_i)^2)^{0.5}}{length_i^2} \quad (3)$$

Where $length_i$ is the length of facade i , $width_i$ is the width of facade i , $height_i$ is the height of facade i .

During combustion, the wall receives thermal energy via radiation and convection from the burning hot gases. Heat exchange is expressed by the following relationship.

$$m_m C_{pm} \frac{\partial T}{\partial t} = \sum_{i \neq m} g_{i \neq m} (T_i - T_m) \dot{m}_a^{inlet} + h_a^{inlet} - \dot{m}_a^{outlet} h_a^{outlet} \quad (4)$$

With :

$$g_{i \neq m} = \left(h + h_r + \frac{\lambda}{e} \right) A_m \quad (5)$$

Where m_m is the mass of the m facade of the wall, C_{pm} is the mass calorific capacity of the m facade of the

wall, $g_{i \neq m}$ is the heat exchange coefficient with the m facade of the wall, T_i is the temperature of the i facade of the wall, T_m is the temperature of the m facade of the wall, \dot{m} is the air flow rate, h_a is the air convection heat exchange coefficient, A_m is the area of the facade m of the wall, h is the internal gas convection coefficient, h_r is the radiation coefficient of the i facade on the m facade, λ is the thermal conductivity of the wall, e is the wall thickness.

To calculate the heat flux transmitted by forced convection, we must first calculate the Reynolds number and the Prandtl number. Equations (6) and (7) give the Reynolds number and Prandtl number respectively [21], [22].

$$R_e = \frac{u_{\infty} d}{\nu} \quad (6)$$

$$P_r = \frac{\nu}{\alpha} \quad (7)$$

Where R_e is the Reynolds number, P_r is the Prandtl number, $u_{\infty} d$ is the product of air velocity and path length, α is thermal diffusivity, ν is fluid viscosity.

From the Reynolds and Prandtl numbers and depending on the geometry of the combustion chamber, the correlation is chosen [23]. The Nusselt number is expressed as a function of the Reynolds and Prandtl numbers according to the following equation [24].

$$N_u = \begin{cases} 0.332 R_e^{1/2} P_r^{1/3} & \text{for } R_e < 5 \cdot 10^5 \text{ and } P_r < 0.7 \\ 0.029 R_e^{4/5} P_r^{1/3} & \text{for } R_e > 5 \cdot 10^5 \text{ and } 0.6 < P_r < 60 \end{cases} \quad (8)$$

Where N_u is the Nusselt number, R_e is the Reynolds number, P_r is the Prandtl number.

The following relationship can be used to calculate the local or average exchange coefficient.

$$h = \frac{k N_u}{L} \quad (9)$$

Where N_u is the Nusselt number, k is the conductivity of the wall, L is the path length along the facade.

Heat flow (local or global) is calculated using Newton's law for each wall [25].

$$\phi_i = h A_i \quad (10)$$

Where h is the internal fluid convection coefficient, A_i is the surface area of the wall face i .

The radiation coefficient from façade i to façade m is as follows :

$$h_r = \varepsilon \sigma F (T_i - T_m) (T_i^2 + T_m^2) \quad (11)$$

Where h_r is the radiation coefficient of facade i on facade m , ε is the emission coefficient of the wall facade, σ is Boltzman's constant, F is the form factor, T_i is the temperature of facade i , T_m is the temperature of facade m .

The heat equation is given by the following relationship :

$$\frac{\partial T}{\partial t} - \alpha \nabla^2 T = 0 \quad (12)$$

Where T is the quantity to be found, t is the time variable, α is the diffusivity constant.

For 2D solving, equation (12) is transformed into the following equation.

$$\frac{\partial T}{\partial t} - \alpha \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) = 0 \quad (13)$$

Where T is the quantity to be found, t is the time variable, x and y are spatial variables, α is the diffusivity constant.

Since the heat equation is a partial differential equation, the finite-difference method will be used to solve it. In numerical analysis, the finite-difference method is a common technique for finding approximate solutions to partial differential equations. It involves solving a system of relations linking the values of unknown functions at points that are sufficiently close to each other [26].

In this work, the finite-difference method is used, discretizing the spatial domain and the time interval ; x, y and t.

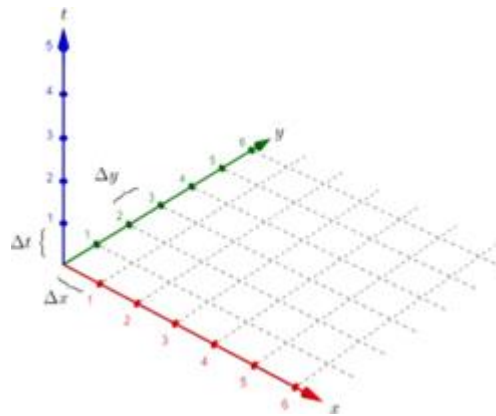


Figure 3. Discretization of spatial domain and time interval.

With the discretization of spatial domain and time interval, the heat equation becomes the following relationship.

$$T(x, y, t) = T_{i,j}^k \quad (14)$$

Where i, j and k are the steps of each difference for x, y and t respectively.

The heat equation (relation 14) can be written using the finite difference method as follows:

$$\frac{T_{i,j}^{k+1} - T_{i,j}^k}{\Delta t} - \alpha \left(\frac{T_{i+1,j}^k - 2T_{i,j}^k + T_{i-1,j}^k}{\Delta x} + \frac{T_{i,j+1}^k - 2T_{i,j}^k + T_{i,j-1}^k}{\Delta y} \right) = 0 \quad (15)$$

By posing Δx = Δy, the equation (15) gives the following equation.

$$T_{i+1,j}^{k+1} = \gamma (T_{i+1,j}^k + T_{i-1,j}^k + T_{i,j+1}^k + T_{i,j-1}^k - 4T_{i,j}^k) + T_{i,j}^k \quad (16)$$

With :

$$\gamma = \alpha \frac{\Delta t}{\Delta x^2} \quad (17)$$

Where γ is the stability coefficient, α is the thermal diffusivity, t is the time variable, x is the spatial variable, Δ is the Laplacian.

To obtain a solution to the heat equation, an explicit method will be used, so that it will always be numerically stable. The heat equation can be approximated by the following equation.

$$\Delta t \leq \frac{\Delta x^2}{4\alpha} \quad (18)$$

Where α is the thermal diffusivity, t is the time variable, x is the spatial variable, Δ is the Laplacian.

2.3. Simulation Parameters

The parameters used for the simulations are given in Table 1.

Table 1: Simulation parameters

| Parameter | Value |
|--------------------------------------|---|
| Combustion chamber surface | 0.75(m ²) |
| Radiation form factors for each wall | |
| Boltzman constant | 5.67 10 ⁻⁸ (Wm ⁻² K ⁻⁴) |
| Emissivity | 0.7(-) |
| SkyFactor | 0.5(-) |
| Ground factor | 0.5(-) |
| Waste radiation form factors | 0.310 |
| Heat capacity of materials | 765((Jkg ⁻¹ K ⁻¹) |
| Thermal conductivity | 350(wm ⁻¹ K ⁻¹) |
| Absorptivity | 0.1 (-) |
| Transmitting | 0.72 (-) |
| Density | 1820 (kgm ⁻³) |

3. RESULTS AND DISCUSSION

The aim is to study the heat exchange performance of a centimetric fixed-bed kiln for incinerating municipal waste, with pottery walls. The materials used for the walls are either natural laterite or kaolin laterite.

3.1 Fill Rate

Taking into account the dimensions of the furnace, an evolution of the filling rate of the furnace was carried out as a function of the weight of the waste. In the case of the present study, the fixed-bed furnace operates in a straight line, achieving a filling rate of 100% with 5 kg of solid waste. Figure 4 shows the mass of solid waste as a function of the filling rate.

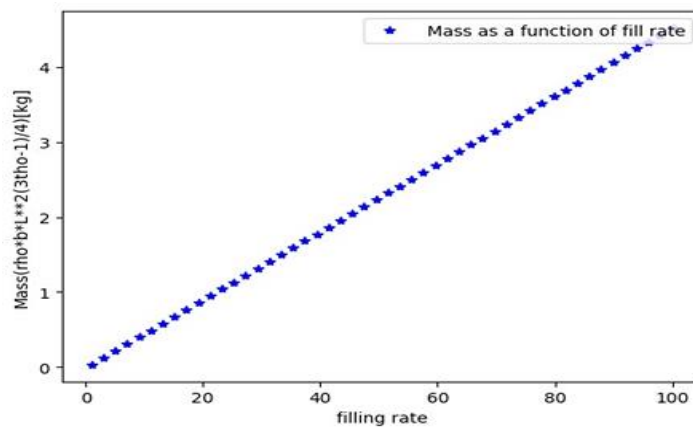


Figure 4. Mass as a function of fill rate

The numerical study is carried out considering two types of internal wall, namely natural laterite and kaolinite laterite. The initial waste temperature is 500°K. Figure 6 shows the evolution of the internal wall temperature.

3.2. Temperature Distribution Inside The Oven

Figure 5 shows the internal wall temperature curves.

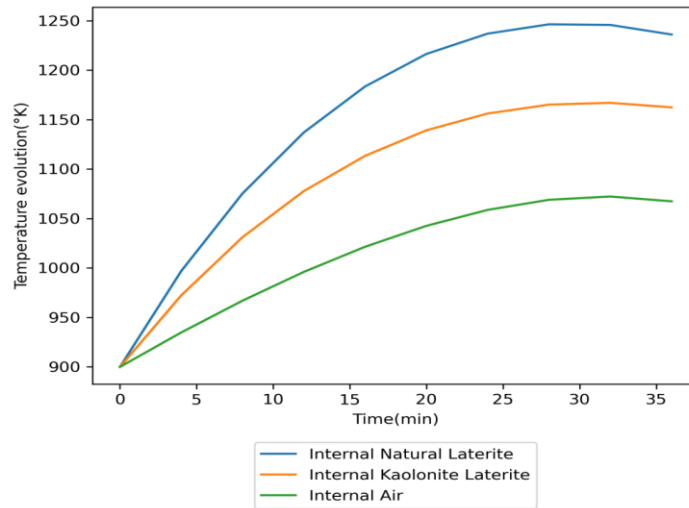
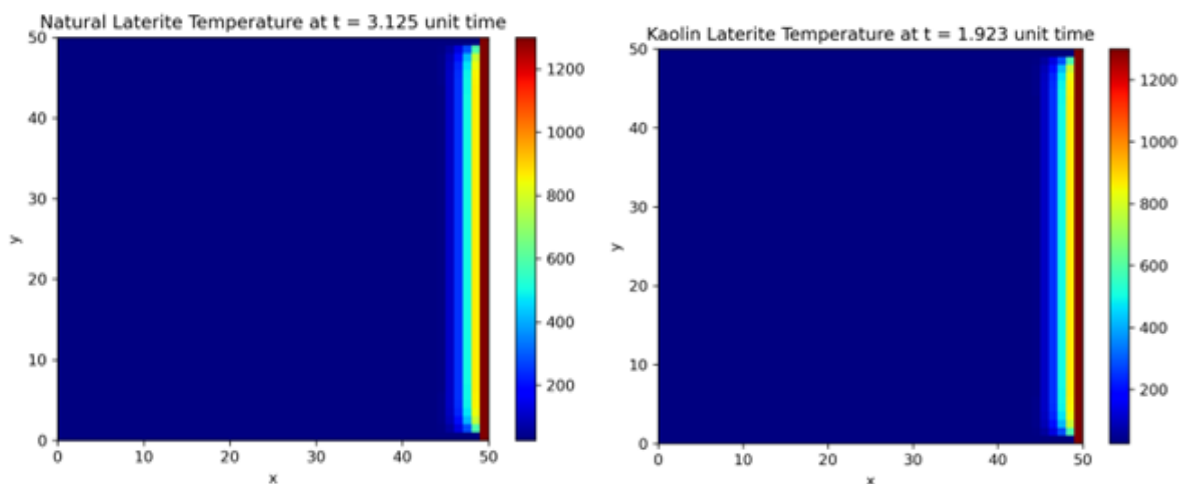


Figure 5. Internal wall temperature trend.

Figure 5 shows that the temperature curves over time are approximately the same for both natural and kaolinite laterite. For natural laterite, the variation in inner wall temperature reaches a maximum of 1225°C after 30 minutes of combustion. For kaolinite laterite, the internal wall temperature reaches a maximum of 1150°C, after 30 minutes of combustion. The internal air temperature reaches a maximum of 1050°C, after 30 minutes. These maximum temperatures are all reached around 30 minutes of operation.

However, after 35 minutes of operation, the temperature begins to drop in both cases, due to the degradation of the inputs, i.e. the waste to be burned and the exhaust gases. Under the effect of waste incineration, the air temperature inside a given wall gradually rises until it reaches its maximum, then gradually falls due to energy loss and total waste incineration. These results show that the furnace can reach favorable temperatures and extract enough thermal energy to convert it into useful energy as soon as possible. Appropriate insulation can therefore be used to reduce heat loss and increase furnace performance.

Figure 6 (a, b) shows the evolution of internal temperature across the walls per unit time. On the left, the evolution of the internal temperature of the natural laterite wall and on the right that of the kaolinitic laterite. Both results are recorded at equal temperatures, allowing us to see the difference in evolutionary time for each wall.



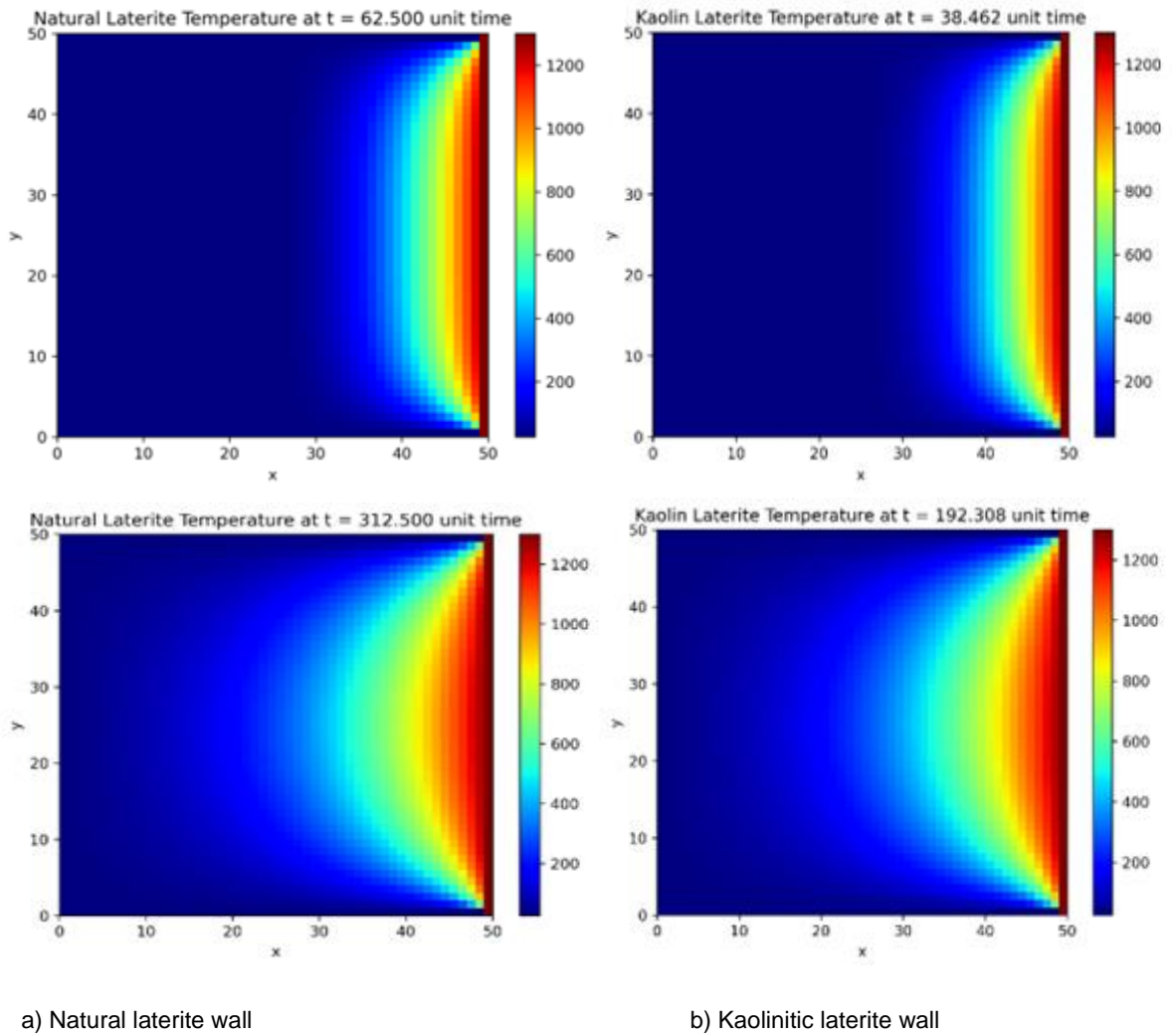


Figure 6. Temperature variation as a function of wall width.

We found that, for a given temperature unit, kaolinite releases temperature more rapidly along the wall than natural laterite.

As the comparison was based on three increasing temperatures, the results in Figure 6 shows a considerable and growing gap between the time differences per temperature. The higher the temperature, the greater the time difference, and the faster kaolinite dissipates heat than natural laterite.

3.3 Effect of Air Injection On Internal Temperature

The graph in figure 7 illustrates the variation in air flow rate on the evolution of internal temperature for the two types of wall.

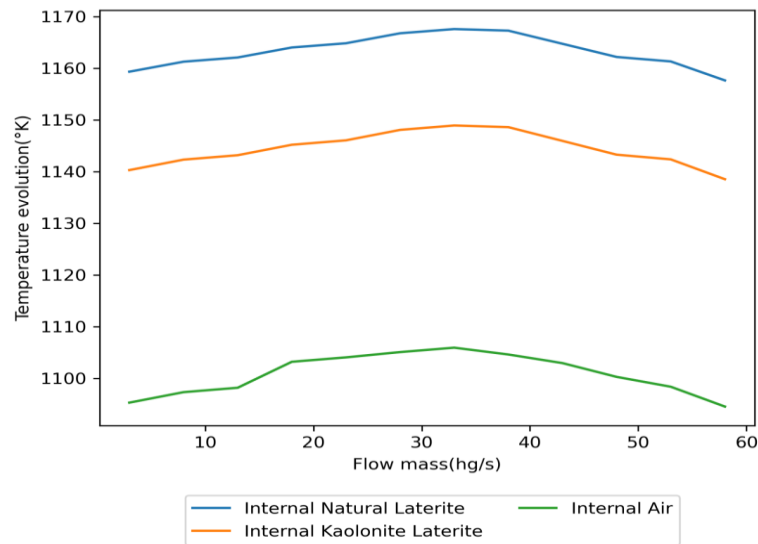


Figure 7. Evolution of walls temperature and air flow rate

At the beginning of the process, when the incoming air flow increases, the temperature also rises, due to the turbulence created in the furnace, which reduces the outgoing hot air flow. Subsequently, as the inlet and outlet flows tend to even out, there are numerous convection losses due to the outlet flow. This in turn reduces the temperature inside the furnace. The optimum internal temperature is around $35 \pm 5 \text{ hg. s}^{-1}$ of internal air flow.

CONCLUSION

The aim of this study is to choose between natural laterite and kaolinic laterite walls, the most suitable for a centimetric fixed-bed grate furnace with pottery walls, intended for the incineration of municipal waste. A simulation of the thermal evolution of a centimetric fixed-bed grate furnace with walls made of natural laterite and kaolin laterite was carried out. The results showed that furnace operation is optimal around 30 minutes of combustion and an internal air flow rate of $35\% \pm 5 \text{ hg. s}^{-1}$. However, there is a drop in performance from 35 minutes due to heat loss. Natural laterite tends to retain heat inside the kiln more than kaolinite, which dissipates too quickly. The performance of this type of kiln depends on both the absorption and conductivity of the wall. A thick wall layer and adequate control of air injection and gas ejection can reduce heat loss and increase furnace performance. Depending on the heat recovery method, the computational study concludes that the natural laterite inner furnace wall retains more heat inside than the kaolinite wall. In the event that the recovery method requires more heat transfer from the wall, the kaolinite inner wall would be better at transferring heat to an exchanger. This study has shown that this type of furnace can reach favorable temperatures and extract enough thermal energy to convert it into useful energy for conversion into electricity.

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Nomenclature

| | |
|--------------|---------------------------|
| T | Temperature |
| g_{im} | Wall thermal coefficient |
| ∂ | Partial derivate |
| m^{inlet} | Inletairmassflux |
| m^{outlet} | Outletairmassflux |
| λ | Conductivity |
| α | Coefficient of absorbtion |
| Δ | Laplacian |
| C_p | Heat capacity |
| ∇ | Divergence |

| | |
|---------------|--|
| Re : | Reynolds number |
| Nu | Nusselt number |
| Pr | Prandtl number |
| e | Thickness |
| γ | Validity coefficient |
| h : | Heat convection coefficient |
| h_r | Heat radiative coefficient |
| h^{inlet} | Inlet air heat convection coefficient |
| h^{outlet} | Outlet air heat convection coefficient |
| Σ | Sum |
| t | Time |
| x | x-coordinate |
| y | y-coordinate |
| m_m | Wall mass |
| ν | Viscosity |
| κ | Air conductivity |
| L | Wall length |
| ε | Emissivity |
| σ | Stefan-Boltzman constant |
| F | Form factor |

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