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Research article

Modeling, simulation and optimization of solid fuel bread ovens commonly used in developing countries

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ABSTRACT

In this work, we propose a mathematical model describing thermal behavior and heating process optimization of solid fuel bread ovens. Numerical simulation leads to temperature profiles of the oven. The design and implementation of an operating prototype permits us to obtain, with type K thermocouples, experimental temperature profiles in some points of the oven. There is a good agreement between the experimental results and those obtained from the numerical simulation of the proposed model. A permanent temperature value of 220 °C is reached in the baking chamber. It is obtained that the energy efficiency of the oven is 49%. Making use of the objective gain function, it is found that the optimal parameters of the oven are the following: 50 W as optimum operating value of the electric power of the blower, 3 m^2 as the optimum operating value of the total surface of the baking chamber; and 0.67 as the optimum operating value of the filling factor between the heating chamber and the baking chamber. The developed model serves to better understand the operation, the optimization and to rationally manage energy expenditure related to solid fuel bread ovens in developing countries.

1. Introduction

The use of locally fabricated bread ovens is widespread in several developing countries. But, most of the time, these ovens are technically constructed and the problem of finding optimal parameters for their construction is still open. One of the reasons explaining this lack of optimization is the fact that most oven configurations do not yet have good mathematical models based on heat exchanges. This has been a special problem for researchers interested by the optimization of bread ovens since there are impacts on the environment, society and economy (Vershinina et al., 2019). In particular, wood and charcoal obtained from natural forest resources are the main energy sources in several developing countries (Sagouong and Tchuen, 2018a,b). Because the bread baking sector is an important industrial activity, the question of optimizing the existing oven configurations is of special importance since it can lead to new designs (Kokolj et al., 2017; Khatir et al., 2015). Ovens heat bread by conduction, convection, and radiation. These three heat transfer modes should be included when modeling the temperature in the bread ovens. It has been found that at low air speeds, the radiation is predominant while at high speeds, the convection is much more important (Park et al., 2018; Nicolas et al., 2017; Isleroglu and Kaymak, 2016).

Technology designers look for simple and accurate prediction methods to simulate different processes in order to obtain appropriate material characteristics and operating conditions (Purlis, 2019). Indeed, the products qualities such as the temperature uniformity, heating efficiency, baking time, humidity and limited consumption of energy, highly depend on an efficient optimization of oven parameters (Chhanwal et al., 2019; Pask et al., 2014; Boulet et al., 2010; Khatir et al., 2012).

But this is a difficult problem and scientists are still looking for appropriate solutions taking reference in some other fields such as using charcoal in heating home (Zhao, 1992). An interesting work was conducted by Manhiça et al. (2012) on two types of bread ovens in Mozambique (ovens with direct and indirect heating using firewood). The experiment was conducted to obtain the temperature profiles in different locations of the oven. But, no mathematical model was derived in order to help in the optimization of the system.

The purpose of this work is threefold. Firstly, we propose a mathematical model for a parallelepiped bread oven using the mass exchange rate of the combustible and thermal rate variation of temperature in

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different compartments. The set of equations obtained is solved numerically using the fifth order Runge-Kutta method. Secondly, we fabricate a prototype of an oven, heat the prototype and develop an experimental measurement method to assess the temperature profiles in different parts of the oven. Thirdly, we make use of an optimization procedure, well known for heat exchangers, to find out good values of parameters leading to an optimal functioning of the oven.

The structure of the work is as follows. In section 2, we present the bread oven configuration and derive its mathematical equations. The experimental method on a prototype is presented in section 3. The main results are presented and discussed in section 4 and section 5 is devoted to the conclusion.

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2. Presentation of the oven configuration and mathematical modeling

2.1. Bread ovens in the field

Several visits to local bakeries allowed us to apprehend the types of bread ovens in the field, and to differentiate them according to their physical and technological characteristics.

Figure 1A, shows a traditional bread oven. Its main drawbacks are that it is large, bulky and can only make limited range of breads (Kumba bread, brioche). In the field, we found that ashes, fumes and Scrap coals are messy, and burns are very common due to a long exposure to fire during cleaning and charging. The heating process and baking time are uncertain since they depend on the quality and quantity of the fuel (wood, charcoal), the state and age of the oven, the time spent since the last heating, the air flow entering in the oven; and the products in the ovens. Geometry and volume depend on the level of difficulty for each constructor. The temperature is uncontrolled and is very unstable. The life of the oven is very short (after a few months the efficiency drops considerably).

The two images (Figure 1B and C) show the different parts of the industrial bread baking ovens most present in the Cameroonian market. These bread ovens are very expensive. The operation of the burner requires up to two important sources of energy (electricity and oil fuel). The position of the burner is always in front of the oven (Figure 1 B) and the work environment is very hot due to the burner position.

With the energy shortage in the country, added to multiple load shedding, voltage drops and multiple malfunctions in the public electricity grid, the owners of bakeries are obliged to use generators to satisfy their daily production. However, the contribution of diesel generators increases the costs of production, and somehow the cost of bread in the market.

So the quest for optimal oven working with charcoal remains open in our environment since in some parts of the country, electricity is lacking and one needs to bake bread. This is thus the aim of this study.

2.2. The bread oven configuration used in this work

Considering the above constraints, a bread oven configuration is proposed and studied in this work. It consists of three coupled subsystems described as follows:.

- A movable and independent burner (BR) of cylindrical shape with insulated walls where burning solid biomass (wood debris and charcoal occurs) are inserted.
- A heating chamber (HC): parallelepiped envelope made of local materials with insulating structure.
- A baking chamber (BC): parallelepiped enclosure with conductive walls of low thickness.

Figure 2 shows us a detailed view of the bread oven used in this work.

2.3. Mathematical modeling

2.3.1. Simplifying assumptions

In order to obtain a relatively simple and easily exploitable set of equations, it is essential to make some simplifying assumptions. These assumptions have been justified in a previous work related to similar processes (Zhao, 1992) and are given as follows.

- ➢ The charcoal is homogeneous.
- ➢ The gas flow in the furnace compartments is laminar and unidirectional.
- ➢ The temperatures of gases and charcoal on a section are taken by their average values.
- ➢ The oven walls have a uniform temperature in each compartment.
- ➢ The charcoal flow entering the burner is constant.

2.3.2. Mathematical model

The modeling will focus on heat transfers in all compartments of the oven. In the burner (BR), one considers the solid biomass (charcoal), the air flow and the walls. In the heating chamber (HC), the air flow and the walls are considered. Same for the baking chamber (BC).

Four systems are thus considered. The system 1 is the biomass in the burner. The air flow in the burner is the system 2. System 3 is the air flow in the heating chamber and finally the air flow in the baking chamber is system 4. The mass balance equation and the thermal balance relations are used to come out with the coupled Eqs. (1), (2), (3), (4), and (5).

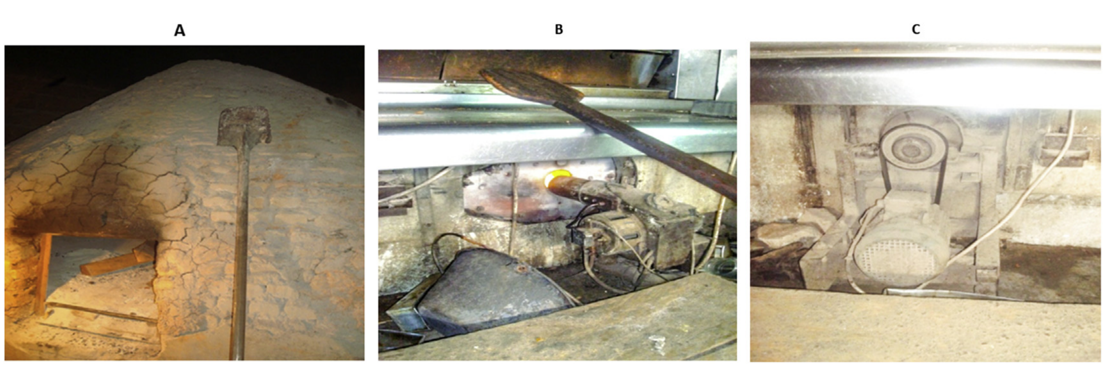

Figure 1. A. traditional oven, B. electrical oil burner, C. electrical blower.

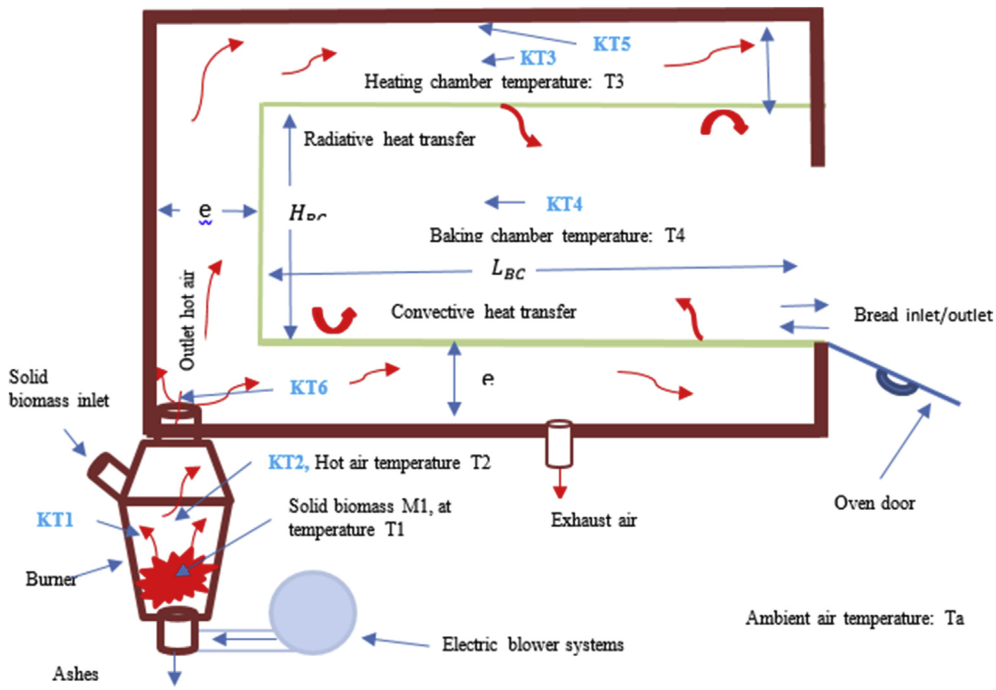

Figure 2. Bread oven configuration used in this work and thermocouples positions KT.

$$
\begin{cases}
\dfrac{dM_1}{dt} = -V_{mc}M_1 + (1 - \tau_v - \tau_c)\,D_1 & (1)\\[2ex]
C_1M_1\left(\dfrac{dT_1}{dt}\right) = V_{mc}M_1LHV + C_1D_1T_{01} - h_cS_{cv}(T_1 - T_2) - \varepsilon_1\sigma S_{ra}\left(T_1^4 - T_{WBR}^4\right) & (2)\\[2ex]
C_{T2}\left(\dfrac{dT_2}{dt}\right) = V_{mc}M_1LHV - \varepsilon_1\sigma S_{ra}\left(\varepsilon_2T_2^4 - T_{WBR}^4\right) - C_2D_2(T_{Ob} - T_a) - \dfrac{(T_2 - T_a)}{R_{BR}} & (3)\\[2ex]
C_{T3}\left(\dfrac{dT_3}{dt}\right) = C_2D_2(T_{Ob} - T_a) - \varepsilon_{HC}\sigma S_{HC}\left(\varepsilon_3T_3^4 - T_{wHC}^4\right) - \varepsilon_{BC}\sigma S_{BC}\left(\varepsilon_3T_3^4 - T_{wBC}^4\right) - h_{HC}S_{BC}(T_3 - T_{wBC}) - C_3D_{Ea}(T_{Ea} - T_a) - \dfrac{(T_3 - T_a)}{R_{HC}} & (4)\\[2ex]
C_{T4}\left(\dfrac{dT_4}{dt}\right) = h_{BC}S_{BC}(T_{wBC} - T_4) + \varepsilon_{BC}\sigma S_{BC}\left(T_{wBC}^4 - T_4^4\right) - \varepsilon_d\sigma S_d\left(T_{wd}^4 - T_a^4\right) - h_dS_d(T_{wd} - T_a) & (5)
\end{cases}
$$

where, M_1 is the mass of the solid biomass (charcoal) in the burner; T_1 is the average temperature of the solid biomass (charcoal); T_2 is the average temperature of the air flow in the burner; T_3 is the average temperature of air flow in the heating chamber and T_4 is the average temperature of air flow in the baking chamber.

The constants are:

T_{Ob} : outlet air temperature of the burner.
V_{mc}: combustion mass rate (rate of consumption of biomass in the burner).
LHV : lower heating value.
R_{HC}: thermal resistance of the heating chamber wall.
R_{BR}: thermal resistance of the burner wall.
D_1 : rate of charcoal flow entering the burner.
D_2: air flow rate at the burner outlet/baking chamber inlet.
C_T: thermal capacitance of the body.
τ_v : rate of ash in the charcoal.
τ_c: rate of volatile matter in the charcoal.

The radiation is the predominant heat transfer mechanism for industrial tunnel ovens. For this study, the radiation was described using the surface-to-surface *(S2S)* model as in Chhanwal et al. (2010, 2019) and Boulet et al. (2010). This model uses the modeling Eqs. (6), (7), (8), and (9). Eq. (6) gives us the model used for radiation model.

$$Q_{out\ w} = \varepsilon_w\sigma T_w^4 + (1 - \varepsilon_w)Q_{in\ w} \tag{6}$$

where $Q_{out\ w}$ is the radiosity, $Q_{in\ w}$ is the lighting of the surface; T_w, is the wall temperature; ε_w, is the emissivity factor of the surface. The S2S model applied on different oven walls, can permit us to have the wall temperature of different parts of the oven.

The temperature of the burner wall is given by Eq. (7).

$$T_{wBR} = \left(T_1^4(2\varepsilon_{wBR} - 1)S_{ra}/(\varepsilon_{wBR}S_{wBR})\right)^{1/4} \tag{7}$$

where S_{CS}, is the charcoal radiative surface into the burner, S_{WBR} is the burner wall surface.

The temperature of the heating chamber wall is given by Eq. (8).

$$T_{wHC} = \left((1 - (1 - \varepsilon_{wHC})S_{BC}/(\varepsilon_{wHC}S_{HC}))\left(\varepsilon_3T_3^4 + (1 - \varepsilon_3)\varepsilon_{wBC}T_{wBC}^4\right)\right)^{1/4} \tag{8}$$

where, S_{BC} is the baking chamber radiative surface; S_{HC} is heating chamber radiative surface.

The continuity of the heat flow through the wall of the baking chamber leads to the following temperature expressions (9) and (10):

$$T_{wBC} = \left(\left(\varepsilon_3 T_3^4 + T_4^4\right)/2\right)^{1/4} \tag{9}$$

$$T_{wd} = \left(\left(\varepsilon_4 T_4^4 + T_a^4\right)/2\right)^{1/4} \tag{10}$$

2.3.3. Thermal parameters and numerical method

The gases flow rate from the wind tunnel (in kg/s) is expressed as in (Khatir et al., 2013; Bashir et al., 2019) by Eq. (11) below

$$D_2 = P_m / \Delta p \tag{11}$$

where P_m is the mechanical power of the blower and Δp is the pressure of gases passing through the burner. The charcoal gas convective surface is expressed as (Zhao, 1992) by Eq. (12):

$$S_{cv} = S_d * M_1 \tag{12}$$

where, S_d, is the surface density of charcoal (kg/m^2), and M_1 is the masse of the charcoal (in kg)

The convective heat coefficient is expressed by Eq. (13)

$$h_{BC} = h_{HC}\,(T_3 - T_{wB}) / (T_{wB} - T_4) - \varepsilon_{BC}\sigma\left(T_{wB}^2 + T_4^2\right)(T_{wB} + T_4) \tag{13}$$

The fixed values are as follows: $T_1(0) = 37°C$, $T_2(0) = 37°C$, $T_3(0) = 37°C$, $T_4(0) = 27°C$, $T_a = 27°C$, $M_1(0) = 0{,}0425\ kg$.

The set of Eqs. (1), (2), (3), (4), and (5) and their associated thermal parameters are solved using the fifth order Runge-Kutta method.

2.4. Optimization procedure

2.4.1. Heat flux and oven efficiency

Knowledge of different temperature profiles can permit us to plot the thermal heat generated by the combustion and the heat flux exchanged trough the oven (Zhao, 1992). It is given by Eq. (14)

$$P_{BR} = M_1 * V_{mc} * LHV \tag{14}$$

where, V_{mc} is the mass velocity of combustion (kg/kg.s), M_1 is the charcoal mass in the burner (in kg) and LHV is the lower heating value (in J/kg). The thermal power transmitted to the baking chamber is given by Eq. (15):

$$P_{BC} = h_{BC}S_{BC}(T_{wB} - T_4) + \varepsilon_{BC}\sigma S_{BC}\left(T_{wB}^4 - T_4^4\right) \tag{15}$$

Consequently, Eq. (16) defines the oven efficiency

$$\eta = P_{BC}/P_{BR} \tag{16}$$

2.4.2. Heating process optimization

The objective function (OF) in this context finds its meaning in the way of understanding the gain of the heat flux when a parameter of the system varies. This corresponds to the difference between the profits and the thermal losses, during all the bread oven heating process. The analysis of this function leads to the limits on parameters values at which the bread oven begins to lose energy. This OF is defined here as the difference between the heat flux transferred to the baking chamber (P_{BC}) and the total heat flux lost by the heating chamber (PL_{HC}). Eq. (17) gives its expression

$$OF = P_{BC} - PL_{,HC} = h_{BC}S_{BC}(T_{wB} - T_4) + \varepsilon_{BC}\sigma S_{BC}\left(T_{wB}^4 - T_4^4\right) - C_3D_3(T_{Ea} - T_a) - (T_3 - T_a)/R_{HC} \tag{17}$$

The filling factor FF in this context refers to the volume occupancy of the baking chamber relative to the heating chamber, thus describing the ratio of the external volume of the baking chamber to the internal volume of the heating chamber. This factor is independent of the content of baking chamber and is expressed by the following formula (18):

$$FF = V_{BC}/V_{HC} = (L_{BC}W_{BC}H_{BC}) / ((L_{BC} + e)(W_{BC} + 2e)(H_{BC} + 2e)) \tag{18}$$

where, V_{BC} is the baking chamber volume; V_{HC} is the heating chamber volume, e represents the thickness of the gap along the length, the width and the height between the heating chamber and the baking chamber. L_{BC}, W_{BC} and H_{BC} are respectively the length, the width and the height of the baking chamber. Knowledge of the filling factor FF indicates the permissible relative spacing between the two chambers which obeys the optimal limits of the bread oven heating process.

Once the objective function (OF) has been established, the mathematical formulation of the optimization consists in finding the maximum of the OF function during the entire heating period $t \in [0, 11800s]$, for each value of the parameter to be considered, using the parameters given in Eqs. (19) and (20).

$$\begin{cases} OF_{S_{BC}} = MAX_{PS_{BC}}\ OF(t, S_{BC}) \\ S_{BC} \in \left[1.5m^2,\ 8.5m^2\right] \end{cases} \tag{19}$$

-the filling factor, $FF \in [0{,}35,\ 0{,}85]$,

$$\begin{cases} OF_{FF} = MAX_{FF}\ OF(t, FF) \\ FF \in [0.35,\ 0.85] \end{cases} \tag{20}$$

The useful surface of the exchanger is $S_{BC} \in [1.5m^2,\ 8.5m^2]$.A similar reasoning is applied to the burner on the electric power $p \in [14W, 84W]$. This leads to the corresponding objective function given in Eq. (21)

$$\begin{cases} OF_P = MAX_P\ OF(t, p) \\ p \in [14W,\ 84W] \end{cases} \tag{21}$$

The minimum and maximum limit values of the electric power, the area of the exchanger, and the filling factor are limit data for the operation of the system. This optimization procedure is similar to the description made by Pardalos et al. (2017).

3. The experimental method

3.1. Oven description and physical parameters

In the literature, many charcoal ovens used for domestic cooking have a cylinder or a truncated cone shape (Sagouong and Tchuen, 2018a,b). The in-furnace mixing and the combustion characteristics depend on the furnace chamber shape (Tu et al., 2015). Table 1 presents the values of the geometric dimensions and the shape of each part of the oven designed in this study (see Figure 2).

3.2. Materials and method for data collection and set up

Figure 3 shows the experimental prototype used to collect the data (the experimental values of the temperatures in different compartments). Table 2 indicates the positions of the thermocouples inside the oven.

- **Materials:** Operating bread oven prototype (biomass Burner, heating chamber, baking chamber), electric box (electric blower system, wattmeter, a main solar photovoltaic system, coupling with a backup system), data acquisition box (Computers, Arduino microcontrollers, the max6675K Thermocouples amplifier (each having digital serial data output and supplied along with K-type thermocouple)).
- **Preparation:** the charcoal used is of the « semi-heavy » type. It is dried, sorted according to an average diameter of 3 cm. Then, it is classified in small packages of 0.0425 kg on average. The oven is preheated during three hours a day, and this for a whole week.

Table 1. The geometric dimension and shape of the different oven parts used for simulation.

Burner		Heating chamber		Baking chamber	
Internal Shape:	Truncated cone	Shape:	Parallelepiped	Shape	Parallelepiped
Internal radius R1	0.070 m	Internal Length	$L_{BC} + e = 0.720$ m	Length	$L_{BC} = 0.680$ m
Internal radius R2	0.084 m	Internal Width	$W_{BC} + 2e = 0.43$ m	Width	$W_{BC} = 0.350$ m
Height of burner	0.600 m	Internal Height	$H_{BC} + 2e = 0.30$ m	Height	$H_{BC} = 0.220$ m
Thickness of wall	0.250 m	Thickness of gap	e = 0.04 m		
Power of blower	38 W	Thickness of wall	0.100 m		

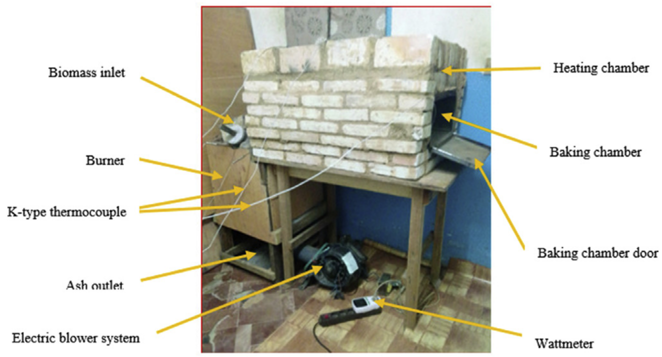

Figure 3. Operating prototype bread oven.

- **Data acquisition process:** Once the oven is preheated, the K-types thermocouples are correctly inserted in the indicated points of the oven (see Table 2). The electric blower system is activated. Then the data acquisition system is activated. At the end, we first introduce 0.0425 kg of charcoal into the burner before the charcoal charging system. Temperature data is recorded simultaneously for each oven compartment in real time, for three hours from 10:00 AM to 1:00 PM. This experiment is repeated for a week, at the same times of the day, and during the same time interval.
- The amount of biomass needed to maintain this temperature is 42.5g/mn.
- In this work, we analyze the thermal behavior of the oven heating (step before placing the bread in the oven) because for indirect heating ovens using solid biomass, the heating process is a delicate step, which cannot be easily predicted. Thus, this step is necessary before one places the bread in the oven.

The tests were carried out in two phases. Seven tests were made (one test per day) for the first phase which was for the conditioning of the oven.

4. Results and discussion

4.1. Temperature profiles of heating process

The numerical results obtained from the set of Eqs. (1), (2), (3), (4), and (5) are plotted in Figures 4, 5, and 6.

In the burner, we observe two temperature profiles for the wall (TWBR in Figure 4) and inside the burner (T2 in Figure 4). It is found that the heating is almost instantaneous leading to 800 °C in less than 300 s. The rise in temperature reaches a saturation value at 1000 °C after 2000 s only. The heating of the gases in the burner takes more time and reaches 800 °C after 2000 s and the saturation value of 870 °C is reached. The overlap of the numerical and experimental results are fairly consistent. These results are similar to those obtained by Zhao (1992), and those of Sagouong and Tchuen (2018a,b).

In the heating chamber, two other temperature profiles are presented. Figure 5 A shows the time variation of the gas while Figure 5 B corresponds to the variation of the wall temperature. One finds that both temperatures have almost the same behavior. This is explained by the proximity of the gas flow in the heating chamber. Also, we note that the heating of the gases and the walls is progressive, up to the value of 300 °C which is reached after 2000 s. Beyond this value, the temperature changes slightly, and then reaches a saturation value of 330 °C. As one can also see here, the experimental and the numerical results are almost the same. They are compatible with the experimental results of Manhiça et al. (2012), who also obtained a permanent value of 340 °C for the similar model.

As concerned the baking chamber, the temperatures profiles are given in Figure 6 A. Unlike what was observed in the burner and in the baking chamber, the rise in temperature is not instantaneous. This delay is understood by the fact that the heating of the baking chamber is indirect. Indeed, the air/air exchanger of the baking chamber receives the flow of heat from the heating chamber before conveying it by convection and by radiation to the baking chamber. After about 2000 s, the temperature attains saturation at about 220 °C. Our result is comparable to the one obtained in References (Chhanwal et al., 2019; Ploteau et al., 2015; Mistry et al., 2006, 2011; Paton et al., 2013). These results reflect the reality of what happens in bakeries using these ranges of ovens. In Figure 6B, we plot the temperature profile at the outlet of the burner. It is observed that the heating of the air flow reaches a saturation of 450 °C in 2200 s. It is also interesting to mention that the model reflects the thermal behavior of the temperature profile of the outlet air of the burner.

4.2. Heat flux generated in the oven

Knowing the oven temperatures for different points allows us to plot the heat fluxes generated and exchanged in the oven. Figure 7A presents the thermal power (P_{BR}) generated by the reaction and the useful thermal power (P_{HC}) provided by the burner. We observe that the thermal power developed by the reaction evolves up to a limit value of 3200 W while the

Table 2. K types thermocouple positions in the oven.

Measured temperature	Type K thermocouple position (KT)
T_{wBR} : average temperature of the burner walls	On the burner walls (KT1 on Figure 2)
T2: average air temperature in the burner	In the center of the free cavity of the burner (KT2 on Figure 2)
T3: average air temperature in the heating chamber	In the center of the heating chamber (KT3 on Figure 2)
T4: average air temperature in the baking chamber	In the center of the baking chamber (KT4 on Figure 2)
T_{wHC} : average temperature of the heating chamber walls	On the walls of the heating chamber (KT5 on Figure 2)
T_{ob} : average temperature of the air leaving the burner	At the burner outlet/heating chamber inlet (KT6 on Figure 2)

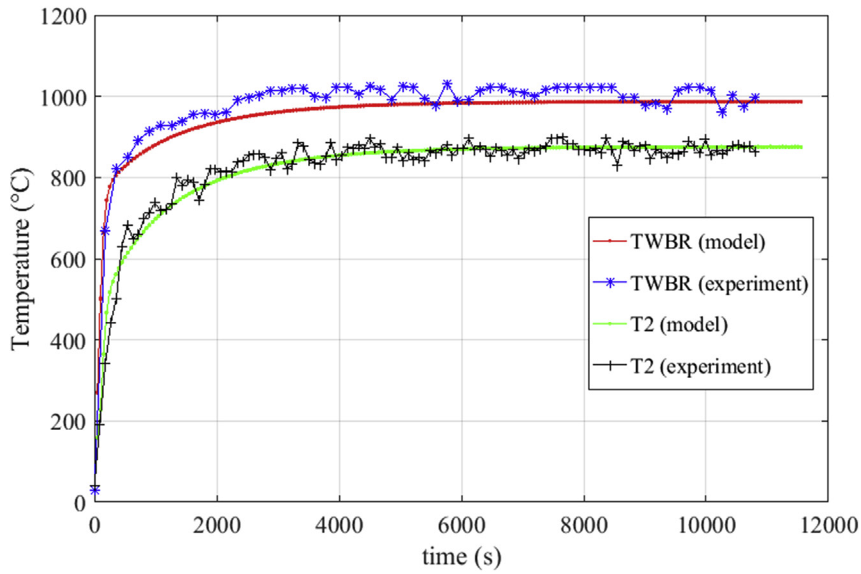

Figure 4. Temperature profile of burner wall (TWBR), and Temperature profile of air in the burner (T2).

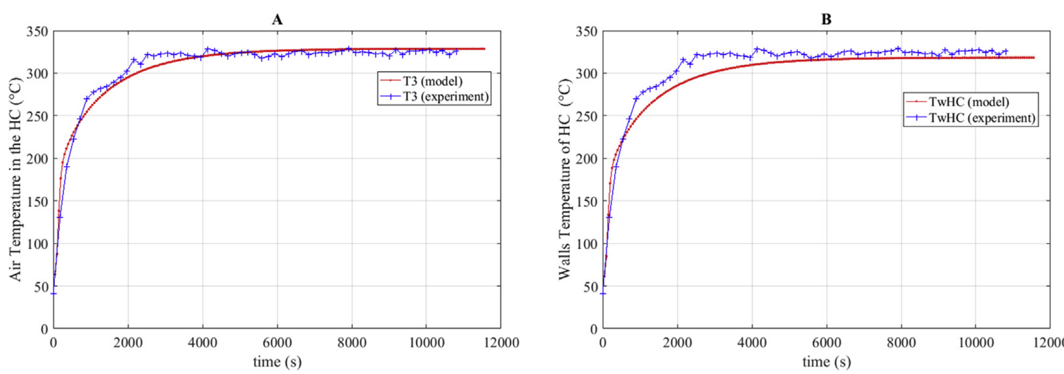

Figure 5. A: temperature profile of the air in the heating chamber, B: Temperature profile of the heating chamber wall.

useful thermal power reaches its limit at 2900 W. The difference of 300 W between these values is due to the fact that a part of the heat flow produced by the reaction is lost by the burner at the walls. Figure 7B shows the time evolution of heat flow temperature transferred to the baking chamber (P_{BC}), and lost by the heating chamber (PL_{HC}). We observe a saturation at the steady state of 1550 W for the heat power transferred in the baking chamber.

4.3. Optimum oven efficiency and optimum oven parameters

Evaluation of different heat flow allows us to determine the behavior of thermal efficiency and objective function of the oven. They appear in Figure 8.

Figure 8 A presents the time evolution of the oven efficiency. It is found that its maximal value is about 0.85 after only 400 s from where it

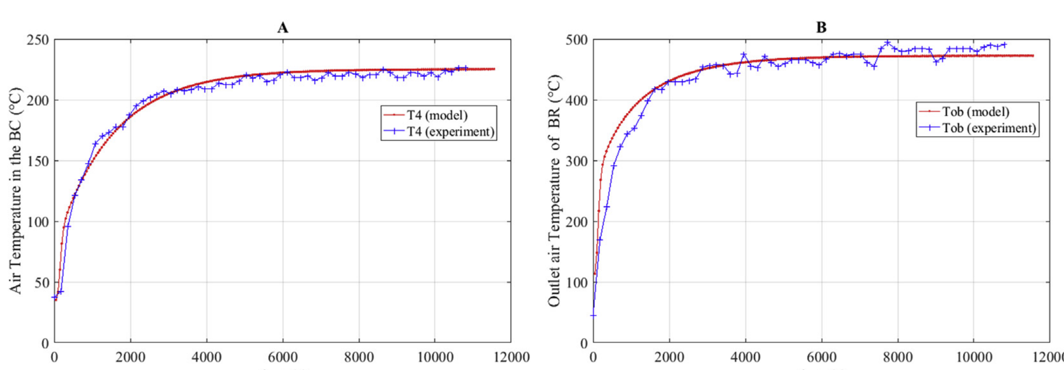

Figure 6. : A: Temperature profile in the baking chamber. B: Temperature profile of the outlet air from the burner.

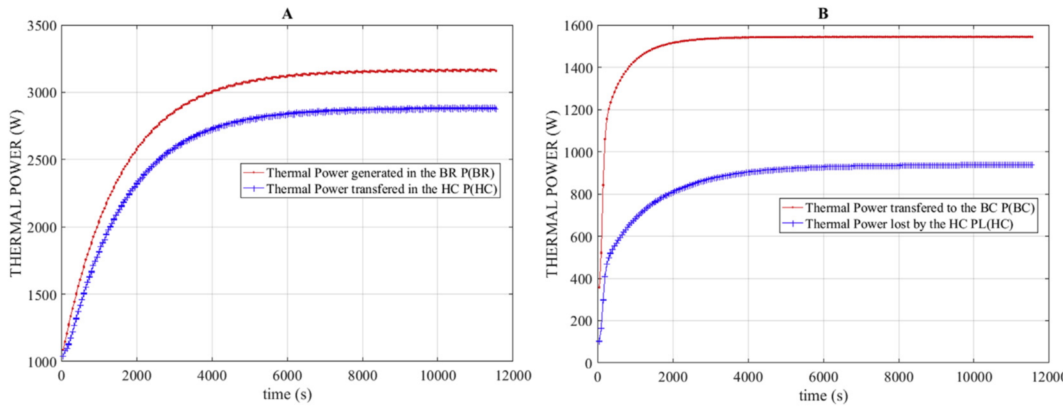

Figure 7. A.: Thermal power in the burner, B. Thermal power transferred to the backing chamber.

decreases till 0.40 after 4000 s. This result is explained by the fact that the heating is carried out with an empty oven. So the oven has a limited energy storage capacity. Observing this behavior can tell us the optimal time for baking bread. At this precise moment of the heating phase, the temperature in the center of the oven is about 100 °C (see Figure 6A) and the efficiency is at its maximum of 0.85. Also, an efficiency of 0.70 will be observed at a temperature of 170 °C after only 1000 s. This can be considered as the reference value for the charging of bread dough in the oven.

The objective function is plotted in Figure 8B. It optimal value is 0.65. This value is important for local bread oven technicians and builders. Indeed, once the optimal physical dimensions of the baking chamber are known, the value of the thickness of gaps between the two chambers can be determined. Using this value in Eq. (17), one finds that the approximate value of 4 cm as the optimal value of the gap between the two chambers.

From Figure 8 C, one finds that 3 m^2 is the optimal value of the baking chamber area. Consequently, one can fix two dimensions of the chamber and calculate the third one from the value indicated above since the total area depends on the three dimensions. This is an important information for the design and fabrication of local bread ovens. Indeed, most of the

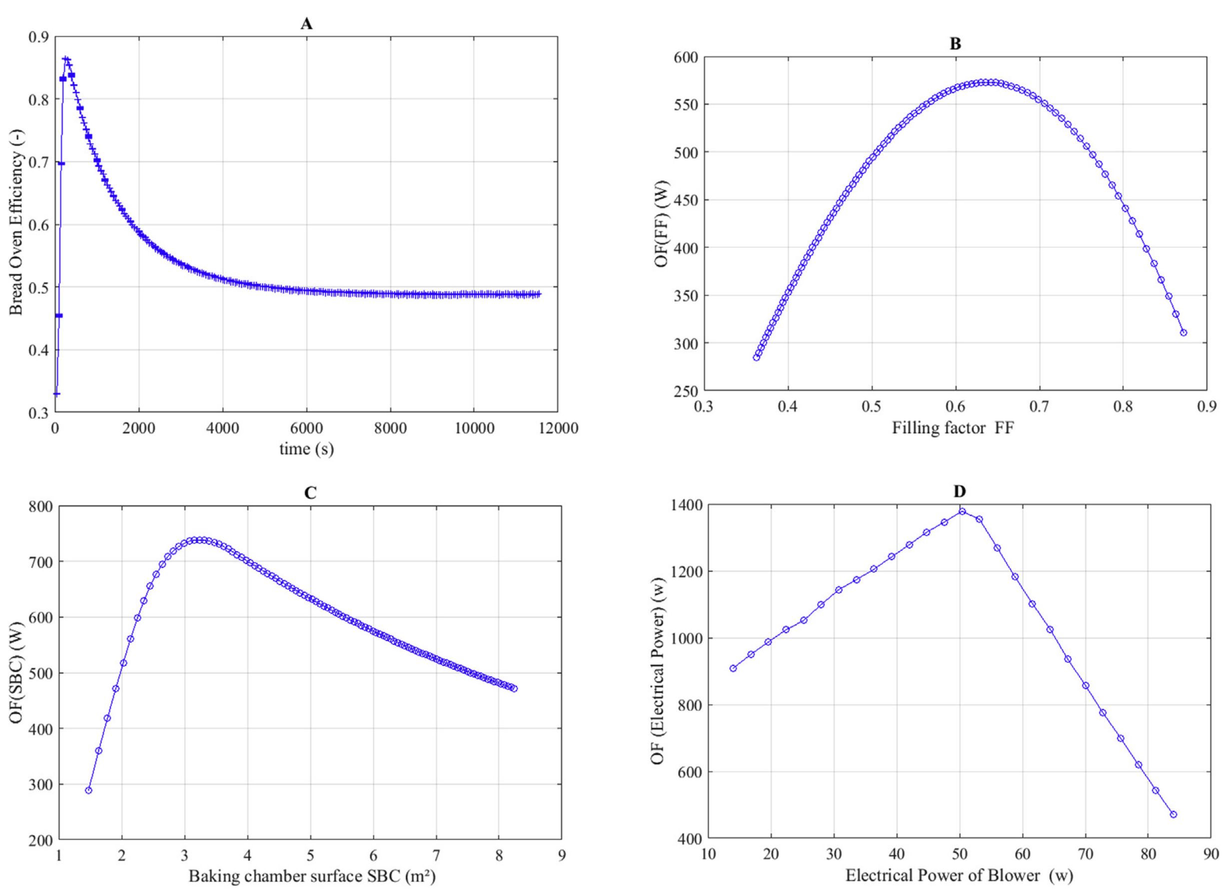

Figure 8. A: Bread oven efficiency. B. Objective function of filling factor FF. C: Objective function of the surface. D. Objective function of the electrical power of the blower.

time, the technicians fabricating locally the ovens do not have such information for the optimization.

Figure 8 D presents 50 W as the optimal value of the electric power for the blower system of the burner. Beyond this value, the losses caused by the wind tunnel become greater than the benefit that this wind tunnel brings. The value of this electrical power is quite low, compared to the powers of the electric blowers of modern bread ovens. This is beneficial for the owners of local bakeries in regions where electricity provision is difficult. In an environment where the solar irradiation is 3.9 h/d, a 100 Wp solar panel will produce approximately 250 Wh/d of energy. An operation of 5 h of baking will consume an electrical energy of 250 Wh. This approach is economic and ecological for remote areas in developing countries. This will allow any local producer to settle cheaply and have hot bread every day.

5. Conclusion

This work aimed at setting up a mathematical model describing the thermal behavior of the heating process of bread ovens commonly used in developing countries. Starting from the observations obtained after multiple visits to local bakeries, we have proposed a mathematical model with real operating conditions. The numerical simulation of the mathematical model obtained allowed us to obtain the temperature profiles of the oven. An oven prototype has been constructed. The temperature profiles obtained from the K-type thermocouples are consistent, qualitatively and quantitatively with those obtained from the mathematical model, and those existing in the literature for similar works. The exploitation of the results obtained from our model indicates a total heat flux of about 3200 W generated by the burner and the other different thermal heat flux exchanged through the oven. The oven efficiency has been determined and is about 0.49. The objective function of the power gain leads to an optimal power of 50 W for the electric power of the blower. Also, we obtain the optimum physical parameters of 3 m^2 total area of the baking chamber and a filling factor of 0.67 between the heating chamber and the baking chamber. In view of the small quantity of ash discharges, the low consumption of electrical energy, the absence of exhaust smoke discharge, and the low cost of acquiring local building materials, the oven prototype designed and studied in this work is economical, ecological and of low cost. The mathematical model developed here will contribute to optimally design and construct local bread ovens, lead to good energy management and to a good bread baking process.

Although the above results are interesting and constitute a good contribution in the design, modeling and experimental study of affordable and locally made bread ovens, there is a need for further study such as finding the way of increasing the oven efficiency and the inclusion of the bread baking process in the system. Indeed, the baking chamber was empty in this work. There is no doubt that the presence of bread in the chamber will modify the temperature profiles in the baking chamber and has impact on other temperature profiles.

Declarations

Author contribution statement

C. F. Kouemou Hatou: Conceived and designed the experiments; Performed the experiments; Analyzed and interpreted the data; Wrote the paper.

G. Tchuen: Analyzed and interpreted the data.

P. Woafo: Analyzed and interpreted the data; Wrote the paper.

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Data availability statement

Data will be made available on request.

Declaration of interests statement

The authors declare no conflict of interest.

Additional information

No additional information is available for this paper.

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