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DEVELOPMENT AND EVALUATION OF A WOOD FURNACE FOR GRAIN DRYING

Dissertation submitted to the Agricultural Engineering Graduate Program of the Universidade Federal de Viçosa in partial fulfillment of the requirements for the degree of *Magister Scientiae*.

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VIÇOSA - MINAS GERAIS
2022

Ficha catalográfica elaborada pela Biblioteca Central da Universidade Federal de Viçosa - Campus Viçosa

T

Y95d
2022

Yusuf, Kamaldeen Ayinla, 1983-
Development and evaluation of a wood furnace for grain drying / Kamaldeen Ayinla Yusuf. – Vicosa, MG, 2022.
1 dissertação eletrônica (68 f.): il. (algumas color.).

Texto em inglês.
Orientador: Evandro de Castro Melo.
Dissertação (mestrado) - Universidade Federal de Viçosa, Departamento de Engenharia Agrícola, 2022.
Referências bibliográficas: f. 65-68.
DOI: https://doi.org/10.47328/ufvbbt.2022.686
Modo de acesso: World Wide Web.

1. Fogões à lenha. 2. Fornos - Combustão - Métodos estatísticos. 3. Grãos - Secagem. I. Melo, Evandro de Castro, 1959-. II. Universidade Federal de Viçosa. Departamento de Engenharia Agrícola. Programa de Pós-Graduação em Engenharia Agrícola. III. Título.

CDD 22. ed. 697.07

Bibliotecário(a) responsável: Euzébio Luiz Pinto CRB-6/3317

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APPROVED: October 03, 2022.

Assent:

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To God almighty and my family

I dedicate.

ACKNOWLEDGEMENTS

To God almighty, for the gift of life, guidance, protection and for illuminating my path.

To my parent, Alh. Ibrahim Magaji and Alh. Aishat Ibrahim, for all the love, prayers, education, support, trust and for always believing in me.

To the Federal University of Viçosa (UFV) and the Department of Agricultural Engineering (DEA) for the opportunity and enabling environment to study and support.

To Professor Evandro de Castro Melo, for his dual responsibility as advisor and father in Brazil, for the guidance, trust, teachings and for the opportunities granted.

To Professor Luís César da Silva, for his teachings, attention, encouragement, understanding, guidance and for always taking time out of his busy scheduled to ensure that the work is perfect.

To Juarez Sousa e Silva, for his assistance, encouragement and for providing enabling environment during the execution of the experiment.

To the Forum for Agricultural Research in Africa (FARA), for the opportunity given to be one of the beneficiary of Agricultural Research and Innovation Fellowship for Africa (ARIFA) fellowship.

To Tertiary Education Trust Fund (TetFUND), for the financial sponsorship of the program.

To Auchi Polytechnic, Auchi, for the for granting my study leave which enables me to explore this opportunity.

To my family, for their love, support, understanding, prayer and patience and for always believing in me.

To Ayinke, for her support, love, friendship, understanding and prayer, in fact, you have really proved to be a blessing to me.

To Caroline, Marcus and everyone who, in some way or the other, assisted with the realization of this work.

To the Federal University of Viçosa, for the opportunity to complete the postgraduate course.

This study was financed in part by the Coordenação de Aperfeiçoamento de Pessoal de Nível Superior – Brasil (CAPES) – Finance Code 001.

Thank you all.

“Success is a product of determination plus hard work."

ABSTRACT

KAMALDEEN, Ayinla Yusuf, M.Sc., Universidade Federal de Viçosa, October, 2022. **Development and Evaluation of a Wood Furnace for Grain Drying**. Adviser: Evandro de Castro Melo. Co-advisers: Luís Cesar da Silva and Juarez Sousa da Silva.

The use of wood furnaces for thermal energy application in grain dryers is common in Brazil, however, contaminations of the products and environmental effects due to incomplete combustion and its excessive use in drying are major challenges. In this study, a wood furnace was designed, build and its performance was evaluated for mixed and indirect heating of the drying air. The design of the combustion chamber volume, combustion cell, grate area, and heat exchanger were based on the following demands: thermal energy from 15 to 30 kW (12,898 to 25,795 kcal. h^{-1}), drying air flow rate from 35 to 45 m^3 min^{-1} and drying air temperature from 50 to 80 °C. Considering experimental data, quadratic regression models were adjusted to describe the temperatures in furnace compartments. These regressions were used to develop a simulation model using Extend™ simulation language to predict the performance of the furnace considering the desired drying air temperature. To determine the efficiency, furnace operation was simulated considering four drying air temperatures 50, 60, 70, and 80 °C, and six combinations of operation times in mixed and indirect heating during 0 and 119 min, 5 and 114 min, 10 and 109 min, 15 and 104 min, 20 and 99 min, and 30 and 89 min, respectively. The simulation result shows that for mixed and indirect heating of 5 and 114 min, respectively, the best outcomes for drying air temperatures of 50, 60, and 70 °C were the efficiency of 32, 44, and 55%, during 99.2, 98.4, and 95.5% of operation time. However, the best outcome was the efficiency of 64% during 93.5% of operating time using the drying air temperature of 80 °C and mixed and indirect heating during 10 and 109 min, respectively.

Keywords: Eucalyptus. Combustion. Indirect and mixed heating. Modeling and simulation. Thermal efficiency.

RESUMO

KAMALDEEN, Ayinla Yusuf, M.Sc, Universidade Federal de Viçosa, outubro de 2022. **Desenvolvimento e Avaliação de um Forno de Madeira para Secagem de Grãos**. Orientador: Evandro de Castro Melo. Coorientadores: Luís Cesar da Silva e Juarez Sousa da Silva.

A utilização de fornalhas a lenha para aplicação de energia térmica em secadores de grãos é comum no Brasil, porém, contaminações dos produtos e efeitos ambientais devido à combustão incompleta e seu uso excessivo na secagem são grandes desafios. Neste estudo, uma fornalha a lenha que opera em fogo indireto e direto foi projetada, construída e avaliada. O projeto do volume da câmara de combustão, célula de combustão, área da grelha e trocador de calor foram baseados nas seguintes demandas: energia térmica de 15 a 30 kW (12.898 a 25.795 kcal. h^{-1}), vazão de ar de secagem de 35 a 45 m^3 min^{-1} e temperatura do ar de secagem de 50 a 80 °C. Considerando os dados experimentais, modelos de regressão quadrática foram ajustados para descrever as temperaturas nos compartimentos da fornalha. Essas regressões foram empregadas na implementação de um modelo de simulação, usando a linguagem de simulação Extend™, para prever a eficiência térmica da fornalha para uma dada temperatura do ar de secagem. Para determinar a eficiência, a operação da fornalha foi simulada considerando quatro temperaturas do ar de secagem 50, 60, 70 e 80 °C, e seis combinações de tempos de operação com aquecimento misto e indireto do ar conforme os intervalos de 0 e 119 min, 5 e 114 min, 10 e 109 min, 15 e 104 min, 20 e 99 min, e 30 e 89 min, respectivamente. Resultados da simulação demonstraram que para aquecimento misto e indireto de 5 e 114 min, respectivamente, os melhores resultados para temperaturas do ar de secagem de 50, 60 e 70 °C foram a eficiência de 32, 44 e 55%, durante 99,2; 98,4 e 95,5% do tempo de operação. No entanto, o melhor resultado foi a eficiência de 64% durante 93,5% do tempo de operação ao de utilizar a temperatura do ar de secagem igual a 80°C para o aquecimento misto e indireto durante 10 e 109 min, respectivamente.

Palavras-chave: Eucalipto. Combustão. Aquecimento misto e indireto. Modelagem e simulação. Eficiência térmica.

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LIST OF ABBREVIATIONS

C	-	Carbon.
CNP	-	National Petroleum Council
CO	-	Carbon monoxide.
CO_2	-	Carbon dioxide.
FLR	-	Fuel loading rate.
GTC	-	Global temperature coefficient.
H_2	-	Hydrogen.
HHV	-	Higher Heating Value.
H_2O	-	Water.
LHV	-	Lower Heating Value.
N_2	-	Nitrogen.
NO_x	-	Nitrogen oxide.
O_2	-	Oxygen.
S	-	Sulfur.
SO_2	-	Sulfur oxide.
VTT	-	Volumetric thermal tension.

LIST OF SYMBOLS

$m_{O_2}^{ts}$	-	Theoretical mass of oxygen, kg.kg $^{-1}$ of dry fuel.
$m_{O_2}^{tu}$	-	Theoretical mass of oxygen, kg.kg $^{-1}$ of wet fuel.
$m_{dry\ air}^{ts}$	-	Theoretical mass of dry oxidizing air, kg.kg $^{-1}$ of dry fuel.
$m_{dry\ air}^{tu}$	-	Theoretical mass of dry oxidizing air, kg.kg $^{-1}$ of dry fuel
$m_{moist\ air}^{ru}$	-	Actual mass of moist oxidizing air, kg.kg $^{-1}$ of wet fuel
W	-	Moisture ratio of moist air, kg of steam.kg $^{-1}$ of dry air.
λ	-	Excess air coefficient
$m_{dried/moist\ air}^{r}$	-	Actual mass flow of dry or moist oxidizing air, kg.s^{-1}
$m_{dried/moist\ air}^{t}$	-	Theoretical mass flow rate of dry or moist oxidizing air, kg.s^{-1}
$m_{dry\ gases}^{ts}$	-	Theoretical mass of dry gases, kg.kg^{-1} of dry fuel.
$m_{dry\ gases}^{rs}$	-	Real mass of dry gases, kg.kg $^{-1}$ of dry fuel.
$m_{dry\ gases}^{ru}$	-	actual mass of dry gases, kg.kg $^{-1}$ of wet fuel
U_{bu}	-	Fuel moisture content % wb
qu	-	Thermal energy required for heating the air, kW;
η	-	Estimated Furnace Efficiency, %
U	-	Global temperature coefficient ($W/m^2.°C$)

SUMMARY

1. INTRODUCTION

Due to fossil fuel scarcity, high price, and high potential negative environmental impact, biomass fuels have presented as an attractive alternative for agricultural processing activities, such as grain drying (KUMAR et al., 2022). Biomass fuels are natural and renewable resource, that, in many countries, prices are competitive with fossil fuels (BALAT et al., 2009; BILDIRICI et al., 2013).

The combustion of hydrocarbons present in fossil fuels and biomass leads to the production of carbon dioxide (CO_2), carbon monoxide (CO), and nitrogen oxides (NOx). These substances may be associated with negative environmental impacts, such as increased greenhouse gas emissions, acid rain, and photochemical smoke generation if complete combustion is not achieved (BILGILI et al., 2016).

Biomass is important renewable source energy around the world, which can be categorized as firewood, wood processing, wood chips, charcoal, rice husk, sugarcane bagasse and manures (USMANI et al., 2021).

For drying products such as corn, soybeans, wheat, and coffee, firewood is frequently used as fuel in Brazil. In this case, there are risks of product quality depreciation and negative environmental impacts that occur as a result of incomplete combustion of the fuel. To reduce this risk requires well-designed furnace and the adoption of process control methodologies for stabilizing the drying air temperature and minimizing CO emissions and smoke. Another important issue is the quality of the firewood, which must have a lower caloric value above 10,500 kJ. kg^{-1}, and moisture content below 30% (DA SILVA et al., 2019).

A furnace is equipment designed to promote the conversion of chemical energy from fuels into thermal energy through the combustion reaction. Due to the nature of fuels, solids, liquids, gases, and powders, it is necessary to use the appropriated furnace to achieve a complete combustion.

In grain dryers, when using firewood, furnace can be classified as direct or indirect heating. Direct-heating furnaces characterize by mixing the gases resulting from combustion with the ambient air and blown by a fan directly into the grain mass. While at indirect-fired furnaces, drying air is heated by means of a heat exchanger.

Most coffee producers in Brazil use a firewood furnace for drying. Unfortunately, a lot of this material is wasted, due to the inefficient operation and poor control which leads to excessive smoke emission, affecting the farm and nearby cities' environment, traffic on roads, and workers' health. These problems have held the Public Ministry and environmental inspection authorities to establish rules and penalties that in some cases leads to sealing of grain processing facilities and producer farms.

Therefore, the objective of this work was to develop and evaluate an indirect and direct heating furnace for heating drying air in grain dryers, with low levels of smoke emission, labor hour use, and firewood loading frequency.

2. LITERATURE REVIEW

2.1. Biomass Fuel

Fuels are substances rich in carbon and hydrogen which chemically react in the presence of oxygen to release CO_2, water vapor, and energy in the form of heat and light. Heating value is the main characteristic of fuels and refers to the amount of energy released during the complete combustion of a unit of mass of fuel. For solid fuels, it is usually expressed in $kJ.kg^{-1}$, and for gaseous fuels, in $kJ.m^{-3}$ (DJUROVIC´ et al., 2012).

When the latent heat of condensation of the moisture present in the fuel is considered in the determination of the heating value, it is called higher heating value (HHV); but when it is not considered, the lower heating value (LHV) of the fuel is obtained, which is the most important. Higher calorific power is measured experimentally with calorimeters; however, in the absence of this, it can be estimated from the elemental composition of the fuel, on a dry basis using Dulong formula (Equation 1): (THIAGO DE PAULA, 2019)

$$HHV = 33774C + 141744\left(H_2 - \frac{O_2}{8}\right) + 9238S \qquad (1)$$

Where;

HHV	=	higher heating value, $kJ.kg^{-1}$ of dry fuel;
C	=	carbon fraction in the fuel, kg of $carbon.kg^{-1}$ of dry fuel;
H_2	=	hydrogen fraction of the fuel, kg of $hydrogen.kg^{-1}$ of dry fuel;
O_2	=	oxygen fraction in the fuel, kg of $oxygen.kg^{-1}$ of dry fuel; and
S	=	sulfur fraction in the fuel, kg of $sulfur.kg^{-1}$ of dry fuel;

As reported in Equation 2, the LHV can be determined analytically as a function of HHV by subtracting from the HHV the enthalpy of vaporization of water due to the water vapor formed in the reaction of hydrogen with oxygen (BAZZO, 1995).

$$\mathrm{LHV} = \mathrm{LHV} - 2.440(9\mathrm{H}_2) \quad (2)$$

Where;

LHV = lower heating value, $kJ.kg^{-1}$ of dry fuel;

H = Proportion of hydrogen present in the fuel, %

2.1.1. Firewood as furnace fuel

Firewood is the most important source of renewable energy because it is considered the "CO_2 neutral" fuel, since the amount of CO_2 absorbed by plants during growth by the photosynthesis process and the amount that is released during its thermal decomposition are equal (DJUROVIĆ et al., 2012; SILINS, 2012; LAURI et al., 2014).

Firewood and charcoal are widely used biomass in Brazil with largest production from the South, followed by the Southeast. Together, these regions account for more than 86% of national production (BICHEL and TELLES, 2021).

More recent statistics showed that energy obtained from renewable sources represents 42.9% of the Brazilian energy matrix, while firewood and charcoal represented 8% of this total (IGNACIO et al., 2019).

The particle size and moisture content are important physical properties of the solid biomass materials because together, they influence the flow properties of the material in transport and storage systems. The moisture content of firewood indicates the proportion of water present in it and is one of the important parameters that affects the quantity of energy released. The procedure to determine the moisture content consists of weighing the empty container/dish and the samples. The samples are dehydrated at a temperature of 105 °C in an electric oven for 24 hours. Finally, the moisture content of the samples is calculated as the percentage of fraction of the difference between the wet and dried firewood (WU et al., 2011).

The specific apparent mass of firewood and is another important parameter in determining fuel consumption. Numerically, it represents the amount of firewood in kilograms contained in a volume of one cubic meter.

2.2. Combustion process

The combustion of a product is defined as chemical reactions during which the fuel elements are rapidly oxidized to release heat and light energy (SIMONA *et al.*, 2020). Combustion of organic fuels is an exothermic process in which the fuel and combustion air are consumed to produce combustion gases and solid substances (ash/slag) as by-products.

Combustion can be complete or incomplete depending on the volume of primary air to oxidize the fuel during the combustion process. If the fuel supplied is greater than the volume of air or the volume of air is greater than the fuel available, there will be an incomplete combustion process. Thus, a complete combustion of a fuel can only be possible in the presence of an adequate supply of oxygen. That is, if both the fuel and the air are measured and mixed in such a way that they are completely consumed (SIMONA *et al.* 2020).

The emission during burning of solid fuels has been reported as the main source of air pollution in domestic environments, responsible for death and disabilities. To reduce this, the combustion process must be monitored and adequate control of factors such as combustion air volume is necessary (DEGUCHI *et al.*, 2002). The combustion efficiency of a furnace can be affected by the time it takes for the drying air to reach the desired value, the temperature difference between the heat source and the materials being heated, and the turbulence of the fuel, air, and heat sources.

SIMONA *et al.* (2020) reported that the composition of combustion gases depends on the value of the excess air coefficient λ . If λ is less than or greater than 1, combustion is incomplete and the composition of the flue gases contains CO, CO_2, SO_2, H_2O and N_2; λ is equal to 1 for a complete or stoichiometric combustion that produces a flue gas composition that contains CO_2, SO_2, H_2O and N_2.

2.2.1. Combustion reactions

The main chemical reactions of exothermic oxidation of fuels, constituted by carbon, hydrogen and sulfur and the composition of dry air, can be approximated in 21% of O_2 and 79% of N_2. (SIMONA *et al.*, 2020). The chemical reactions are shown in Table 1.

Table 1: Exothermic Chemicals Reactions and Enthalpy in Oxidation of Fuel involving C, H and S Elements

Reactants		Products	Standard Enthalpy of Reaction (kJ.kg $^{-1}$)
$C + O_2$	→	CO_2	- 33,613
$C + ½ O_2$	→	CO	-10,004
$CO + ½ O_2$	→	CO_2	- 23,609
$2H_2 + O_2$	→	$2H_2O$ (liquid)	-143,000
$S + O_2$	→	SO_2	- 9,259

Source: (DJUROVIC´ et al., 2012)

2.2.2. Primary/oxidizing or theoretical air

The exact quantity of air necessary to provide the complete combustion of all the elemental composition of fuel is called “theoretical” or “stoichiometric” air. As it was reported, combustion process involves a rapid oxidation of fuel by air containing oxygen in the right proportion. During combustion, there is some deliberate excess of one of the two reactants (fuel and air), usually oxidizing air, which is required to complete combustion (SILINS,2012).

The chemical equations of the stoichiometric reaction between these elements and oxygen lead to the equation proposed by PERA, (1990), which determines the theoretical mass of oxygen required for combustion (Equation 3):

$$m_{O_2}^{ts} = \frac{32}{12}C + 8H_2 - O_2 + S \quad (3)$$

Where;

$m_{O_2}^{ts}$ = theoretical mass of oxygen, kg.kg $^{-1}$ of dry fuel;

and C, H_2, O_2, and S are proportion of carbon, hydrogen, oxygen and sulfur that are respectively present in the fuel.

It is common to express the consumption of oxygen and combustion air per kilogram of wet fuel because the fuel has certain moisture content, and for that purpose, the Equation 4 must be used:

$$m_{O_2}^{tu} = m_{O_2}^{ts}(1 - U_{bu}) \quad (4)$$

Where,

$m_{O_2}^{tu}$ = theoretical mass of oxygen, kg.kg $^{-1}$ of wet fuel.

Ambient air is the main oxidant used in combustion processes. Because oxygen is one of the major compositions of atmospheric air in the proportion of 23% of total air, the minimum mass of dry air necessary for combustion can be estimated by the Equation 5:

$$m_{dry\ air}^{ts} = \frac{100}{23} m_{O_2}^{ts} \tag{5}$$

Substituting Equation 3 into Equation 5, to obtained Equation 6.

$$m_{dry\ air}^{ts} = \frac{100}{23}\left(\frac{32}{12}C + 8H_2 - O_2 + S\right) \tag{6}$$

Where;

$m_{dry\ air}^{ts}$ = theoretical mass of dry oxidizing air, kg.kg $^{-1}$ of dry fuel

The stoichiometric consumption of dry oxidizing air per kilogram of wet fuel can be obtained by Equation 7:

$$m_{dry\ air}^{tu} = \frac{100}{23} m_{O_2}^{tu} \tag{7}$$

Where;

$m_{dry\ air}^{tu}$ = theoretical mass of dry oxidizing air, kg.kg $^{-1}$ of dry fuel

As there is water vapor in the composition of the oxidizing air, the theoretical consumption of moist air, per unit of wet fuel, must be calculated by Equation 8:

$$m_{moist\ air}^{tu} = m_{ar\ dry}^{tu}(1 + W) \tag{8}$$

Where;

$m_{moist\ air}^{tu}$ = theoretical mass of moist air, kg.kg $^{-1}$ of wet fuel.

W = moisture content of moist air, kg of steam.kg $^{-1}$ of dry air.

2.3. The three T's of combustion

For efficient fuel economy and consumption in combustion process it is necessary to control the presence of three elements (temperature, fuel, and oxygen) that makes up the so-called “combustion triangle”. It is necessary that the three elements are combined in the proportion and time to have a perfect and complete combustion. In addition, for efficient operation of the furnace, the 3Ts of combustion (Fuel Temperature, Running Time, and Air Turbulence) should be considered: (EDNEY, 2007).

a) **Fuel temperature**: for combustion to occur the fuel must reach the self-sustaining ignition temperature of 300 ^{0}C in the case of firewood. If the temperature is lower than this, there will be incomplete combustion which leads to production of CO and smoke. On the other hand, if the temperature is much higher than that of ignition fusion of the fuel will occur.

b) **Running time**: the fuel and generated volatile gases during combustion process must be retained in the furnace during a required time for complete combustion to occur.

c) **Air turbulence:** the furnace design should favor the turbulent movement of the air inside the furnace, so that the whole fuel can be involved, and the ideal amount of oxygen necessary for complete combustion can be provided.

2.4. Wood furnace

Furnaces are equipment designed to efficiently and continuously promote the complete burning of a given fuel to provide thermal energy at the desired temperature. For these reasons, the furnace must be designed to: allow the fuel to heat up to a self-sustaining ignition temperature; promote the homogeneous mixture of air with fuel in an optimal proportion; and allow time for complete combustion of the fuel to occur (EDNEY, 2007).

Generally, furnaces for burning solid fuels such as firewood and charcoal have the following components:

1. **Fuel tank**: the furnaces are designed with space for a certain volume on the grill in the combustion chamber to receive firewood and charcoal or other form of biomass. The size of the tank is designed and built as a function of the specific mass of the solid fuel.

2. **Combustion chamber:** it is a space destined for the combustion process to occur, where all combustible compounds must be oxidized to release thermal energy. The combustion chamber is the most important component of the fire chamber and must be designed to have adequate primary air inlet to provide enough oxygen for complete combustion of the fuel. Combustion chambers can be rectangular, circular, hexagonal, or any other shape. The most importantly, is to provide the ignition temperature necessary to initiate combustion, ensure efficient combustion of the fuel and a regular flow of gases.
3. **Grate**: this is structure that keeps solid fuel in suspension while oxidizing air circulates over its surface during the combustion process. VLASSOV (2001) mentioned some advantages of using the grate in the combustion chamber, such as: possibility of burning different solid biomasses, possibility of burning fuels with different moisture contents, efficiency and stability of the combustion process and possibility of varying the amount of burned fuel.
4. **Ash collector:** It is located below the grate. Consists of space for storage of combustion residues which are composed of ash and unburned material (coal).
5. **Primary air or oxidizing air inlets**: Are openings located around the combustion chamber for the easy passage of primary or oxidizing air into the furnace, and must be located at a strategic point to facilitate the correct oxidizer/mixture of fuel that will support complete combustion.
6. **Gas outlet**: also, called chimney. It is an opening for the exhaustion of combustion gases and the removal of smoke and other residues. Gas outlets must be located according to the furnace configuration.
7. **Drying air outlet**: It is an opening for the drying air gases from the mixing chamber to the drying cabinet.

   Based on the use of the heat obtained in the furnace, they are classified as: (i) direct heating – when the flue gas is mixed with the drying air; (ii) indirect heating – when the heat obtained is transferred through a heat exchanger, and the flue gases do not meet the drying air; and (iii) mixture – when the heated air from the direct method is allowed to mixed with the ambient air before conveying into the drying cabinet

2.4.1. Furnace with indirect heating

In furnaces with an indirect heating system, the gases from the combustion are introduced into a heat exchanger that, in contact with the air, will heat it up. In this type of furnace there are losses of thermal energy through the chimney and in the heat exchanger, resulting in a lower efficiency when compared to the direct fire furnace (EDNEY et al., 2008).

Furnaces with indirect heating are intended for agricultural products that require a controlled and moderate temperature during drying. The furnace projects with indirect heating use different heat exchangers. Some common types were designed with a tube-casing heat exchanger where the hot fluid receives energy from the gases in the combustion chamber. Here, the cold air entering the heat exchanger tubes is heated by the circulating fluid in the heat exchanger housing to a maximum temperature determined by equilibrium with the boiling temperature of the circulating fluid (EDNEY et al., 2008).

2.4.2. Furnace with direct heating

Furnaces for direct heating operate in a situation in which the gases resulting from combustion are mixed with ambient air and forced to pass through the grain mass into the drying chamber, by the aid of a fan. Contaminant compounds generated during mixing of part of the oxidizing gas with the resulting flue gases can become undesirable in cases where the combustion process is incomplete.

Direct heating furnaces produce more heat energy and are more efficient when operating with the complete combustion process. These furnaces require a decanter or tangential cyclone to retain glowing particles formed, mainly coal, separated from the gas stream by the action of centrifugal force. MELO (1987) and GOMES (1988) developed and evaluated a furnace for direct heating that uses wood as fuel, both as a function of the height of the fuel load on the grate and as a function of the excess air admitted into the system.

2.5. Furnace design

One of the methodologies used for designing wood furnaces consists of the calculations of the thermal energy required for heating air, furnace fuel consumption, the combustion chamber volume, grate area, and heat exchanger dimensions

(EDINEY, 2007).

Thus, considering the demands highlighted above, the design of a furnace that operates simultaneously in indirect and mixed-fired that minimizes the use of labor in the operation and the emission of smoke into the environment was proposed.

2.5.1. Thermal energy required for heating of air

The useful energy of a furnace consists of the energy made available for heating the air, which can be calculated by mean of Equation 9, considering the specific heat of the air constant and equal to 1.0035 kJ kg $^{-1}$ ºC $^{-1}$ (CEVIZ and KAYMAZ, 2005).

$$qu = \dot{m}_{DAir} . c_{Air} . (T_{DAir} - T_{amb}) \quad (9)$$

Where,

qu = thermal energy required for heating the air, kW;

$\dot{m}_{DAir}$ = drying mass flow rate kg s^{-1};

c_{Air} = specific heat of air Kj kg^{-1} ^{0}C^{-1}

T_{DAir} = drying air temperature ^{0}C; and

T_{amb} = ambient air temperature ^{0}C

2.5.2. Firewood consumption

By determining amount of useful energy to be supplied by the furnace per unit of time, the fuel (firewood) consumption can be determined by means of Equation 10.

$$\dot{m}_{firewood} = \frac{qu.100}{LHV.\eta} \quad (10)$$

Where;

$m_{firewood}$ = firewood consumption kg s^{-1}

LHV = lower heating value, kJ kg^{-1} and

η = estimated furnace efficiency, (Equation 11)%

$$\eta = \frac{qu}{qd}.100 \quad (11)$$

Where,

qd = total available energy (kW)

2.5.3. Volume of the combustion chamber

The volume of the combustion chamber is the most important requirements in designing furnaces, since all energy released from combustion process would be converted into gases. The volume must accommodated the gases developed from the flame and must ensure complete combustion before the hot gases got escaped.

The volume of the combustion chamber can be calculated by means of Equation 12.

$$V_{cc} = \frac{\dot{m}_{fuel}.LCV}{VTT} \quad (12)$$

Where;

V_{cc} = volume of combustion chamber, m³

$\dot{m}_{fuel}$ = firewood consumption per unit time, kg/s ; and

VTT = volumetric thermal tension, kW/m³.

SILVA *et al.* (1991) found the values of volumetric thermal tension for wood-fired furnaces to be between 177 and 278 kWm^{-3}

2.5.4. Determination of the furnace grate area

The size of the grate was determined by considering a fuel loading rate of 47 kg m^{-2} h^{-1}, the value reported by GOMES (1988) and the air consumption for burn in 1 kg in fuel. The grate area was determined by Equation 13. The grate free area was determined by Equations 14 and 15 respectively.

$$\mathrm{A_g} = \frac{3600.\,\dot{\mathrm{m}}_{\mathrm{fuel}}}{\mathrm{FLR}} \quad (13)$$

$$\mathrm{A_{free}} = \frac{3600.\,\dot{\mathrm{m}}_{\mathrm{fuel}}.\,\mathrm{V_T^{air}}}{0{,}20.\,\mathrm{v_{air}}} \quad (14)$$

$$\mathrm{A_{free}} = \mathrm{fa.\,A_g} \quad (15)$$

Where,

A_g = total grate area, m²;

FLR = fuel loading rate, kg of firewood, h^{-1} m^2

A_{free} = grate free area, m²;

V_T^{air} = theoretical volume of air to burn 1 kg of fuel, m³ of air kg^{-1} of fuel;

v_{air} = primary air velocity, m s^{-1};

fa = relation factor among total a free grate area (0.14 to 0.25)

2.5.5. Determination of heat exchanger area

The surface area of the heat exchanger was determined as a function of heat transfer in the combustion chamber, Volume of combustion chamber and Global Temperature Coefficient (GTC) by Equation 16.

$$A_{she} = \frac{q_{air}.1000}{U.\Delta T_{la}} \tag{16}$$

Where;

A_{she} = Surface area of heat exchanger, m²;

q_{air} = Rate of heat flow, kW;

U = Global temperature coefficient, (W m^2. ^{0}C)

ΔT_{la} = Volume of combustion chamber, m^3

The global temperature coefficient (30 W m^{-2} ºC $^{-1}$) considered in the calculations reported by (EDINEY, 2007).

But, the surface area of heat exchanger (ΔT_{la}) was determined as a function of the logarithmic mean of the fluids temperatures by Equation 17.

$$\Delta T_{la} = \frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \tag{17}$$

$$\Delta T_1 = T_{in\_ha} - T_{out\_a} \quad (18)$$

$$\Delta T_2 = T_{out\_ha} - T_{in\_a} \quad (19)$$

Where;

T_{in_ha} = inlet temperature of overheated air ºC;

T_{out_a} = outlet temperature of overheated air, ºC;

T_{in_a} = inlet temperature of the air to be heated, ºC;

T_{out_a} = outlet temperature of the air to be heated, ºC

2.6. Determination of airflow rate

The inlet/outlet air channel for primary air, chimney, heat exchanger and mixing air chamber was made up of rectangular shape, and the airflow rate was determined by means of Equation 20. The drying air outlet was made from cylindrical pipe and the airflow rate was determined from Equation 21.

$$\mathrm{Q} = \mathrm{L.W.V_{in/out.air}} \quad (20)$$

$$\mathrm{Q} = \pi \mathrm{r}^2 . \mathrm{V_{dry.air}} \quad (21)$$

Where,

Q = airflow rate; $m^3.s^{-1}$

$V_{in/out.air}$ = inlet /outlet air speed; $m.s^{-1}$

L = length of primary air inlet channel; m

W = width of primary air inlet channel; m

r = radius of cylindrical channel of the drying air outlet; m

$V_{dry.air}$ = drying air speed; $m.s^{-1}$

2.7. Modeling and simulation

System modeling is the process of producing a model, and a model is considered a representation of a system of interest or a logical description of how a system performs. A model is considered similar to but simpler than the system it represents. The major function of a model is to enable the prediction of the effect of any changes on the system. A model has a close approximation to the real system and incorporates most of its salient features (ANDRADBTTIR et al., 1997; STEWAR, 1997).

A model could be physical (prototype of a system) or mathematical that is developed with the help of simulation software such as Extend, Arena, and Powersim which are mostly used for simulation (KLEIJNEN, 1995). Mathematical models may include algebra, calculus, probability theory, or experimental equations which can be classified as: static or dynamic model - the static model predicts the system state for a specified moment, whereas the dynamic model considers the time variable advances during the simulation, so it can be used to predict the system states for different moments; stochastic or deterministic model - the stochastic model involves one or more random variables but the deterministic does not; and continuous or discrete model - in continuous models, the variable time value advances during the simulation according to a predefined increment, whereas indiscrete models, the variable time value advances according to the occurrence of events that change the state of the system (SILVA et al., 2012).

2.7.1. Extend-simulation language

Extend is a graphically oriented simulation language developed by Imagine That, Inc., which can be used for performing discrete and continuous simulation (KRAHL, 2000). The Extend simulation environment provides an integrated structure for building simulation models and developing new simulation tools (DAVID, 2001).
Extend models are constructed with library-based iconic blocks which can be applied to run pre-assembled and/or assemble models using the standard blocks or to program or modifying existing code blocks in the library (DAVE, 2016).

Each library represents a grouping of blocks with similar characteristics such as Discrete Event, Plotter, Electronics, or Business Process Reengineering. Blocks are placed on the model worksheet by dragging them from the library window onto the worksheet before establishing flow between them (DAVID, 2001). There are two types of logical flows between the Extend blocks. The first type of flow uses “objects” that move through the system while the second type of logical flow is “values”, which will change over time during the simulation run. During and after the simulation run, the results of the simulation are reported within the blocks, displayed on plotters or sent to reports, and can copied or exported to other applications.

3. MATERIALS AND METHOD

3.1. System description

The wood furnace was projected, built, and evaluated with the following demands: thermal energy from 15 to 30 kW (12,898 to 25,795 kcal. h^{-1}), drying air flow rate from 35 to 45 m³ min^{-1}, drying air temperature from 50 to 80 °C. The prototype was built in the Sitio Santa Luzia (20^0 52’ 43” South, 42^0 51’ 21” West, altitude 774 m) located in Monte Celeste, a district of São Geraldo, Minas Gerais State, Brazil.

This furnace presents two options of operation, indirect and mixture-fired. Indirect-fired occurs when drying air is heated using the heat exchanger, while mixed-fired complementary heating comes from the resulting combustion gas directly. For proposing operational procedures, a simulation model, developed according to experimental data was used.

3.2. Furnace design calculation

Considering the demands of thermal energy, drying air flow rate and drying air temperature highlighted above, the design of a furnace that operates simultaneously in indirect and mixed-fired, that minimizes the man-hour operation and the emission of smoke into the environment was proposed.

The methodology used in the design of the wood furnace consists of the following calculations: (i) the thermal energy required for heating air, (ii) furnace fuel consumption, (iii) the combustion chamber volume, (iv)grate area, and (v) heat exchanger dimensions

3.2.1. Determination of thermal energy required for heating of air

The useful energy of a furnace consists of the energy made available for heating the air, which can be calculated by mean of Equation 9, considering the specific heat of the air constant and equal to 1.0035 kJ kg $^{-1}$ ºC $^{-1}$ (CEVIZ and KAYMAZ, 2005). Also, the atmospheric pressure calculated for the Monte Celeste, Sao Geraldo, Minas Gerais, the location of the site (altitude of 774 meters) where the experiment was conducted was 92.37 kPa.

Thus, considering data shown in Table 2 and using the Equation 9, the thermal energy required for heating the air is 18.41 kW (66,264 k cal. h^{-1}).

Table 2: Thermal energy required for heating the air.

Properties	Units	Value
Air flow rate	$m^3.min^{-1}$	44
Air density	$kg.m^{-3}$	1.012668
Drying mass flow rate	kg s^{-1}	0.73
Specific heat of air	kJ kg^{-1} ^0C	1.00400
Drying air temperature	^0C	70
Ambient air temperature	^0C	45

3.2.2. Determination of firewood consumption
3.2.3.

By determining amount of useful energy to be supplied by the furnace per unit of time, the fuel (firewood) consumption can be determined by means of Equation 10.

Thus, from the thermal energy obtained and other data shown in Table 3 and using the Equation 10, the firewood consumption estimated is 0.0020 kg s^{-1} (7.03 kg h^{-1}).

Table 3: Firewood consumption

Properties	Units	Value
Energy required	kW	18.41
Lower calorific value	kJ kg^{-1}	15,533.31
Estimated efficiency of furnace	%	60

3.2.4. Determination of volume of the combustion chamber

The volume of the combustion chamber is the most important requirements in designing furnaces, since all energy released from combustion process would be converted into gases. The volume must accommodated the gases developed from the flame and must ensure complete combustion before the hot gases got escaped.

The volume of the combustion chamber can be calculated by means of Equation 12.

Table 4: Volume of combustion chamber

Properties	Units	Value
Firewood consumption	kg s^{-1}	0.0020
Lower calorific value	kJ kg^{-1}	15,533.31
Volumetric thermal tension (VTT)	kW m^{-3}	177

Thus, from Table 4 and using the Equation 12, the volume of combustion chamber is approximately 0.30 $m^{3.}$

3.2.5. Determination of the furnace grate area

The area of the grate was determined by considering a fuel loading rate of 47 kg m^{-2} h^{-1}, the value reported by GOMES (1988) and the air consumption for burn in 1 kg in fuel. The grate area was determined by Equation 13. The grate free area was determined by Equations 15.

Table 5: Total grate area and free area of the grate

Properties	Units	Value
FLR	kg h^{-1} m^2	47
$\dot{m}_{fuel}$	kg s^{-1}	0.0032
V_T^{air}	m^3 kg^{-1}	15533.31
fa	(0.14 to 0.25)	0.14

Thus, from the data shown in Table 5 and using the Equation13 and 15, the total grate area and grate free area of the furnace are 0.247 m^2 and 0.035 m^2 respectively.

3.2.6. Determination of heat exchanger area

The surface area of the heat exchanger was determined as a function of heat transfer in the combustion chamber, Volume of combustion chamber and Global Temperature Coefficient (GTC) by Equation 16.

The global temperature coefficient (30 W m $^{-2}$ ºC $^{-1}$) considered in the calculations reported by (EDINEY, 2007).

But, the surface area of heat exchanger (ΔT_{la}) was determined as a function of the logarithmic mean of the fluids temperatures by Equation 16.

Table 6: Surface area of heat exchanger

Properties	Units	Value
Global Temperature Coefficient	W m $^{-2}$ ºC $^{-1}$	30
T_{in_ha}	ºC	650
T_{out_a}	ºC	250
T_{in_a}	ºC	45
T_{out_a}	ºC	70
ΔT_1	ºC	306.84
ΔT_2	ºC	-68.16
ΔT_{la}	ºC	87.41

Thus, from the data shown in Table 6 and using the Equation 16, the surface area of the heat exchanger is 1.70 m^2.

The isometric view of the furnace is as shown on Figure 1, and Figure 2 shown the exploded view showing various components of the furnace.

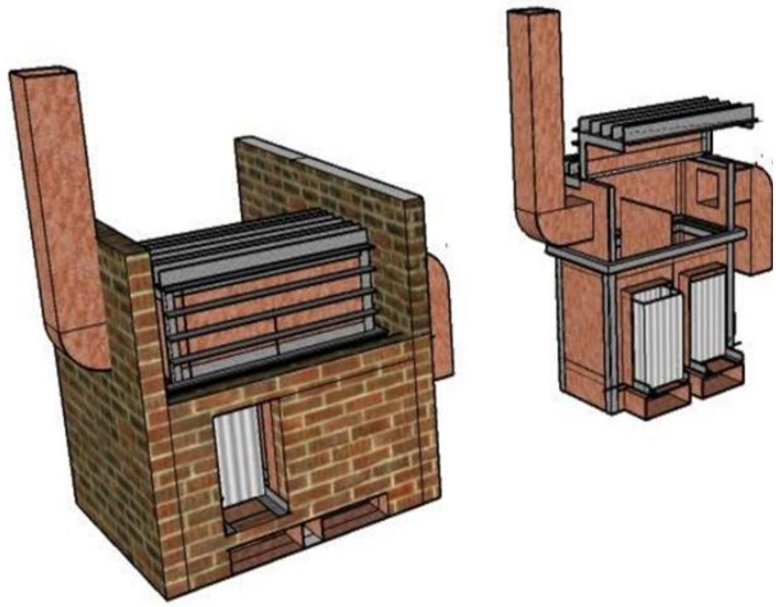


Figure 1: Isometric view of the wood furnace (Source: Author)

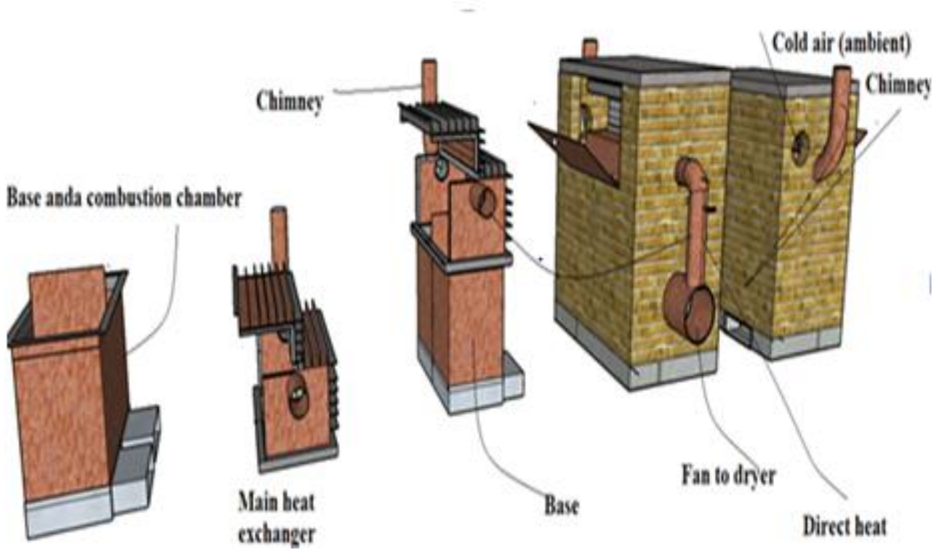


Figure 2: Exploded view of the wood furnace showing various components (Source: Author)

3.3. Furnace working system

As shown in Figure 3, the developed furnace consists of five compartments: (2) combustion chamber, (2) preheating of the drying air, (3) heat exchanger, (4) direct fire mixing chamber, and (5) drying air mixing chamber.

The combustion chamber consists of a grate, where the load cell was placed. This cell, built in steel plates, shaped as a parallelepiped with a screened bottom for firewood loads, with recharging planned every two hours. The recharging procedure consists of replacing an empty load cell with a charged one. The replacement takes a maximum of two minutes.

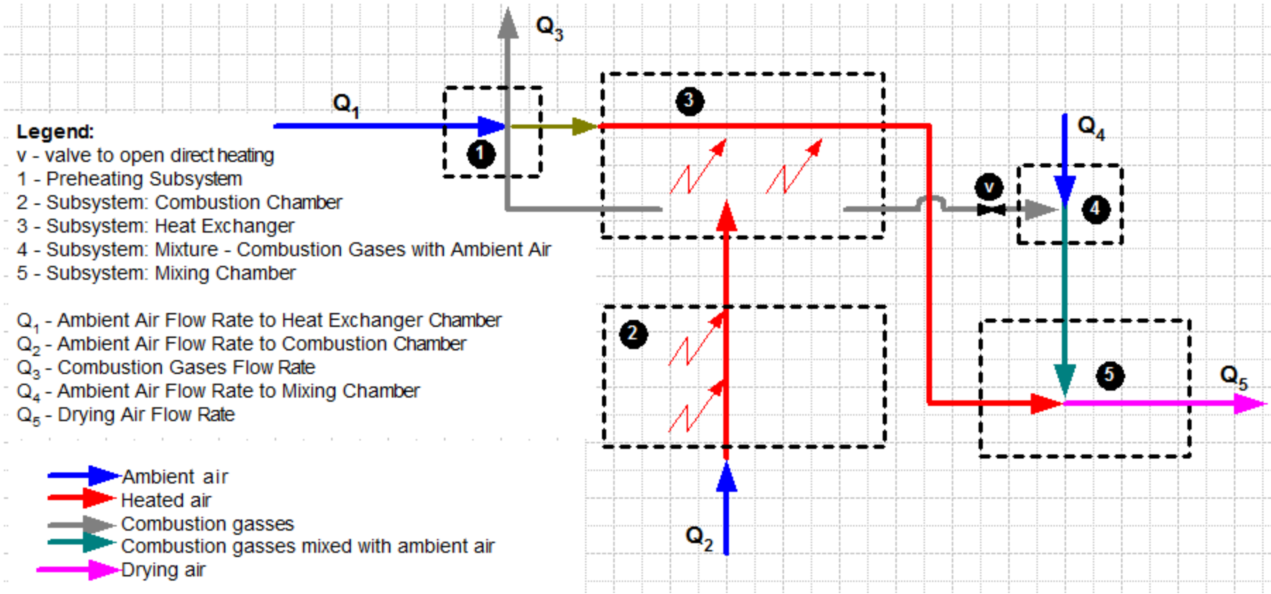


Figure 3: Furnace working system showing the flow of gases inside the furnace (Source: Author).

At preheating of the drying air compartment, the ambient air sucked by the fan coupled to the furnace was heated as it passes through the metal tube of the chimney and cross the surface of the heat exchanger.

The heat exchanger, built in steel plate, was installed on the ceiling of the combustion chamber, and the outflow of the air passed through a duct connected to the drying air mixing chamber. Using the heat exchanger, the furnace operates as indirect fired.

The direct fire mixing compartment, Figure 3, consists of the T-shaped connection of the ducts in which the combustion gases and ambient air flow, respectively. In this connection, the flows mixed, if the valve installed in the first duct is open, which must occur when operating the furnace as direct fire. Otherwise, only ambient air flows into the drying air mixing chamber.

The drying air mixing chamber has one outlet and two inlets. At the outlet, a centrifugal fan was installed with two functions. The first is to blow the drying air, and the second is to suck flows from the heat exchanger and the mixing direct fired compartment. Thus, when the furnace operates as indirect fired, in the drying, in mixing chamber, the ambient air is mixed with flows from the heat exchanger, However, if the valve in the direct fire duct is open, in the mixing drying air chamber, mixed air flow from the heat exchanger is mixed with the mixed air from direct fired. In this situation, the operation of the furnace is referred as a mixed fired operation.

3.4. Field experiments

To evaluate the performance of the furnace, several tests were carried out for both indirect and mixture configuration of the furnace. During the test, the weight of fuel tank (16 kg) filled with firewood was measured on a scale, with a sensitivity of 0.1 kg and the weight different was calculated to know the actual weight of the firewood, 2-3 pieces of charcoal soaked with ethanol were used to ignite the wood top surface in the tank before it was introduced inside the combustion chamber and the door was closed. The centrifugal fan was switched on and each test took an average duration of two hours. For each test, measurements such as primary air speed, drying air speed, mixing air speed, flue gasses speed outlet at the chimney, inlet air speed at the heat exchanger, ambient temperature and relative humidity, burning flame temperature, temperature of the gasses inside the furnace, heat exchanger and drying air temperatures were monitored at time intervals. The proportion of CO and CO2 in the drying air outlet and flue gasses outlet was also monitored. Each of the tests was replicated 4 times and the summary of the experimental trials for different dates are as presented below:

26th July, 2022 ➡ The first day of the trial, the furnace was instrumented by positioning the temperatures sensors at various and appropriate points for measurement, the furnace was test-run to take measurements to establish a good working condition of the sensors

27th July, 2022 ➡ The second days of the trial, three replications of test were carried out. For each of the replications, the furnace was tested for indirect and mixture configuration at the same time by switching between the two methods to monitor speeds and temperature at various points.

28th July, 2022 ➡ The third days of the trial, four replications of the test were carried out using indirect heating configurations. The speed and temperature at various positions on the furnace were monitored. Also, the CO and CO_2 proportion were monitored at the drying air and chimney flue gasses outlet respectively.

2nd August. 2022 ➡ The fourth days of the trial, four replications of the test were carried out using mixture heating configurations. The speed and temperature at various positions on the furnace were monitored. Also, the CO and CO_2 proportion were monitored at the drying air and chimney flue gasses outlet respectively.

3.4.1. Instruments and method of measuring temperatures

To monitor, evaluate and control the operation of the furnace, temperature is most important parameter that influence the performance of the furnace and was studied: temperature of dry bulb and relative humidity of ambient air, temperature from combustion gases, temperature of the heated air, temperatures in the walls of the furnace, temperature of the bed of firewood in the burning tank, gas temperature inside and outside the furnace, temperature of the heat exchanger plate, temperature of the grate door were monitored with temperature sensors.

3.4.1.1. Positioning of the temperature sensors

Previously calibrated type K thermometric probes (Figure 4) were used to measure the gas temperatures (ranges 45-1110^oC), the ambient temperature and relative humidity was measured using thermo-hygrometer (Figure 5) and an infrared gun-thermometer (Figure 6) was used to measure the temperature of the wall and that of the grate door.

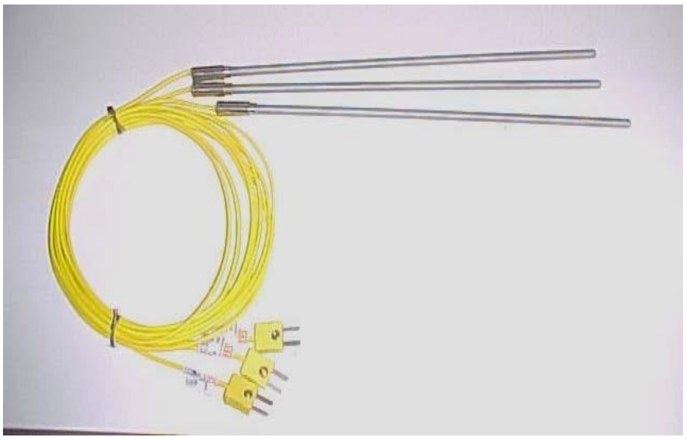


Figure 4: Type K thermometric Probes (Source: Author)

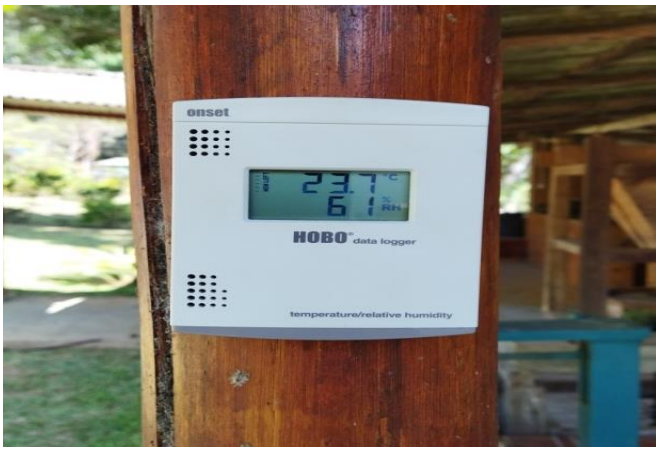


Figure 5: The Thermo-Hygrometer (Source: Author)

Figure 6: Infrared Gun-thermometer (Source: Author)

The positioning of the temperature sensors at various point of measurement is as shown on Figure 7. The temperature of the drying air (T1) at the drying air outlet was measured by a sensor located at 0.9 m away from the fan, the combustion chamber/the firing cell temperature sensor (T2) was located just at the top of the firing cell, and the temperature sensors T3 and T7 measure the temperature of heat exchanger and the exchanger fins respectively. T4 measures the temperature of gasses directly from the combustion chamber when the furnace is operating in direct-fired, T5 measures the mixture of ambient air with the temperature T4, the temperature of the gases at the outlet of the furnace when the furnace is working in indirect configuration was measured by a sensor (T6) located at the gas outlet from the heat exchanger, T8 measures the temperature of the pre-heated ambient air that enters the heat exchanger and the sensor T9 measures the temperature of the gasses at the chimney.

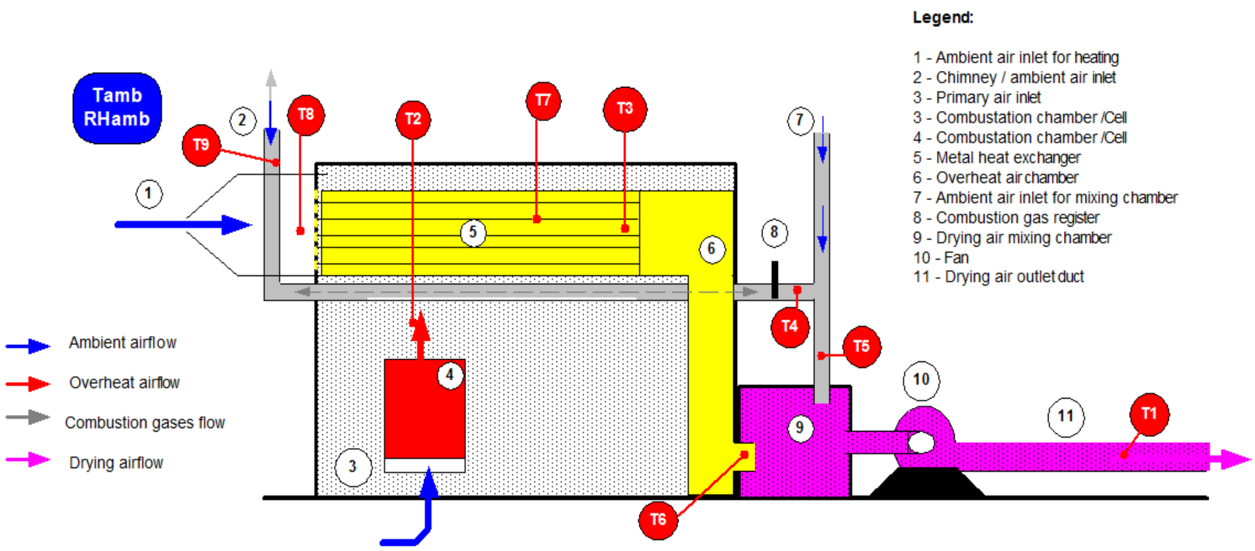


Figure 7: Positioning of the temperature sensors at various points of measurement (Source: Author).

The pictorial view of the furnace setup showing the position of various temperature sensors is as shown on Figure 8.

Figure 8: pictorial view of the furnace setup (Source: Author)

3.4.2. Instruments and method of measuring air velocities

To calculate the excess air, the velocity of the combustion/primary air that enters the combustion chamber through the primary air inlet situated directly opposite the combustion chamber was measured, using a digital anemometer with a reading accuracy of 0.1 ms^{-1} as shown on (Figure 9).

Figure 9: Measurement of the oxidizing air velocity at the primary air inlet (Source: Author)

To obtain the drying air flow rates, the digital anemometer was also used to measure the velocity of the drying air at the drying air outlet as shown on Figure 10.

The measurement was repeated severally to be able to obtain accurate average value of the velocity of the drying air.

Figure 10: Measurement of the drying air velocity at the air inlet (Source: Author)

The outlet flue gas speed from the chimney, the inlet air speed from heat exchanger and mixing chamber were also measured with the same instrument.

The volumetric flow rate of inlet primary air, heat exchanger, mixing chamber and flue gasses outlet from the chimney was calculated by Equation 17 and outlet drying air was calculated as a function of the air velocity and the area of the channel through which the air flows by Equation 18.

3.4.3. Instrument and measurement of quality of combustion gases

The flue gases during combustion process were monitored and analysis with two different equipment: a portable CO_2 meter (7755/77535 Models) and a portable CO meter ITMCO-500 (Model 7701). The portable meters use NDIR (Non-Dispersive Infrared) technology to ensure the reliability and long-term stability but can be damaged with temperature above 75 ^{0}C. The flue gasses qualities were monitored at the chimney and the drying air outlet. The measurement was done directly from the drying air outlet as shown on Figure (11).

Figure 11: Positioning of the Sensors during the Measurement of Drying air Quality (Source: Author)

An electric pumping machine was used to sucks the gasses out of the chimney because of high temperature (200-250^{0}C) that can damage the sensor. The setup for the measurement consists of an electric pump positioned in a box, a tube that connected the pump to the chimney and a container to retain the water of condensation the was produced as a result of suction of the flue gasses as shown on the Figure (12).

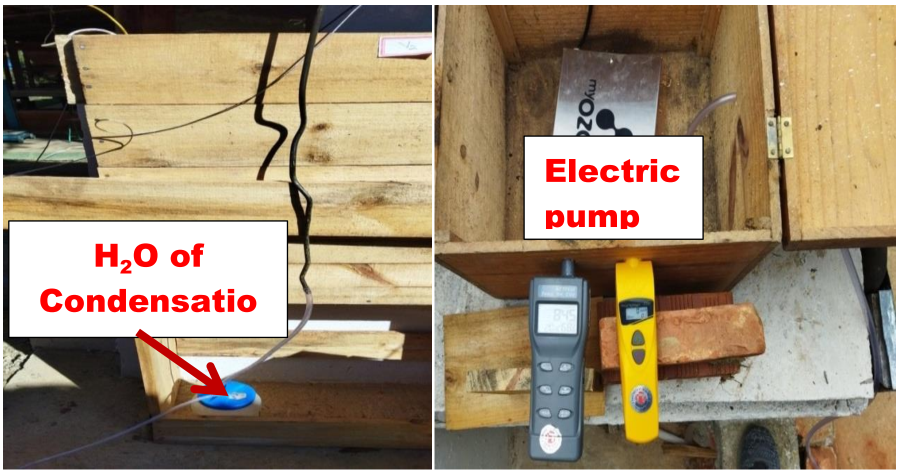


Figure 12: Setup for measuring the Quality of Flue Gasses from the Chimney (Source: Author).

3.4.4. Measurement of fuel consumption

To determine the fuel consumption, a scale of platform with a capacity of 200 kg and accuracy of 0.1 kg was used. During the test, the weight of empty fuel tank and a full load of fuel-tank with firewood were determined. The time taken for the firewood to get completely burnt through combustion processes was monitored. At the end of the tests, the fuel consumption was obtained by dividing the weight of firewood with time taken for fuel to completely burnt.

3.5. Description and preparation of fuel

The firewood used for the evaluation was Eucalyptus (Eucalyptus *sp.*), cut into pieces of approximately the same height and diameter. The pictorial view of the arrangement of the fuel before the combustion process is as shown on Figure 13.

Figure 13: Stock of eucalyptus firewood used for the experiment (Source: Author)

The higher calorific value of the fuel was determined with a calorimeter bomb, at the Madeira Energy Laboratory, according to ASTM D-2015-66 standards. The lower calorific power was calculated using Equation (2), presented in item 2.1. Since the lower calorific power depends on the fuel's elemental composition, firewood's elemental composition on a dry basis was reported by VLASSOV (2001) with average values: carbon 47.5%, hydrogen 6.0%, oxygen 44.0%, nitrogen 1.0%, and ash 1.5%.

The moisture content of the fuel was also determined at the Madeira Energy Laboratory, according to ASTM D4442 standard. Four samples of the total fuels were randomly selected, from which the moistures were determined; the fuel samples used in each test were placed in an oven for 24 hours, at a temperature of 105°C, until reaching constant weight. The moisture content was calculated as percentage of fraction of dried and wet samples. The specific mass of firewood was calculated as a function of the volume of the fuel tank and mass of the firewood.

3.6. Calculation of the furnace thermal efficiency

The thermal efficiency of the furnace was determined based on variation of heat energy of the ambient air that enters the system and the thermal energy provided by the fuel for the system to heat up the air volume. Considering the mass balance shown in Figure 3, and that:

- combustion is complete.
- the amount of carbon particles entrained or present in the ash is negligible.
- the system operates in a steady state; and
- the difference between the calculated and the ideal efficiency is due exclusively to the heat losses in the furnace (PAYNE and CHANDRA, 1985, cited by LOPES, 2002).

The thermal efficiency of the furnace was determined by Equation 22.

$$ב = 60 \frac{\rho_{ar} Q_{ar} Cp_{ar} (T_s - T_{ar\ amb})}{M_{cb} LHV} \quad (22)$$

Where;

ρ_{ar} = Specific mass of ambient air, kg m^{-3};

Q_{ar} = Heated air flow, m^3 min^{-1};

Cp_{ar} = Mean specific heat of air, kJ kg^{-1}.°C

T_s = drying air temperature, °C; and

$T_{ar\ amb}$ = Ambient air temperature, °C

3.7. Modeling and Simulation of the furnace performance

Considering the regressions adjusted for temperature and air flow, a dynamic, continuous, and stochastic model was developed to simulate operational scenario, considering drying air temperatures ranging from 50 to 80°C, and furnace operating in indirect or mixed fire. To develop the model, the software Extend^{TM}, a simulation language, was employed.

A continuous model was developed with a block named "Control" in which the main function is to control the advance of time variable called "*cycleTime*", that maximum value corresponds to the time without refueling. The advance of "cycleTime" occurs in increments of one minute.

In the dialog windows of “Control Block" the user needs to define the working time without refueling - h, and period that furnace operated as indirect and mixed fired. This way at each minute of the advance the "Control Block" will report the "cycleTime" value, and if the furnace is operating as indirect or mixed fired.

In “Control Block", user also have to input altitude - m, and specific heat of air - kJ. Kg^{-1}. K^{-1}. These variables are continually made available to the other blocks in the model.

Other variables continually made available by “Control Block" are ambient air temperature - °C, and relativity humidity - %. Values of these variables are obtained from time series, which correspond to the moving average of experimental data.

The other blocks in the model are: “Combustion Chamber”, “Pre-Heating”, “Heat Exchanger”, Air Pre-Mixing, Mixing Chamber, and Efficiency.

- “*Combustion Chamber Block*” – according to the value of the "*cycleTime*" and operation option, at each minute advance are calculating the temperature in the combustion cell, temperature of gases in the chimney, temperature in the combustion chamber, and temperature of gases direct-fired duct if it is open. The equations used to calculate the temperature is show in Tables 10 and 11 respectively for indirect and mixed-fired operations.
- “*Heat Exchanger Block*” – the outlet temperature of heat exchanger is calculated at each minute advance of "*cycleTime*" and operation option an according to equations in Tables 10 and 11.
- “*Air Pre-Mixing Block*” - In this block, the air outlet temperature and mass flow are simulated. This air flow is direct to the drying air

mixing chamber. When the furnace option is operated as indirect fired, the outlet air flow corresponds to the ambient air condition. Otherwise, it is simulated the mixing of ambient and gases from the combustion chamber. In this simulation is taken the temperature and mass flow rate of ambient air, the temperature gases from the combustion chamber and the temperature of air flow after the mixing. Thus, by means of mass and heat balance equations, it is possible to calculate the mass flow rate after the mixing process.

- "Mixing Chamber Block" - this block simulates the mixing of air flows in the drying air mixing chamber (Figure 3), which has two inlets and one outlet. In the simulation is known the temperature and mass flow rate of air that comes from the direct fired mixing chamber (Figure 3) and the temperature of air flow that comes from the heat exchanger, and the outlet temperature and mass flow rate of the drying air. Using mass and heat balance is calculated the mass flow rate that comes from the heat exchanger. Simultaneously, in this block, the air flows mixing simulation considers two situations without control, and with control, when the user needs to inform the desired drying air temperature. Thus, during the simulation, this block reports: the moment that inlet of ambient air in the direct fired mixing chamber (Figure 3) should be open, and mass flow rates and temperatures of air in the inlets and outlet.
- "*Efficiency Block*" - Psychrometric properties of ambient air and drying air calculated continuously during the simulation. And knowing the fuel properties and drying air flow rate, at each increment as "cycleTime" value is simulated the efficiency - % (Equation 12), energy flow rate supplied by combustion kW, and useful energy flow rate - kW.

The pictorial view of the main window of the Extend^{TM}, a simulation model developed is shown on Figure 14.

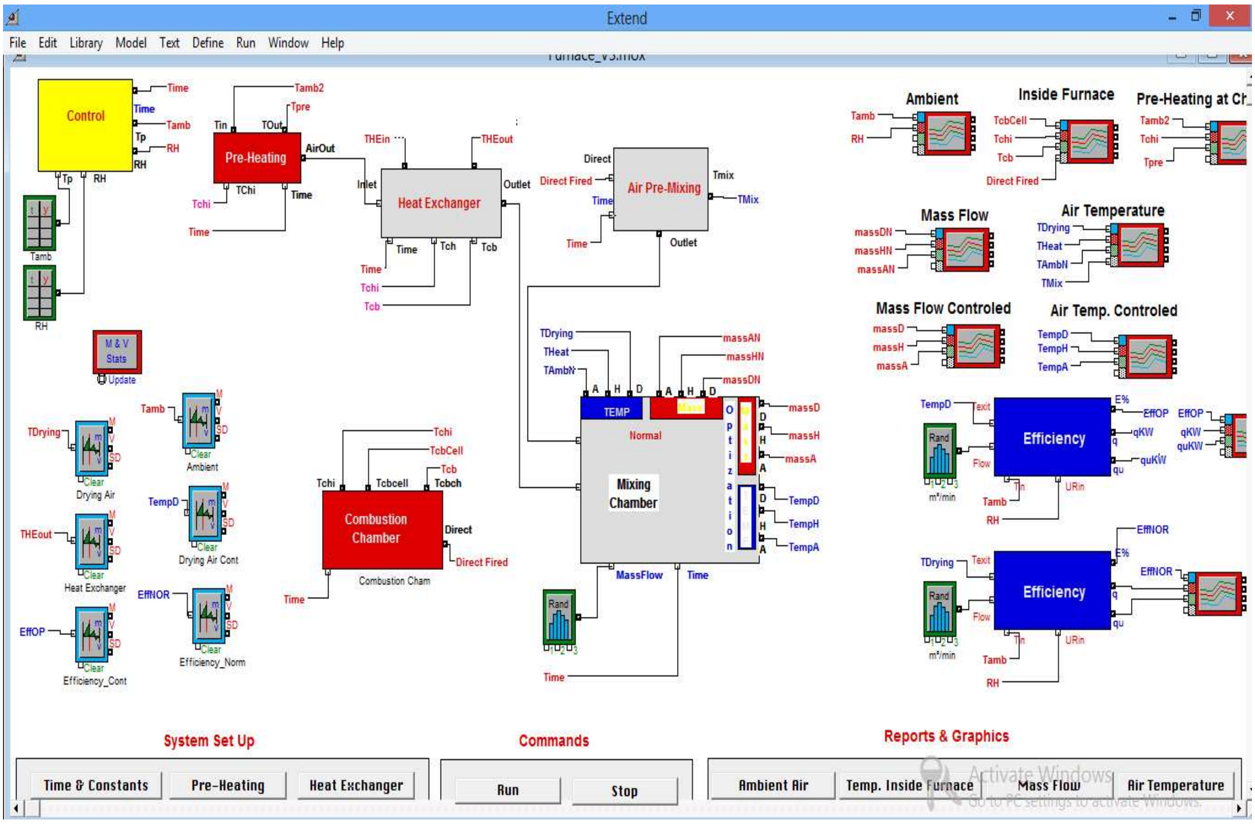


Figure 14: Pictorial view of the main window of the Extend^{TM}, a simulation model (Source: Author)

4. RESULTS AND DISCUSSION

4.1. Theoretical and actual values of furnace dimensions

In Table 8, the dimension of the furnace is recalculated according to the measurement obtained during the evaluation of the furnace and compared with the designed dimensions used to build the prototype. However, the obtained data for the calculation of the actual values are shown on Table 7.

Table 7: Obtained data used for the calculation of the actual values

Properties	Units	Value
Thermal energy required for heating the air		
Drying mass flow rate	kg s^{-1}	0.68
Air density	kg m^{-3}	1.012668
Specific heat of air	kJ kg^{-1} ^{0}C	1.00400
Drying air temperature	^{0}C	73.81
Ambient air temperature (preheated)	^{0}C	47
Firewood Consumption and volume of combustion chamber		
Thermal energy required for heating the air	kW	18.3
Lower heating value	kJ kg^{-1}	15533.31
Estimated efficiency of furnace	%	0.6
Firewood consumption	kg s^{-1}	0.0020
VTT	kW m^{-3}	177
Grate area and grate free area		
FLR	kg h^{-1} m^2	47
$\dot{m}_{fuel}$	kg s^{-1}	0.0032
V_T^{air}	m^3 kg^{-1}	15533.31
fa	(0.14 to 0.25)	0.14
Surface area of heat exchanger		
Global Temperature Coefficient	W m^{-2} ºC $^{-1}$	30
T_{in_ha}	ºC	660.42
T_{out_a}	ºC	246.38
T_{in_a}	ºC	47
T_{out_a}	ºC	73.81
ΔT_1	ºC	313.45
ΔT_2	ºC	-73.78
ΔT_{la}	ºC	85.67

Table 8: Theoretical and actual values of the furnace dimensions

Properties	Units	Theoretical value	Actual value
Thermal energy required for heating the air	kW	18.41	18.31
Firewood consumption	kg h^{-1}	7.03	7.0719
Volume of the combustion chamber	m^3	0.30	0.30
Furnace grate area	m^2	0.247	0.247
Grate free area	m^2	0.035	0.035
Surface area of heat exchanger	m^2	1.70	1.70

Table 8 reflects that there is no significant difference from the theoretical values calculated and actual value obtained from the results of the evaluation. Thus, validated the design and shown that prototype built is a true representation of the designed furnace.

4.2. Properties of the fuels used

The results obtained for the moisture content, higher calorific value, ash content, percentage volatility and specific mass of the fuel are presented in Table 9.

Table 9: Properties of the fuel used

Properties	Units	Value
Moisture content	% wb	19.399
Higher calorific value	kJ kg^{-1}	19260.2
Ash content	%	0.11
Carbon content	%	12.7125
Volatility	%	87.1775
Specific mass	kg m^{-3}	440

Thus, the firewood properties obtained are within the range recommended for good thermal energy production and are similar to the values reported by (DIEZ AND PEREZ, 2017; IGNACIO, SANTOS AND DUARTE, 2019; LUBWAMA *et al.*, 2021; RESENDE *et al.*, 2022).

4.3. Process control

4.3.1. Process control for temperature data

The process control for the data obtained for the combustion temperature, heat exchanger temperature, temperature of gasses out of heat exchanger, the drying air temperature and other various measurement positions are shown on Table 10 and 11 for indirect-heating and mixed-heating operation respectively. The maximum (Max), mean (Average), minimum (Min) and standard deviation (StaDiv) were obtained at 95% confidence interval (CI).

Table 10: Temperature control data for operational position indirect-heating for a cycle time of 119 minutes (1.98 h)

Temperatures (^{0}C)	Max	Average	Min	StaDiv	95% CI
Ambient air temperature	26.42	23.60	21.32	1.42	23.60 ± 0.40
Drying air temperature – T1	75.06	59.52	34.52	12.04	59.52 ± 3.414
Combustion cell - T2	834.80	639.13	299.33	158.09	639.13 ± 44.72
Heat exchanger – T3	364.00	253.48	96.50	74.72	253.48 ± 21.14
Temperature out of heat. Exchanger – T6	101.38	71.20	38.70	19.12	71.20 ± 5.41
Temperature of fin plate – T7	238.00	141.50	58.40	54.11	141.50 ± 15.31
Preheated air for heat exchanger – T8	146.00	88.24	38.60	34.82	88.24 ± 9.85
Chimney temperature – T9	354.40	239.22	87.00	77.94	239.22 ± 22.06
Furnace wall temperature – T10	39.78	31.79	25.52	3.94	31.79 ± 1.11

Table 11: Temperature control data for operational position mixed-heating for a cycle time of 67.2 minutes (1.12 h)

Temperatures (^{0}C)	Max	Average	Min	StaDiv	95% CI
Ambient air temperature	25.07	25.00	23.57	0.67	5.00 ± 0.33
Drying air temperature – T1	114.00	75.41	53.03	20.13	75.41 ± 9.87
Combustion cell - T2	875.00	708.70	515.00	100.64	708.70 ± 49.42
Heat exchanger – T3	392.50	227.24	91.00	106.50	227.24 ± 52.18
Temperature of air direct from comb. –T4	332.00	180.11	81.00	77.13	180.11 ± 37.79
Temperature of air mixed with Ambt. T5	329.00	175.71	67.50	76.99	175.71 ± 37.33
Temperature out of heat. Exchanger – T6	77.00	53.32	42.00	11.97	53.32 ± 5.87
Temperature of fin plate – T7	242.00	115.51	53.33	64.62	115.51 ± 31.66
Preheated air for heat exchanger – T8	96.00	50.42	32.67	19.83	50.42 ± 9.72
Chimney temperature – T9	297.00	114.00	26.67	113.20	114.00 ± 55.47
Furnace wall temperature – T10	51.60	48.46	37.93	3.35	48.46 ± 1.64

The process control data in Table 10 and 11 shown that the average values of the temperature obtained at various points for mixed-heating operation are significantly higher than the values obtained for indirect-fired operation, except, for the heat exchanger temperature, preheated air for heat exchanger and temperature of the chimney. This is because, in mixed-heating operation, the heat energy produced by combustion chamber flows directly to the mixing chamber without having much impact on the heat exchanger and the temperature obtained at the heat exchanger plate was as a result of radiations that occur during the heat transfer to the mixing chamber.

Also, the flue gases that flow out of the chimney during indirect-heating operation combined with the drying air during mixed-heating operation, and thus, drastically reduced the temperature of the flue gasses that flows out of the chimney and the heat exchanger.

4.3.2. Process control for mass flow rate data

The process control for the data obtained for the mass flow rate of primary air, chimney, inlet air at mixing chamber, heat exchanger (preheated air) and drying air are as presented in Table 12 and 13for indirect-heating and mixed-heating operation respectively. The maximum (Max), mean (Average), minimum (Min) and standard deviation (StaDiv) were obtained at 95% confidence interval (CI).

Table 12: Mass flow rate control data for operational position indirect-heating for a cycle time of 119 minutes (1.98 h)

Mass flow rates (kg min^{-1})	Max	Average	Min	StaDiv	95% CI
Primary air	11.19	8.06	6.07	1.08	8.06 ± 0.38
Chimney air	2.21	1.40	0.99	0.39	1.40 ± 0.14
Mixing air	26.00	24.38	21.99	1.74	24.38 ± 0.88
Heat exchanger (preheated) air	27.26	19.09	10.41	3.68	19.09 ± 1.30
Drying air	45.86	40.81	37.38	2.10	40.81 ± 0.74

Table 13: Mass flow rate control data for operational position mixed-heating for a cycle time of 67.2 minutes (1.12 h)

Mass flow rates (kg min^{-1})	Max	Average	Min	StaDiv	95% CI
Primary air	11.79	8.75	6.07	1.36	8.75 ± 0.65
Chimney air	6.96	4.41	2.60	1.49	4.41 ± 0.71
Mixing air	15.19	14.01	12.03	0.96	14.01 ± 0.46
Heat exchanger (preheated) air	16.98	15.17	12.89	1.54	15.17 ± 0.73
Drying air	46.25	42.95	39.89	0.67	42.95 ± 0.79

Table 12 and 13 shown that the average values of the primary air and drying air mass flow rate obtained for indirect-heating and mixed-heating operation are not significantly difference, but, the mixing air flow rate for indirect-fired operation (24.38 kg min^{-1}) is greatly higher than the value obtained for mixed-heating operation (14.01

kg min^{-1}). Thus, causes the great difference that occurs in the drying air temperatures of the two operations as described above.

Also, there was a significant difference in the chimney air mass flow rate obtained for indirect-fired (1.40 kg min^{-1}) and mixed-heating (4.41 kg min^{-1}) operation. This is due to the fact that, flue gasses at high temperature flow out of the combustion chamber through the chimney during the indirect-heating operation, but, the flue gasses mixed with the drying air during mixed-heating operation. The mass flow rate obtained at the chimney during the mixed-heating operation was as a result of suction provided by the centrifugal fan installed at the drying air outlet.

4.4. Performance of the furnace with indirect heating of the air

The performance of the furnace, with indirect heating of the air, was studied by stabilizing the drying air temperature between 60 to 70 ^{0}C with regulated amount of inlet mixing air admitted to the mixing chamber for the cooling of the heated air temperature from the heat exchanger. The opening for the mixing chamber was closed for 50 to 55 minute to enable a speedy increase of drying air temperature to above 70 ^{0}C before it was opened and remain opened till the end of the experiment which is approximately 2 hours. The combustion temperature, the temperature of the heat exchanger, the heated air temperature from the heat exchanger and the drying air temperature per unit time were studied.

The regression analysis of the drying temperature data obtained at various point of measurement T1 to T10, (Figure 14) per unit time was carried out and fitted with quadratic model as shown on equation 22.

$$Tp = Yo + a.t_{ct} + b.t_{ct}^2 \qquad (22)$$

Where,

Tp = temperature at a point, ^{0}C

Yo = constant

a = coefficient of circle time

b = coefficient of square of circle time

t_{ct} = circle time, h.

Table 14 shown the fitted models, adjusted coefficients of determination (R^2_{adj}), and standard error of estimate (S) for temperature measurement points during a circle time of 1,98 hours in Indirect-heating.

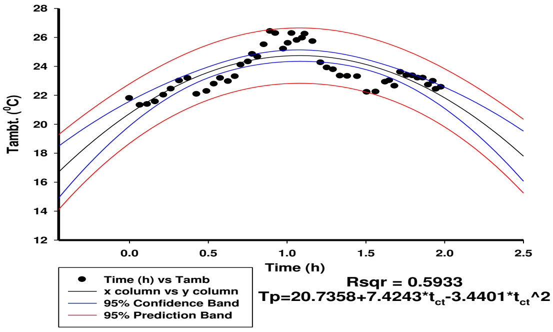
Table 14: Fitted models, adjusted coefficients of determination (R^2_{adj}), and standard error of estimate (S) for temperature measurement points for indirect-heating configuration

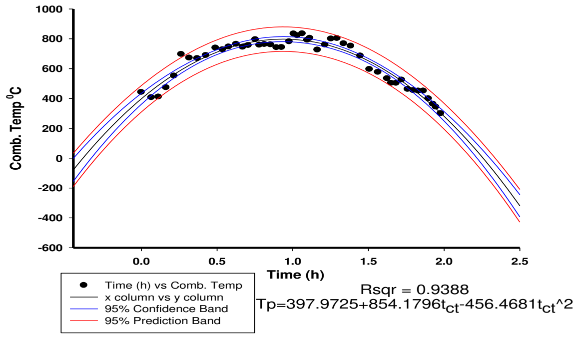
Measure point	Yo	a	b	R^2_{adj}	S
Ambient air temperature	20.7358	7.4243	-3.4401	0.5752	0.9286
Drying air temperature – T1	26.1002	68.8904	-27.1664	0.8743	4.2677
Combustion cell - T2	397.9725	854.1796	-456.468	0.9361	39.9529
Heat exchanger – T3	51.9307	478.8415	-210.049	0.8965	24.0354
Temperature out of heat. Exchanger – T6	27.1704	72.0064	-22.1089	0.7616	9.3378
Temperature of fin plate – T7	5.7392	323.5407	-142.212	0.7735	25.7477
Preheated air for heat exchanger – T8	1.2044	199.561	-85.4397	0.7292	18.1195
Chimney temperature – T9	39.713	510.6565	-234.669	0.9292	20.7443
Furnace wall temperature – T10	31.2607	-6.9355	5.4271	0.5753	2.5666

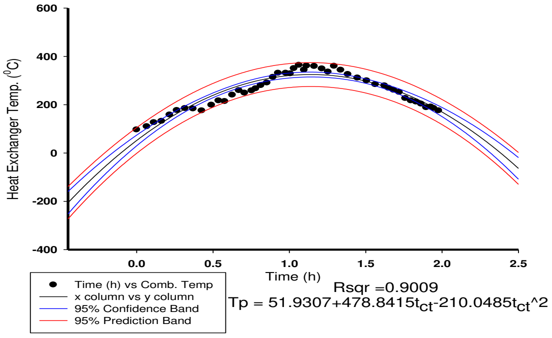
4.4.1. Effects of operation time on the performance of furnace with indirect heating of the air

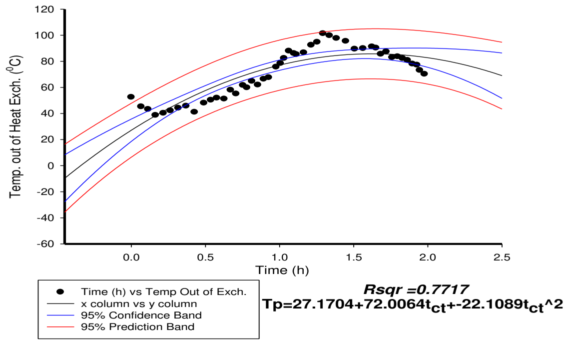
The relationship between the operation time and the ambient air, drying air, combustion, heat exchanger, outlet heated air from heat exchanger, fin plate, preheated air, chimney and furnace wall temperature is as shown on the Figure (15). Non-linear regression analysis of the data fitted with quadratic model was used in the analysis of the data. The figure shows that all the parameters shown positive linear relationship with operation time except the furnace wall temperature, that is, there are gradual increase in temperatures at various positions as the operation time increases, but, the temperatures started declining when the combustion cell temperature has reached its maximum value of approximately 835 ^{0}C.

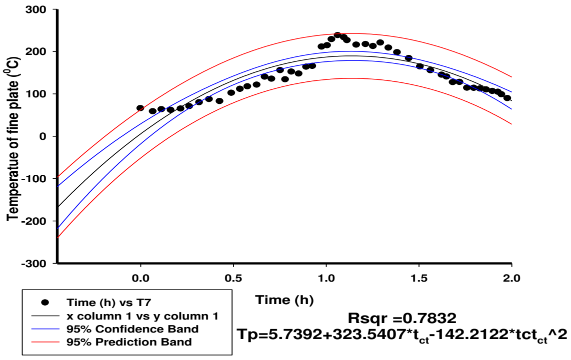
However, the furnace wall temperature tends to remain constant with time, but, increase steadily when the combustion temperature reaches its peak. Also, the results of the analysis showed a strong correlation between the time and parameters under considerations at 95% confidence interval with R^2 values equal to 0.5933, 0.9388, 0.9009, 0.7717, 0.7832, 0.7408, 0.9322 and 0.5934 respectively for the ambient air, drying air, combustion, heat exchanger, outlet heated air from heat exchanger, fin plate, preheated air, chimney and furnace wall temperatures. Thus, it implies that the models significantly predict the behavior of various parameters at an interval of time.

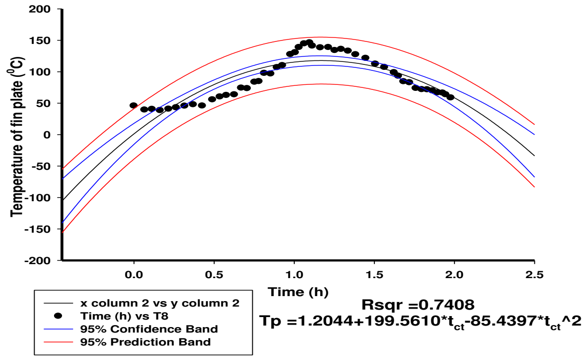


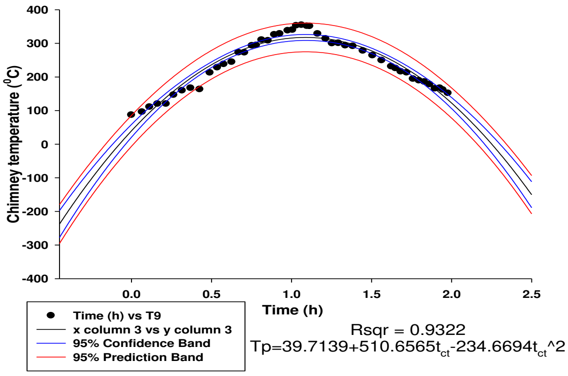












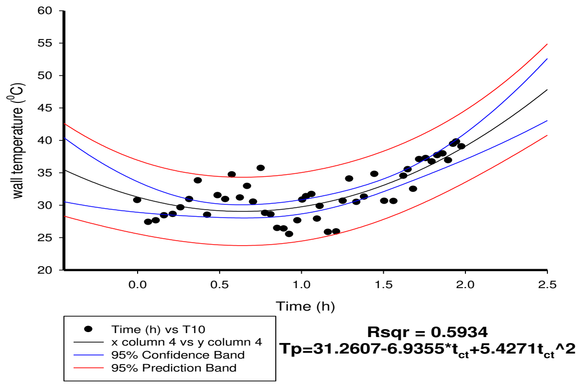


Figure 15: Effects of time on the performances of the indirect-heating furnace.

4.5. Performance of the furnace with mixed heating of the air

The performance of the furnace, with mixed heating of the air, was also studied by stabilizing the drying air temperature between 60 to 70 ^{0}C. The valve for the configuration of mixed-air heating was opened for about 60 minute to convert the system to mixture configuration. Above 60 minute, the drying temperature increased above 70 ^{0}C, then, the opening was closed to return the system to indirect-fired for the remaining hour of the experiment. The combustion temperature, the temperature of the gasses out of combustion chamber, the temperature of the gasses mixed with ambient air and the drying air temperature per unit time were studied.

Table 15 shown the fitted models, adjusted coefficients of determination (R^2_{adj}), and standard error of estimate (S) for temperature measurement points during a circle time of 1.21 hours in mixed-heating.

Table 15: Fitted models, adjusted coefficients of determination (R^2_{adj}), and standard error of estimate (S) for temperature measurement points for mixed-heating configuration

Measure point	Yo	a	b	R^2_{adj}	S
Ambient air temperature	25.4416	-2.3748	2.0923	0.000	0.6811
Drying air temperature – T1	56.1617	-10.0247	56.7682	0.9788	2.9337
Combustion cell - T2	582.8277	156.65	82.0585	0.7095	54.2479
Heat exchanger – T3	58.037	286.2842	11.7798	0.9382	26.4683
Temperature out of heat. Exchanger – T6	67.5268	155.1689	53.813	0.9413	18.6847
Temperature of fin plate – T7	56.2852	197.5867	14.1351	0.909	23.2223
Preheated air for heat exchanger – T8	61.873	-84.7375	90.8602	0.9646	2.2518
Chimney temperature – T9	70.4327	-107.712	242.5843	0.9866	7.468
Furnace wall temperature – T10	49.4503	-86.4939	114.7643	0.9955	1.3274
Ambient air temperature	56.0446	-262.472	473.2525	0.9166	32.7007
Drying air temperature – T1	48.777	-18.0483	11.3999	0.304	2.7988

4.5.1. Effects of operation time on the performance of furnace with mixed heating of the air

The relationship between the operation time and the ambient air, drying air, combustion, heat exchanger, outlet air from combustion chamber, outlet mixed with ambient, outlet heated air from heat exchanger, and chimney is as shown on the Figure (16). Non-linear regression analysis of the data fitted with quadratic model was also used in the analysis of the data. The figure shows that there was no relationship between the operation time and ambient air temperature ($R^2 = 0.00$), all other temperatures shown positive linear relationship with the operation time except the outlet heat exchanger and chimney temperature that shows negative correlation with the operation time, that is, there are steady increase in drying air, combustion, heat exchanger, outlet air from combustion chamber and outlet mixed with ambient temperatures as the operation time increases within the operation time 1.21 hours, when the furnace operate in mixed-heating configuration. The maximum combustion cell temperature obtains within this period reached an approximate value of 875 ^{0}C, which is significantly higher than the value obtained for indirect-heating (835 ^{0}C).

The results of the regression analysis showed a strong correlation between the operation time and drying air, combustion, heat exchanger, outlet air from combustion chamber, outlet mixed with ambient, outlet heated air from heat exchanger, and chimney at 95% confidence interval with R^2 values equal to 0.9816, 0.7482, 0.9465, 0.9491, 0.9212, 0.9693 and 0.9277 respectively. Thus, it implies that the models significantly predict the behavior of various parameters at an interval of time.

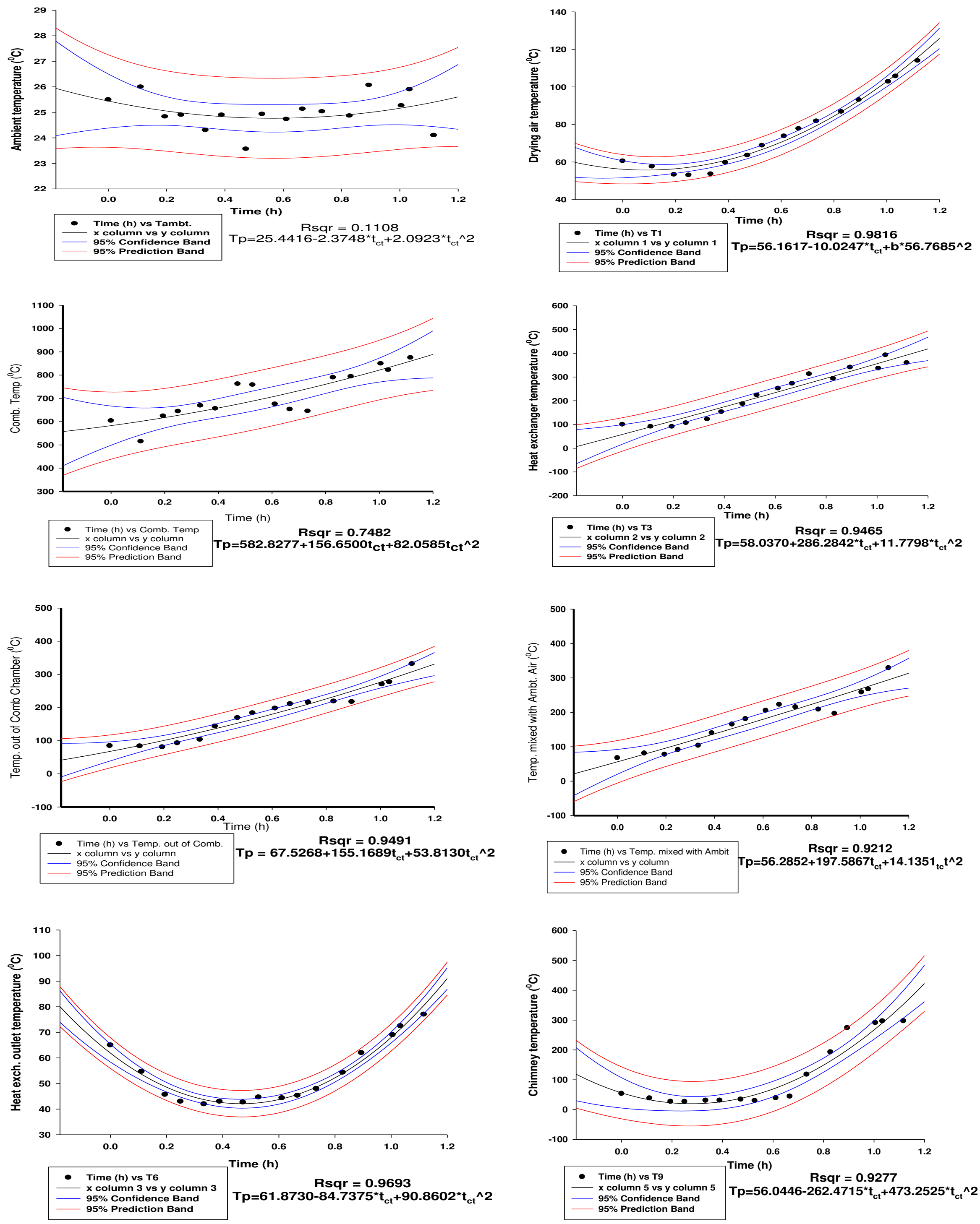


Figure 16: Effects of time on the performances of the mixed-heating furnace.

4.6. Quality analysis of the combustion gasses

The results of the analysis of composition of the flue gasses proportion present in the drying and chimney air during indirect-heating and mixed-heating operations is as presented in Table 16 and Table 17 respectively. The results showed no significant presence of CO in the drying air from beginning to the end of operation cycle and only 2.0 PPM of CO present in the chimney air at the first 10 minute of the indirect-heating operation. Whereas, there is larger proportion of the CO (385 PPM) present in the drying air at the beginning of operation in mixed-heating configuration, while the chimney air was free from CO contaminations.

From the tables, irrespective of the configuration of the furnace, the proportion of CO_2 present in the drying air is significantly higher than the one obtained at the chimney and decreases gradually as the operation time increases. Also, the CO_2 values obtained during operation of furnace in indirect-heating configuration is greatly higher than the one obtained during operation of the furnace in mixed-heating configuration.

Table 16: proportion of flue gasses composition in the chimney and drying air for indirect-heating operation

Operation time (minute)	Proportion CO (PPM)		Proportion of CO_2 (PPM)	
	Chimney air	Drying air	Chimney air	Drying air
10	2.00	0.00	955	3520
30	0.00	0.00	832	2970
50	0.00	0.00	750	2842
60	0.00	0.00	732	2112

Table 17: proportion of flue gasses composition in the chimney and drying air for mixed-heating operation

Operation time (minute)	Proportion CO (PPM)		Proportion of CO_2 (PPM)	
	Chimney air	Drying air	Chimney air	Drying air
10	0.00	385	924	1851
30	0.00	153	892	1191
50	0.00	148	724	956
60	0.00	109	702	845

Thus, the concentration of CO produced at the chimney (2 PPM) during indirect-heating system was below the recommended limit (≥ 9 PPM) reported by U.S. EPA (2013) and IEPO (2012), (≥ 35 PPM) reported by NIOSH (2012), and (≥ 50) reported by OSHA (2012). However, the concentration of CO that is mixed with the drying air during mixed-heating system was greatly higher than the recommended limits.

Also, the concentrations of CO_2 obtained at the chimney and drying air during indirect and mixed-heating configurations of the furnace was greatly lower than the recommended limit (20,000 PPM) of CO_2 in environment (OSHA, 2013).

4.7. Modeling and simulation results

To optimize the efficiency of the furnace, using Extend simulation software, simulation models was developed from the quadratic models obtained in the analysis of experimental data. The normal method of operation without control and with control of the opening time for mixing ambient air inlet and regulation of drying air temperature during working hours without refueling (1.98 h or 119 min) for optimum efficiency was considered.

Six (6) different scenarios were obtained at different drying temperatures to determine the normal and controlled efficiency of the furnace and the opening time of the mixing chamber inlet for a controlled situation. The first scenario was obtained for indirect-heating configuration for five (5) different drying air temperatures of 50, 60, 70, and 80 ^{0}C respectively. In addition, scenarios (II to VI) were obtained for mixed-fired operation at five different times of 5, 10, 15, 20, and 30 minutes of opening of the valve for the conversion of the configuration of the furnace from indirect to mixed-heating considering the same drying air temperatures.

The simulation results for the average thermal efficiency of the furnace for the normal and controlled conditions at different and desired temperatures are shown in Table 18. However, the simulated results for the average efficiency and percentage of operation time of the furnace in the desired temperature for the controlled system at drying air flow rate of 43 m^3 min^{-1} is shown in Table 19.

Table 18: Simulated values for cycle time of 119 minutes (1.98 h)

Scenarios	Drying air temperature	Without control				With control					
		Mass flow rate (kg. m^{-1})			Efficiency	Mass flow rate (kg. m^{-1})			Efficiency	Time to desired temperature	Efficiency after desired temperature
	(^{0}C)	Heat Exchanger	Ambient	Drying air	(%)	Heat Exchanger	Ambient	Drying air	(%)	(min)	(%)
I – Indirect fired (119 min.)	50	17.23	21.69	34.45	33.45	10.17	33.53	43.70	31.68	22.17	32.80
	60	17.23	21.69	34.45	33.45	13.23	27.44	40.67	40.58	33.74	43.97
	70	17.23	21.69	34.45	33.45	15.81	22.03	37.84	47.83	47.23	54.80
	80	17.23	21.69	34.45	33.45	18.08	17.34	35.42	53.23	67.48	64.95
II - Mix-fired (5 min) Indirect-fired (114 min)	50	17.23	21.69	34.45	35.58	9.24	30.47	39.71	32.66	0.96	32.67
	60	17.23	21.69	34.45	35.58	12.70	26.35	39.05	42.05	1.93	44.79
	70	17.23	21.69	34.45	35.56	15.72	21.89	37.62	49.51	4.82	55.07
	80	17.23	21.69	34.45	35.56	18.72	17.95	36.67	54.94	67.48	66.09
III- Mix-fired (10 min) Indirect-fired (109 min)	50	17.23	21.69	34.45	38.24	9.96	32.84	42.80	33.35	0.96	32.50
	60	17.23	21.69	34.45	38.26	12.88	26.71	39.59	43.21	1.93	44.43
	70	17.23	21.69	34.45	38.24	15.98	22.26	38.24	51.10	4.82	55.61
	80	17.23	21.69	34.45	38.23	18.61	17.85	36.46	56.89	7.71	64.86
IV - Mix-fired (15 min) Indirect-fired (104 min)	50	17.23	21.69	34.45	41.45	10.25	33.79	44.03	33.80	0.96	32.52
	60	17.23	21.69	34.45	41.44	12.54	26.02	38.56	44.09	1.93	44.78
	70	17.23	21.69	34.45	41.44	15.85	22.07	37.92	52.44	4.82	54.78
	80	17.23	21.69	34.45	41.45	19.37	18.57	37.94	58.66	7.71	65.37
V - Mix-fired (20 min) Indirect-fired (99 min)	50	17.23	21.69	34.45	45.09	9.35	30.81	40.15	33.98	0.96	32.61
	60	17.23	21.69	34.45	45.11	12.74	26.41	39.15	44.75	1.93	43.84
	70	17.23	21.69	34.45	45.11	15.53	21.63	37.15	53.54	4.82	55.63
	80	17.23	21.69	34.45	45.09	18.34	17.58	35.92	60.15	7.71	65.22
VI - Mix-fired (30 min) Indirect-fired (89 min)	50	17.23	21.69	34.45	54.61	9.68	31.91	41.59	33.99	0.96	32.43
	60	17.23	21.69	34.45	54.62	12.87	26.68	39.54	45.41	1.93	43.61
	70	17.23	21.69	34.45	54.60	16.64	23.17	39.81	55.18	4.82	55.35
	80	17.23	21.69	34.45	54.62	20.22	19.39	39.61	62.72	7.71	65.23

Table 19: Simulated results for percentage of operation time of the furnace in the desired temperature for the controlled system at drying air flow rate of 43 m^3 min^{-1}

Scenarios	Desired drying air temperature (^0C)	Efficiency (with control) (%)	% Operating time in the desired temperature
I – Indirect fired (119 min.)	50	32.8	81.4
	60	44.0	71.6
	70	54.8	60.3
	80	65.0	43.3
II - Mix-fired (5 min) Indirect-fired (114 min)	50	32.8	99.2
	60	44.0	98.4
	70	55.0	95.9
	80	65.0	43.3
III - Mix-fired (10 min) Indirect-fired (109 min)	50	32.5	99.2
	60	44.4	98.4
	70	55.6	95.9
	80	64.9	93.5
IV - Mix-fired (15 min) Indirect-fired (104 min)	50	32.5	99.2
	60	44.8	98.4
	70	54.8	95.9
	80	65.4	93.5
V - Mix-fired (20 min) Indirect-fired (99 min)	50	32.6	99.2
	60	43.8	98.4
	70	55.6	95.9
	80	65.2	93.5
VI - Mix-fired (30 min) Indirect-fired (89 min)	50	32.6	99.2
	60	43.8	98.4
	70	55.6	95.9
	80	65.2	93.5

The efficiency of the furnace after attaining a desired drying air temperature at different furnace configurations (scenarios) is as presented on Figure 17.

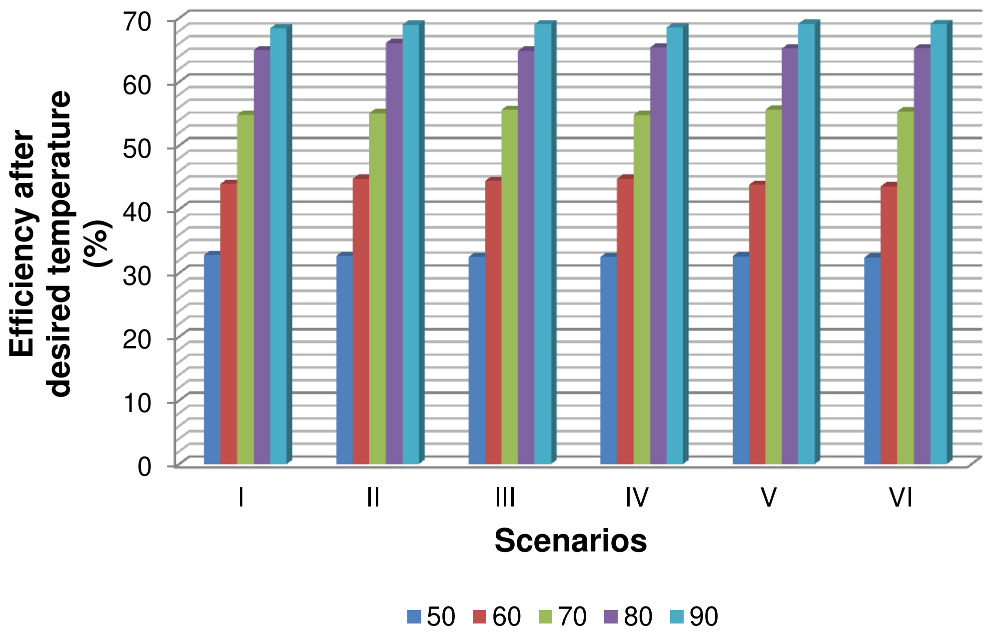


Figure 17: Efficiency of the furnace at desired drying air temperature.

It can be deduced from the figure that the efficiency of the furnace increases with increase drying air temperature. Also, the efficiencies are approximately equal at the same drying air temperature irrespective of the configurations of the furnace. The optimum efficiency of the furnace of 64 % can be attained at a temperature of 80 ^{0}C using any of the configurations.

Therefore, from Tables 19, Scenario II with mixed heating of 5 min and indirect heating of 114 min is considered the best scenario for drying air temperatures of 50 °C (efficiency 32% - during 99.2 % of operating time), 60 °C (efficiency 44% - during 98.4 % of operating time) and 70 °C (efficiency 55% - during 95.5 % of operating time). However, for drying air temperature of 80 °C, scenario III with mixed heating of 10 min and indirect heating of 109 min is considered the best because it yielded the furnace thermal efficiency of 64 % during 93.5 % of operating time.

Also, to improve the thermal efficiency of the furnaces at 50 ^{0}C drying air temperature, during indirect-heating operation, the volume of combustion chamber, surface area of the heat exchanger and the area of the ambient air inlet must be reduced. This will increase the thermal energy generated at the combustion chamber, increase the heat transfer from the heat exchanger to the mixing chamber.

For mixed-heating operation, the thermal efficiency of the furnaces at 50 ^{0}C drying air temperature and its quality can be improved by creating another mixing chamber (inform of cyclone) for direct-fired air from combustion chamber for proper mixing with the ambient air before flowing into the drying air mixing chamber. It is also necessary to increase the ambient air flow rate (considering the change of current fan).

5. CONCLUSIONS

A smokeless furnace was designed, builds and evaluated. The efficiency of the furnace was optimized, using Extend simulation software, developed from the quadratic models obtained in the analysis of experimental data. The normal method of operation without control of drying air temperature and with control of the opening time for ambient air inlet at the mixing chamber when the drying air temperature has attained a desired value during working hours without refueling (1.98 h or 119 min).

According to the results of the evaluation carried out, it was possible to conclude:

- Due to the low temperature in the combustion cell in the first 10 minutes of the operation, when the ignition temperature has not been achieved, the fuel combustion was incomplete, thus cause the flue gasses from the chimney outlet to contain some little proportion of CO, with clean drying air when the furnace operates in indirect configuration. In the case of mixed-fired configuration, the larger proportion of the CO produced mixed with the drying air throughout the duration then furnace operates in the configuration.
- The furnace's efficiency was approximately uniform irrespective of the drying temperature when operated normally. But, the efficiency increases with an increase in drying temperature with a controlled system.

According to the simulation results, the Scenario II with mixed heating of 5 min and indirect heating of 114 min is considered the best scenario for drying air temperatures of 50 °C (efficiency 32% - during 99.2 % of operating time), 60 °C (efficiency 44% - during 98.4 % of operating time) and 70 °C (efficiency 55% - during 95.5 % of operating time). However, for drying air temperature of 80 °C, scenario III with mixed heating of 10 min and indirect heating of 109 min is considered the best because it yielded the furnace thermal efficiency of 64 % during 93.5 % of operating time.

SUGGESTIONS

To optimize the potential of the wood furnace in the grain drying and its application in other air heating systems, the following suggestions are presented for future works:

- Reduction in the size of the furnace, that is, the volume of combustion chamber, surface area of heat exchanger and area of ambient air inlet at the mixing chamber;
- Automation of the control of ambient air inlet at the mixing chamber;
- Evaluation of the system with indirect and mixed heating of the air in the drying grains, and the quality analysis of the dried dried and cost of drying;
- Modification of the combustion chamber to a single grate area with provision for automated feeding mechanism for fuel;
- Checking the influence of factors such as specific mass, sizes and moisture content of the wood used on the performance of the furnace;
- Application of the system with indirect and mixed heating of the air for drying other products and carryout the quality analysis and cost of the drying of the products;
- Evaluation of the furnace performance with other solid fuel (Biomass) such charcoal and some other agricultural residues;

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