Computational Fluid Dynamics Simulation of the Flow Field in Wood-Fired Bakery Ovens

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Abstract

The circulation of hot gases within a bakery oven is used to describe and predict the exchange of heat and its influence on the final product. A bi-dimensional model was used, together with a CFD model, to estimate the flow pattern inside an oven by varying the velocity of the airflow in order to simulate changes in the combustion chamber. Acceptable agreement was found in the comparison between the models and the measured temperatures. The inlet velocity and the geometry affect the flow pattern in a baking oven. Low velocity cause low levels of circulation, which implies better conditions in the baking process. High velocities the temperature decreases in the oven due to excess air even with a high recirculation of gases. During the feeding process, the overall heat transfer was also affected by a changed effective thermal conductivity and recirculation of hot gases in the baking oven.

Keywords: flow pattern, airflow, physical modelling, wood-fired bakery oven

1. Introduction

The performance of a wood-fired bakery oven (WFBO) depends, to a large extent, upon the efficiency of the way in which energy is converted. Such a system includes the conversion of chemical energy in the fuel to thermal energy, and the efficiency with which the thermal energy is transferred to the baking chamber. The system for transporting combustion products through a WFBO and the types of material used in the construction of the furnace are important parameters that impact its overall performance. To some extent, short-term variation is attenuated by the storage capacity of the brick walls. Figure 1 shows a common design of a WFBO in Mozambique.

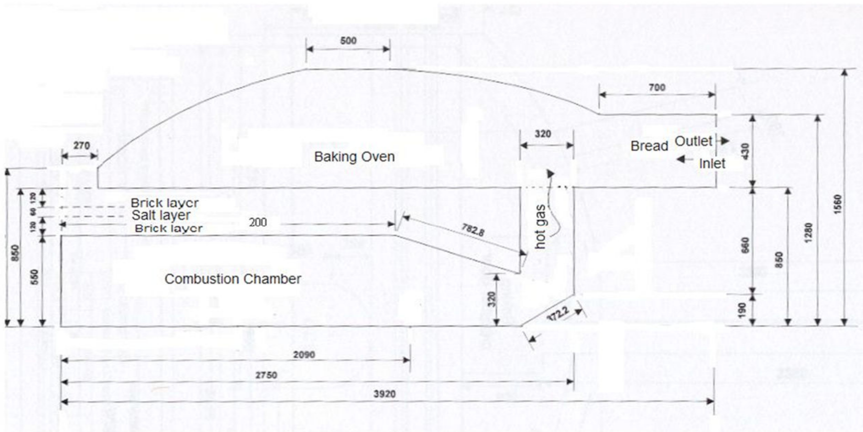


Figure 1: Cross-section of a semi-direct, wood-fired bakery oven.

Designing an optimum bakery oven requires an understanding of the complex interaction between the different processes that occur, such as: combustion; heat transfer and fluid flow in the furnace and the construction material.

The combustion process is dependent on the physical-chemical properties of the fuel (e.g. size, shape, density and moisture content), amount of fuel and mode of air supply (primary and secondary air). The conditions of the immediate environment of the surroundings, such as temperature, wind, particle size and humidity are also important; some of these aspects have been the objective of previous studies on the combustion of biomass. Measuring the heat released using a calorimeter was investigated by modelling fire growth and the endurance of the wood structure using the rate of burning (Tran & White, 1992). The results show that the burning rate depends on several factors that complicate the modelling of wood combustion, e.g. the variability of material between, and within, wood species, the approximation necessary when modelling the chemical properties and the moisture content, along with external factors, such as the boundary conditions. Fluid dynamic in wood combustion research present a method for designing small-scale model experiments for optimizing the geometry of wood-burning chambers (Baillifard et al, 2008). The Reynolds number in the combustion chamber was identified as playing an important role because it not only influences the flow pattern of the air but also the way in which it is mixed. Rath et al., (2003) estimated the average values of the heat of pyrolysis of wood observing that an increase in the char yield resulted in a decrease in the total endothermic heat of pyrolysis and, eventually, in a shift from an endothermic to an exothermic process.

Isothermal physical modelling is an experimental method that involves small-scale water models to understand the correspondent flow behaviour in large units. It could for example be non-isothermal flow and combustion in furnaces or boilers. It enables an opportunity to vary different design parameters, which would be difficult to perform on a real process plant because of the cost and the risk of damage. The range of the experimental technique is very large and new techniques are being developed as new instrumentation and material become available. Physical modelling is a valuable technique for the study of e.g. combustion flow pattern; pressure and velocity distributions found in simulation of mixing and combustion applications (Baranski, 2002; Hwang et al, 2000; Lucas & Blasiak, 2000).

Although a wood-fired bakery oven was used in this work, the discussion applies to most wood-fired devices. Baking is based on simultaneous heat and mass transfer, which cause the series of physical, chemical and structural transformations that are essential to the quality of the final product.

It is a great challenge to produce high quality bread because baking is an irreversible process and production of low quality bread is thus economically unfavourable (Chhanwal et al, 2011). Computational Fluid Dynamics (CFD) has been increasingly applied for design and development of baking oven as well as baking processes (Boulet et al, 2010; Therdthai et al, 2003 & 2004; Wang & Yan, 2007; Wong et al, 2006), and is a tool which can predict fluid flow, heat and mass transfer, chemical reactions and other phenomena. This work was carried out to understand the aerodynamic aspects of a WFBO by applying well proven techniques: bi-dimensional cold flow physical modelling and CFD modelling. The study includes how the airflow affects the flow pattern inside a WFBO and predicts the quality of the produced bread.

2. Modelling and Simulation

2.1. 2D Isothermal Physical Modelling

The present study includes visualization of the flow in the baking chamber of the wood-fired bakery oven. It was performed to achieve qualitative information and measurement of the flow characteristic. The first step was to design the physical model relative the real equipment (scale 1:10). An aluminium plate with a thickness of two millimetres and a width of five millimetres was used for the two-dimensional model. It was arranged on a horizontal plane in a water table. Three valves controlled the velocity of the recirculation water in model and aluminium powder (with a diameter of about 40 μm) was used to visualize the flow pattern. The material does not dissolve in water and it has a high reflective factor and could therefore be reused almost indefinitely. The disadvantage of the powder is that it tends to sink at long running times. The streamline pattern was possible to obtain by allowing fairly long exposures time, on ASA 100 films as shown on Table 1. If the primary aim of the model is flow visualization, it is important that there is a clear access of the areas of interest.

Table1: Experimental parameters scheduled for bi-dimensional experiments

Water velocity [m/s] on inlet of the model	0.020			0.023			0.030			0.038		
Light regulation	5.6	8	11	5.6	8.0	11	5.6	8	11	5.6	8.0	11
Exposures time (s)	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4
	1/8	1/8	1/8	1/8	1/8	1/8	1/8	1/8	1/8	1/8	1/8	1/8
	1/15	1/15	1/15	1/15	1/15	1/15	1/15	1/15	1/15	1/15	1/15	1/15

The velocities were calculated from a typical wood consumption in wood-fired bakery ovens (18 kg/h) where the amount of excess air varied between 2 and 4, which is common in poor combustion conditions (Lucas & Blasiak, 2000, Nussbaumer, 2003). In the velocity calculations it was assumed that the combustion of wood was complete and that atmospheric air entered by natural convection to the combustion chamber.

2.3. Mathematical Modelling

Turbulent flow without any chemical reaction was assumed for the flow of hot gases from the combustion chamber to the baking section in the oven. The turbulent flows are characterized by fluctuating velocity fields primarily due to the complex geometry and/or high flow rates. The Navier-Stokes equations can be solved directly for laminar flows but, for turbulent flows, the direct numerical simulation (DNS) with full solution of the transport equations at all lengths and time scales is too computationally demanding. The reason is that the fluctuations can be of small scale and have a high frequency, i.e. the Re number is below 5000 (Andersson et al, 2009). An alternative is, therefore, to transform the Navier-Stoker equations on the small eddies which directly simulate the Reynolds averaging and filtering processes (Wang and Yan, 2007). The Reynolds-averaged Navier-Stokers (RANS) equations represent transport equations for the mean flow quantities, and are derived by splitting the instantaneous properties in the conservation equations into two parts; mean and fluctuating components. The most common RANS models employ the Boussinesq (eddy viscosity concept, EDC) to model Reynolds stresses (Andersson et al. 2009; TorII and Yanagihara, 1989). The hypothesis states that an increase in turbulence can be represented by an increase in effective fluid viscosity, and that the Reynolds stresses are proportional to the mean velocity gradient by this viscosity. Models based on this hypothesis include e.g. Spalart-Allmaras, standard k-ε, RNG k-ε, Realizable k-ε, k-ω and its variants (Fluent 6.1 User Guide, 2004).

To mimic a wood-fired bakery oven, the selected model must be valid at low Reynolds number and still include turbulence due to the complex geometry that cause fluctuation in the velocity fields. A mathematical model that can be used to describe the flow pattern at this condition is the two-equation k-ω, where the turbulence specific dissipation, ω, is used as a length determining quantity. This quantity is defined by $\omega \propto \varepsilon/\kappa$, and it should be

interpreted as the inverse of the time scale on which dissipation occurs. The modelled k-equation is: dissipation occurs. The modelled k-equation is:

$$\frac{\partial \kappa}{\partial t}+\langle U_j\rangle\frac{\partial \kappa}{\partial x_j}=\nu_T\left[\left(\frac{\partial\langle U_j\rangle}{\partial x_J}+\frac{\partial\langle U_j\rangle}{\partial x_i}\right)\frac{\partial\langle U_i\rangle}{\partial x_J}\right]-\beta\kappa\omega+\frac{\partial}{\partial x_j}\left[\left(\nu+\frac{\nu_T}{\sigma_\kappa}\right)\frac{\partial \kappa}{\partial x_j}\right] \qquad (1)$$

And the modeled ω-equation is:

$$\frac{\partial \omega}{\partial t}+\langle U_j\rangle\frac{\partial \omega}{\partial x_j}=\alpha\frac{\omega}{\kappa}\nu_T\left[\left(\frac{\partial\langle U_j\rangle}{\partial x_J}+\frac{\partial\langle U_j\rangle}{\partial x_i}\right)\frac{\partial\langle U_i\rangle}{\partial x_J}\right]-\beta^*\omega^2+\frac{\partial}{\partial x_j}\left[\left(\nu+\frac{\nu_T}{\sigma_\omega}\right)\frac{\partial \omega}{\partial x_j}\right] \qquad (2)$$

where the turbulent viscosity is calculated from $\nu_T=\frac{\kappa}{\omega}$.

An advantage of this model compared to the more commonly used k-ε model is that it can predict the viscous sub layer near the wall more reliable and thereby eliminating the need to use wall functions except for computational efficiency. However, the low Reynolds k-ω model requires a very fine mesh close to the wall with the first grid below $y^+=5$ (Andersson et al. 2009)

2.4. Modelling Assumptions

Two-dimensional flow: The computational time was reduced by taking a 2-D vertical and longitudinal cross-section in the central of the oven as the calculation domain. Taking only the thickness of the brick into consideration and ignoring the sand that normally covers it, further simplified the geometry. When compared with the cold bi-dimensional physical model no breads were taken into account but when a more realistic flow field and heat transfer were estimated, eight breads were included.

Turbulent flow: The irregular and natural convection of the air flow and complex geometry meant that turbulent flow was considered; in spite of the low Re number (Nussbaumer, 2003)

Steady state: The system was assumed to be in a steady state so that the cold flow physical model could be compared with the mathematical model.

2.5. Boundary Conditions

At the combustion chamber: The combustion chamber was only included in the modelling to provide a realistic flow pattern into the bakery oven. Therefore, the heat released during combustion was modelled as a heat source in the combustion chamber to increase the temperature of the incoming atmospheric air (298 K). The temperature had a range of 770-850 K measured experimentally on the fire-grate leading into the baking chamber using an air flow which corresponds to an amount of excess air between 2 and 4 (Nussbaumer, 2003). Heat losses were modelled as heat transfer through the brick wall by conduction. The volume of air was based on the theoretical amount needed for combustion and the different excess air ratios. The velocities were calculated from the air required to burn the amount of wood measured (18 kg wood /h) and are illustrated in Table 2.
The experimental temperature measurement was performance in one selected point (around 0.1m of start point) in the fire-grate as described by (Manhica et al, 2012).

At the wall surfaces: Convective and radiative heat transfer boundary conditions were applied to all outer walls and to the feeding door. Heat flux to the wall was computed as

$$q=h_f\left(T_w-T_f\right)+q_{rad} \qquad (3)$$

h_f is the local heat transfer coefficient of the fluid (W/m^{2}K), T_w is the wall surface temperature (K), T_f is the local fluid temperature (K) and, q_{rad} is the radiative heat flux (W/m^2). Air, which consists of nitrogen, oxygen, small amount of carbon dioxide and other gases, has been found not showing absorption band in those wavelength regions of importance to radiant heat transfer. The walls are insulated with red bricks (0.22 m). The heat loss, which was calculated from the overall heat transfer coefficient that combines convection and conduction, was estimated as being approx. 13.0 W/m^{2}K (Birds et al, 2001)

2.6. Modelling Equation

Assuming a two-dimensional turbulent and steady state airflow within the oven, the heat transfer mechanisms are described mathematically by the equations of mass conservation (continuity equation), momentum conservation (Navier–Stokes equation) and energy conservation.

Mass conservation (continuity equation):

$$\frac{\partial \rho}{\partial t} = -\left(\frac{\partial}{\partial x}\rho \upsilon_x + \frac{\partial}{\partial y}\rho \upsilon_y\right) \tag{4}$$

Momentum conservation (Navier–Stokes equation):

$$\frac{\partial}{\partial t}\rho \upsilon_x = -\left(\frac{\partial}{\partial x}\rho \upsilon_x \upsilon_x + \frac{\partial}{\partial x}\rho \upsilon_y \upsilon_x\right) - \left(\frac{\partial}{\partial x}\tau_{xx} + \frac{\partial}{\partial x}\tau_{yx}\right) - \frac{\partial P}{\partial x} + \rho g_x \tag{5}$$

$$\frac{\partial}{\partial t}\rho \upsilon_y = -\left(\frac{\partial}{\partial x}\rho \upsilon_x \upsilon_y + \frac{\partial}{\partial x}\rho \upsilon_y \upsilon_y\right) - \left(\frac{\partial}{\partial x}\tau_{xy} + \frac{\partial}{\partial x}\tau_{yy}\right) - \frac{\partial P}{\partial y} + \rho g_y \tag{6}$$

Energy conservation:

$$\rho c_p\left(\frac{\partial T}{\partial t} + \upsilon_x \frac{\partial T}{\partial x} + \upsilon_y \frac{\partial T}{\partial y}\right) = \kappa\left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right] + 2\mu\left[\left(\frac{\partial \upsilon_x}{\partial x}\right)^2 + \left(\frac{\partial \upsilon_y}{\partial y}\right)^2\right]$$
$$+\mu\left[\left(\frac{\partial \upsilon_x}{\partial y} + \frac{\partial \upsilon_y}{\partial x}\right)^2\right] - \frac{2\mu}{3}\left[\frac{\partial \upsilon_x}{\partial x} + \frac{\partial \upsilon_y}{\partial y}\right] \tag{7}$$

Together with the initial boundary conditions (velocity, and temperature), the equations were solved with Fluent on ANSYS 12 Workbench. It was found that 1500 iterations were satisfactory to achieve low residuals for all equations. At the end of the iterations, the residuals were reduced to less than 10^{-3} of their initial values. Simulations were carried out to study the effects on the airflow patterns when the velocity of the air entering the combustion chamber in the wood-fired bakery ovens was varied according Table 2.

Table 2: Velocities used in the different experiments and the corresponding Re number.

Density = 0.23 kg/m3, viscosity = 0.46·10-4 kg/mK, characteristic length of the inlet section of the combustion chamber = 0.54 m.

Parameter /Unit	Simulation #1	Simulation #2	Simulation #3	Simulation #4	Simulation #5	Simulation #6
Velocity, v_i [m/s]	0.30	0.40	0.45	0.59	0.63	0.74
Re number	81·10	$11\cdot10^2$	$12\cdot10^2$	$16\cdot10^2$	$17\cdot10^2$	$20\cdot10^2$

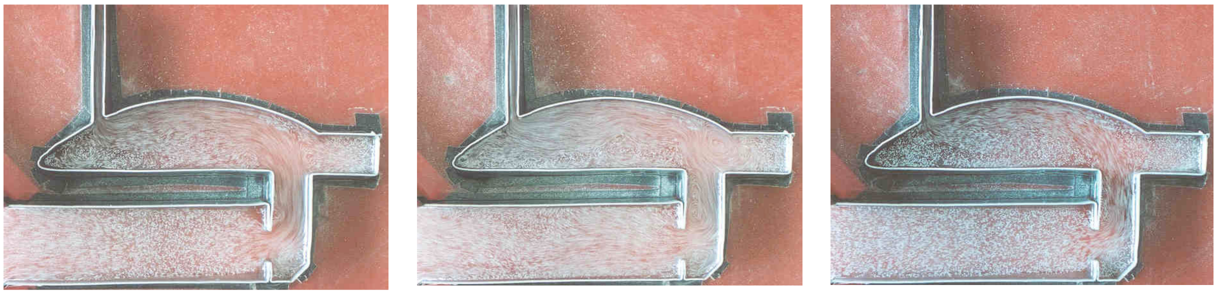
3. Results and Discussion

3.1. Bi-dimensional Physical Model

3.1.1. Flow Pattern

The results of the 2D cold flow model showed the necessity of having the correct combination of exposure times with respect to the velocity of the fluid. Short exposure times are best for understanding high velocity flows and, conversely, longer exposure times for low velocity flows. The different stream lines of the experimental exposure times taken at a constant velocity of water are shown in Figure 2.

In Figure 2a) the stream lines have an exposure time of 1/2 s, presenting a long and continuous trace of the aluminium particles that shows the large flow pattern clearly. Figure 2b) represents an exposure time of 1/4 s, showing clear stream lines that also capture swirls that are quite small; the same behaviour was also captured at 1/8 s. The short exposure time of 1/15 s seen in Figure 2c) allows only stream lines in the high velocity areas to be seen; the other parts show little, or no, movement which is indicated by the individual particles of aluminium. This very short exposure time can then be used to determine the local increase in number of particles more than a flow pattern.



a) b) c)

Figure 2**:** 2D cold flow model stream lines with exposure times of a) 1/2 s, b) 1/4 s and c) 1/15 s.

3.2. Mathematical Modelling

3.2.1. Airflow Velocity

Airflow and velocities affect the flow field in the oven by creating radial and axial pressure gradients. These gradients are the result of the nozzle design in the end of the combustion chamber which increases the velocity in the baking chamber and thereby creating a low static pressure. In case of a strong swirl, the adverse axial pressure gradient is sufficiently large to create an internal recirculation zone along the furnace axis and simultaneously reduce the outer recirculation zone near the wall. The temperature difference created by wood combustion is another source of gas circulation. The hot gases pass through the nozzle and expand in the bakery oven where heat is transferred to the bread. Due to the natural draught in a WFBO the baking process is relative uncontrolled and non-uniform, which affects the quality of the final product. An increase in the velocity of the airflow results in a higher weight loss, lower softness and darker surface. Therefore either the baking time or temperature should be reduced to compensate for the increased heat transfer rate [19]. Carvalho and Nogueira, (1997), showed how the airflow velocity in an oven chamber influenced the heat flux to the bread being baked, as well as how the heat flux could be optimized to obtain a better distribution of heat. There is a lack of data available on the aerodynamics of flow fields, especially in the environment of a wood-fired bakery oven that utilizes natural air draughts in the combustion chamber. Experimental work, in many cases, lacks accuracy due to difficulties in making measurements. Thus the mathematical model was used to qualify and quantify the impact in the bakery oven from variations in the inlet velocities of air to the combustion chamber. Figure 3 shows the different velocities profiles at the fire-grate caused by variation of the inlet air velocity to the combustion chamber, including a case with the velocity 0.59 m/s with an opened feeding door. The x-axis represents the location and the dimension of the fire-grate in the wood-fired bakery oven.

The combination of mathematic modelling and cold bi-dimensional give an interpretation of the flow inside of the baking oven. Figure 4 shows the flow pattern of both the bi-dimensional cold flow model and the mathematical modelling. The upper row in the figure shows the differences in the flow of the inlet velocity between 0.015 m/s and 0.0375 m/s. The lower row shows the contours of the streamlines at inlet air velocities between 0.3 m/s and 0.74 m/s, which correspond to the same Re number as in the cold flow experiments.

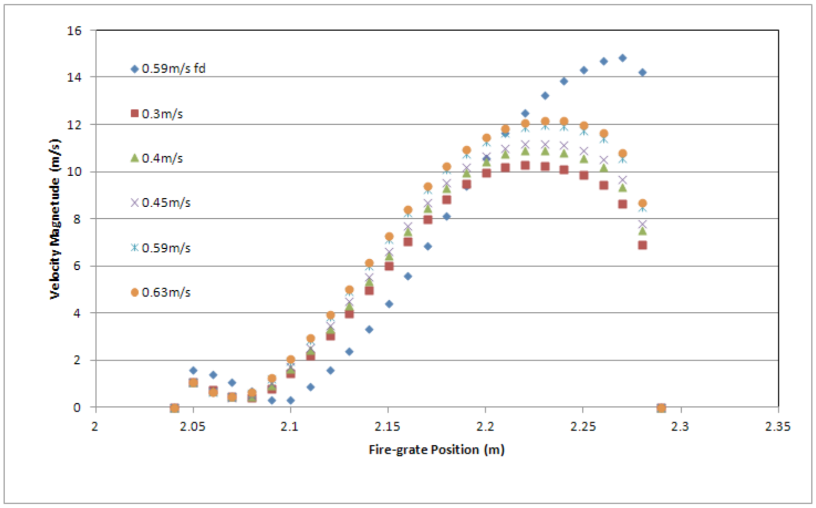
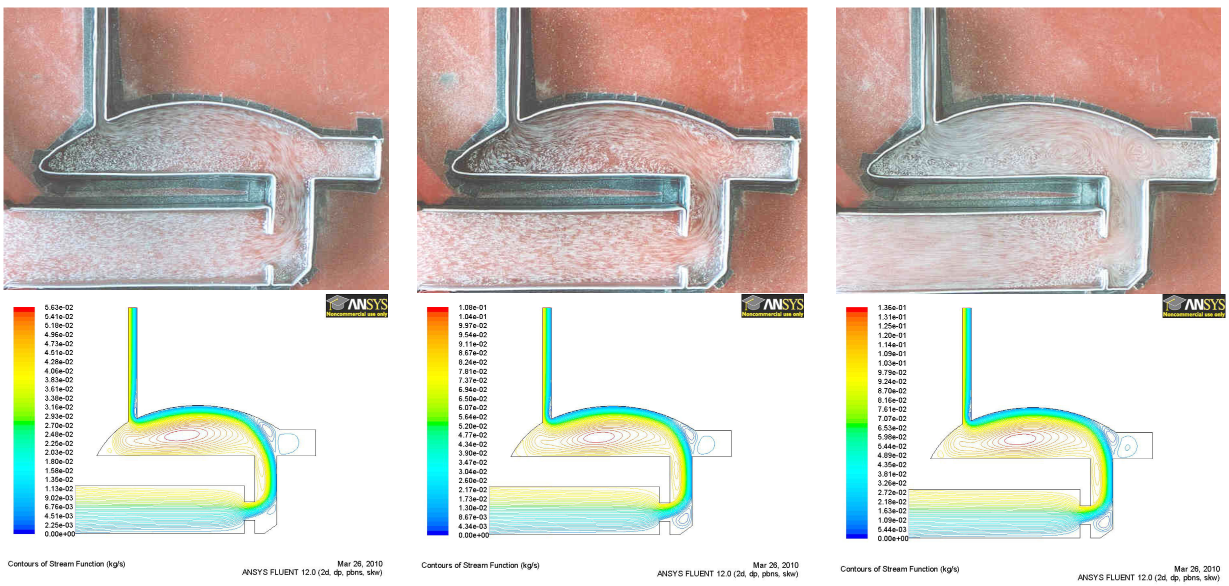


Figure 3: Velocity profile at the fire-grate for different inlet air velocity to the combustion chamber



a) b) c)

Figure 4**:** Comparisons of the streamlines of cold physical model with Mathematical model ; Upper row with 2D Cold Physical Model: a) Water velocity = 0.015 m/s; b) Water velocity = 0.030 m/s; c) Water velocity = 0.0375 m/s; Lower row with Mathematical Model: a) Air velocity 0.3 m/s; b) Air velocity 0.59 m/s; c) Air velocity 0.74 m/s

This flow pattern can be used to predict the conditions that will occur in the oven. In this context, the analysis is based on the relationship between the vortices created by the flow and the heat exchange. The dynamic vortex that appears inside the baking oven varies with the velocity of the inlet air and determines the convective heat exchange between the dough and the hot air as a correlation of the Nusselt number and the absolute vortices flux (Torll & Yanagihara, 1989; Momayez et al, 2004).

Radiation is the most important heat transfer mode for the baking process and the importance increase when there is a low flow rate and only natural convection has to be considered (Carvalho & Nogueira, 1997). The flow pattern in a WFBO show that there is a large part of the gases that leaves directly via chimneys without release all its heat content to the bread. This shows clearly that improvements can be made to enhance the performance of the baking oven via changes on the internal design or the location of the chimneys to achieve high resident time of the hot gases.

The cold model shows that there is a small recirculation of gases inside the oven even at higher velocity, which can be seen in Figures 4. Quantified results can be seen when wood is consumed at a rate of 18 kg/h, the lowest velocity of the inlet air (valid for λ =2) in the cold flow model, i.e. 0.015 m/s, corresponds to 0.30 m/s in the mathematical model. It shows that the flow pattern is characterized by a uniform flow: from the fire-grate inlet to the baking oven and then to the chimney, with little formation of vortices and this means low natural convection in the oven. An increase in the velocity (λ=4) corresponding to 0.59 m/s in the mathematical model implies that the gases can be circulated to a higher degree, measured in terms of velocity magnitude, without changing the flow pattern inside the baking oven, as shown in Figure 4. At the lowest velocity magnitude (0.3 m/s), the maximum quantity of hot gases that circulate in the bakery oven is 0.0056 kg/s; this increases with increasing velocity magnitude to a value of 0.011 kg/s at 0.59 m/s and 0.136 kg/s at 0.74 m/s. This phenomenon is one of the causes of variation in quality of the produced bread.

3.2.2. Temperature Profile

The combustion of wood, the design of the furnace, the composition of the fuel, the way in which the fire is tendered and the rate at which fuel is fed into it are the most important parameters determining the thermodynamic efficiency of a WFBO. Other parameters include the design of the baking chamber, the baking practice employed and meteorological conditions. Poor air inlet control contributes to inefficient combustion; resulting negative effects arise when the excess air is beyond the typical range ($2 \leq \lambda \leq 4$) (Lucas & Blasiak, 2000; Nussbaumer, 2003). Incomplete combustion occurs below this range and if the flow of inlet air is too high, the temperature will be too low. However, even within the range of excess air, the temperature distribution and turbulent kinetic energy inside the oven are influenced strongly by the velocity of the inlet air. At the inlet velocities (0.3 m/s), the temperature is high at the fire-grate inlet of the baking oven and the circulation of the hot gases is weak inside the oven compared with the highs velocity than that, which enhances the heat exchange with the processed bread. In the range of complete combustion, atmospheric conditions may vary and thereby change the velocity of the inlet air entering the combustion chamber.

The temperature is the dominating factor regarding the quality of the product during baking, since it affects the enzymatic reaction, volume expansion, gelatinization, browning reaction and water migration (Patel et al, 2005). The common industrial practice for achieving optimal results is to bake bread at a constant temperature (Therdthai & Zhou, 2003). An uneven distribution of temperature and random disturbances in the oven often result in the dough being subjected to inconsistent heat treatment (Wong et al, 2007). Figure 5 shows the correlation between inlet velocity and measured and calculated temperatures. The x-axis represents the location and the dimension of the fire-grate in the wood-fired bakery oven. It is evident that there is a range (until 0.3) where an increasing inlet velocity increases the temperature in oven. After that the heat released by the combustion will first heat up the excess air before entering the baking oven. As a consequence recirculation of gases due to turbulent kinetic energy and the degree of heat exchange between the gases and the products, determined as the effective thermal conductivity of the gases, will decrease. At higher velocities the temperature will continually decrease until the critical velocity 0.74 m/s, which correspond to λ>4, is reached where recirculation of gases in the oven is strong but at low temperature. This implies a reduction of the effective thermal conductivity between the gases to the products. It is thus clear that the inlet air must be controlled within this range ($2 < \lambda < 4$) to reach a high efficiency and also to have a good quality of the products.

An alternative way of determining the combined influence of baking temperature and other parameters is measurement of the heat flux (Therdthai & Zhou, 2003; Fahloul et al, 1995), which is defined as the heat transfer rate per unit area required by the product. Measuring heat flux has been claimed to be a more useful method than measuring air temperature for controlling the quality of bakery products (Therdthai & Zhou, 2003; Thorvaldsson & Skjoldebrand, 1998), and has the advantage that it can be measured continuously. It takes both air temperature and velocity into account, which has a strong correlation with the properties of the produced bread such as moisture content, weight and colour. It should be noted that the main flow pattern inside a WFBO is almost the same, regardless of the velocity of inlet air within this range. This means that the variation in the velocity of the inlet air mainly affects the amount of heat transferred to the dough by convection and the amount of hot gases that circulate inside the baking oven, and is directly related to the quality of the bread produced.

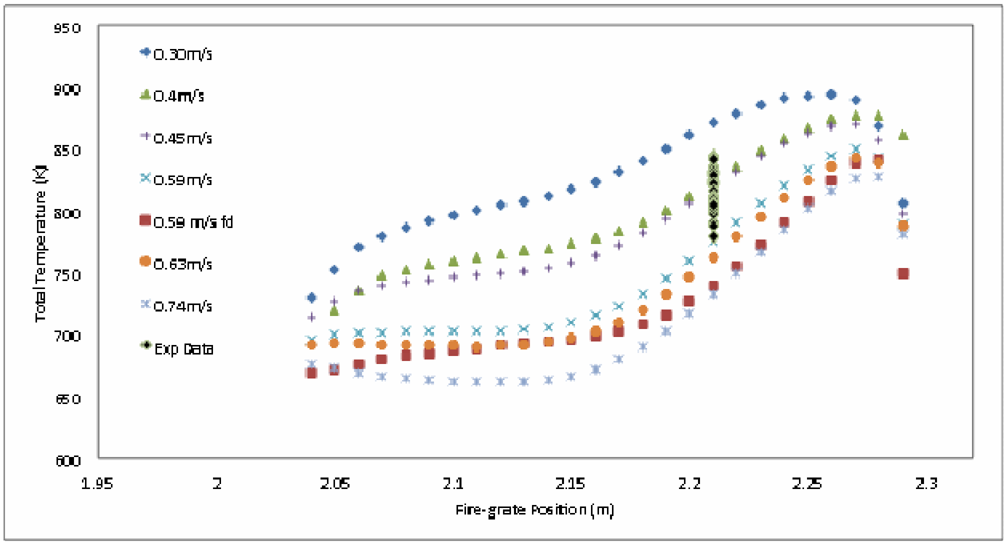


Figure 5: Measured and calculated temperature profile at the fire-grate experimental measured at different velocities of the incoming air to the combustion chamber.

The temperature distribution is not homogeneous in the baking area of the wood-fired bakery oven as consequence of the design. To clarify this phenomenon, the flow pattern in the baking oven was determined by CFD calculation with the bread included. Figure 6a) and b) shows the temperature distribution on the surface of each bread during the baking process. The x-axis in figure 6b) represents available baking space in the oven. The breads located in oven surface between the fire-grate and end in the position mellow the chimney in the baking oven has the highest temperature mainly due to radiation from the top roof heated by the main flow of gases that cross the oven directly to the chimney and a small contribution from convection of the weak vortices formed by the recirculation. Therefore, dough placed in this region will be baked faster and at different conditions compared with others in same batch. The range of temperature measured was 750-820 K at the inlet fire-grate (Manhica et al, 2012). According to experimental results, the desired temperature in a bread baking process is in the range of 500 to 520 K (Therdthai & Zhou, 2003; Hadiyanto et al, 2007; Mondal & Datta, 2008; Zanoni et al, 1993; Therdthai et al, 2002). This means that the remaining temperature represent losses in the oven, which decrease the efficiency of the WFBO.

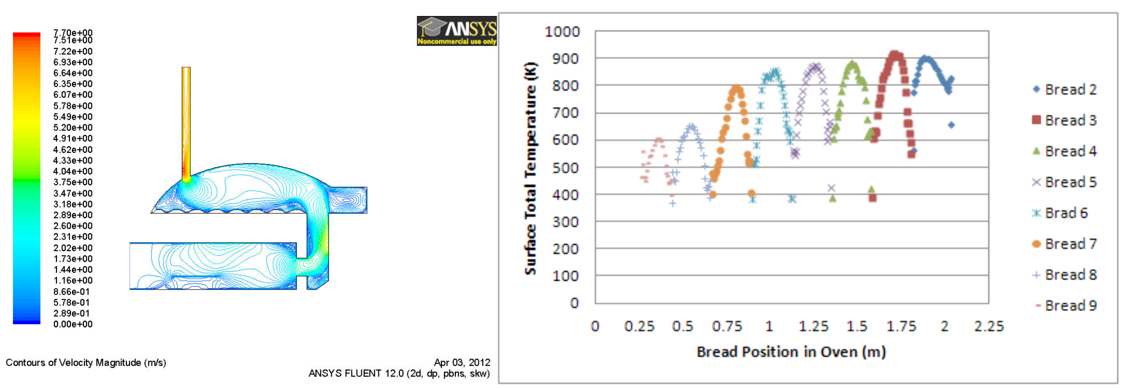
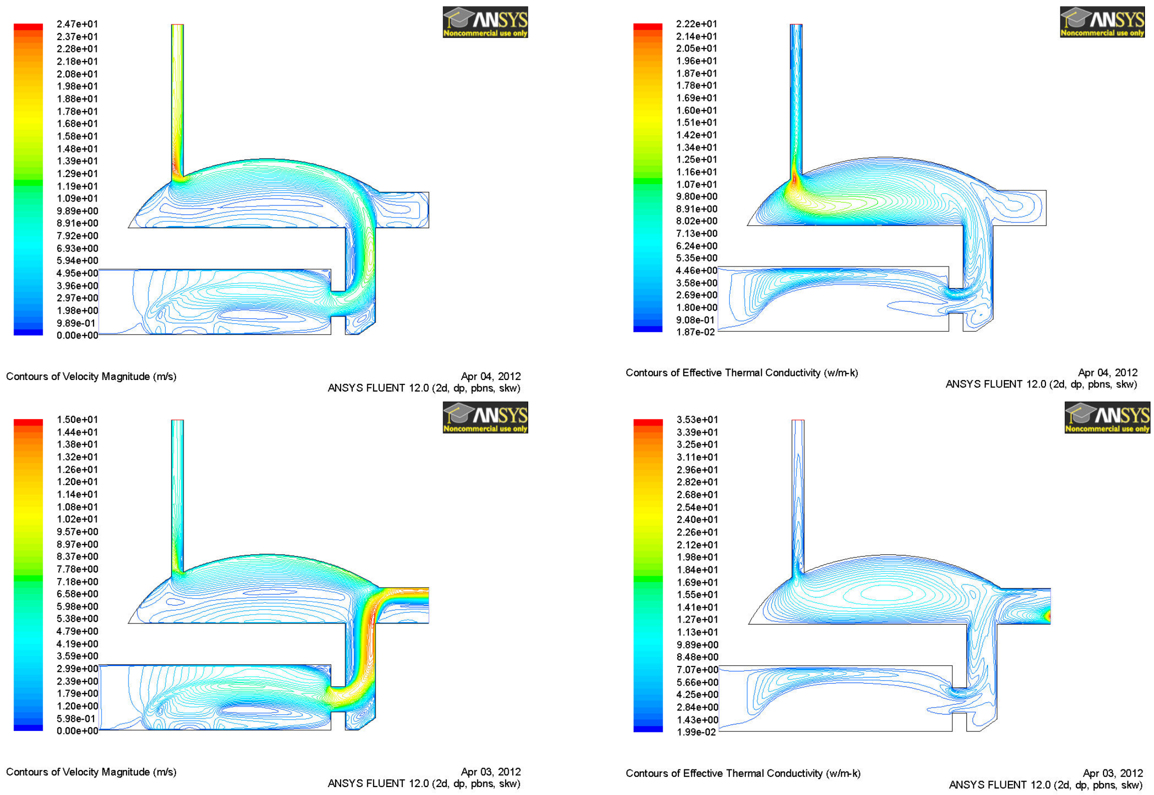


Figure 6: a) Velocity profile in baking oven; b) computed bread surface temperature in oven

3.2.3. Influence of the Feeding Door

The ending section of the combustion chamber is a funnel-shaped nozzle and leads the gases to the duct that connects the combustion and oven chambers via the furnace fire-grate. As a result of this construction, the gases flow at high speed and the static pressure within the bakery oven is slightly lower than that of the surroundings. This difference is sufficient to draw air into the interior of the baking oven via the feeding door. Two phenomena occur as a consequence of this: some of the surrounding air is carried into the oven whilst some of the hot gases escape from the oven due to their high velocity. The feeding process is manually.

This means that a large period of the time, the feeding door is open which affects the efficiency negatively. Figure 7 show the calculated the flow pattern of such phenomena and compare the flow patterns between an open and a closed feeding door at the same inlet velocity (0.59 m/s) with respect to a) velocity, b) effective thermal conductivity. The relative magnitude of these parameters decreases inside the baking oven when the feeding door is opened but the relative magnitude of the velocity at the fire-grate is high. Some WFBOs, therefore, have a collector of hot gases (i.e. a chimney) placed directly above the feeding door to prevent hot gases from entering into the work place. This provides better working conditions for the bakery operator both when placing dough in the oven and removing the bread but at the same time introduces another source of losses in the process.



a) b)

Figure 7: Flow pattern in a WFBO when the feeding door is closed (upper row) and opened (lower row). a) velocity, b) Effective Thermal Conductivity

4. Conclusions

A two-dimensional CFD model was established and it could describe the flow pattern in a wood-fired bakery oven in good agreement with experimental results obtained by 2D physical model and also predict temperatures in the oven which was experimentally verified in the field.

The flow pattern in both models shown that large amount of hot gases passes the baking oven to the chimney without releasing all its heat content. This is a result of the design of the baking chamber and probably of the location of the chimneys. The feeding process is also affecting the flow pattern and this was confirmed with a mathematical model. It further showed that the absence of the control of the inlet air reduce the efficiency in such a way that less amount of the released heat during combustion was used for the baking process.

The way in which variations in velocity affect the temperature provided information of how to achieve an optimum baking process. A temperature increase in the oven can be achieved by increase the air intake until lambda is equal to 2 (which correspond to a velocity into the baking chamber of 0.3 m/s for the specific oven investigated). In that condition both the temperature inside of the baking oven increase and the baking conditions are improved by increasing the effective thermal conductivity of the hot gases. Too high velocities (equal to lambda larger than 4) gives too low temperatures in the oven due to low gas temperature and high recirculation.

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