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ORIGINAL RESEARCH ARTICLE

COMPUTATIONAL FLUID DYNAMICS (CFD) ANALYSES OF ENERGY AND EXERGY IN THIN LAYER DRYING OF OKRA (*ABELMOSCHUS ESCULENTUS*) SLICES USING CENTRE SHAFT ROTARY TRAY CABINET (CSRTC) DRYER

S. K. Oyeniyi[1*], O. S. Olatunbosun[1], A. K. Aremu[1], N. A. Aviara[2], and I. J. Iyilade[3]
([1]Department of Agricultural and Environmental Engineering, Faculty of Technology, University of Ibadan, Ibadan, Nigeria.*
[2]Department of Agricultural and Environmental Resources Engineering, Faculty of Engineering, University of Maiduguri, Maiduguri, Borno State, Nigeria
[3]Department of Agricultural and Bio - Environmental Engineering, The Oke-Ogun Polytechnic, Saki, Oyo State, Nigeria)
** Corresponding author's email address: oyeniyisk@gmail.com*

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ABSTRACT

This paper presents a simulation of the drying process of okra (*Abelmoschus esculentus*) in a Center Shaft (CS) Rotary Tray Cabinet Dryer using three drying temperatures (50, 60 and 70 °C). ANSYS 14.5 Workbench was used to simulate the dryer model in 2D (2 Dimensional). The detail of the CFD simulation was utilized to investigate the energy and exergy of the dryer. The ANSYS Design Modeler was used to model the 2D representation of the dryer and the meshing was done using ANSYS ICEM. ANSYS Fluent CFD solver was then used to calculate the alternative using the normal turbulence-realizable k-epsilon model in a steady-state system with improved wall temperature treatment. The simulation outcome was used in calculating the dryer's exergy and energy analysis based on the thermal efficiency. It was noted that the simulated temperature from the experiment is greater than that of the experiment. The results indicated that the experimental energy utilization (EU), energy utilization ratio (EUR) and energy efficiency increased from 14.1 to 57.93 J/s, 0.15 to 0.20 and 18.89 to 33.98 percent, while the simulated energy utilization ratio increased from 23.91 to 57.68 J/s, 0.19 to 0.20 and 26.21 to 33.40 percent, respectively, and as the drying air temperature increased from 50 °C to 70 °C. Experimental exergy inflow, outflow, exergy loss and exergy efficiency increased from 4.01 J/s to 6.98 J/s, 1.83 J/s to 1.9 J/s, 3.18 J/s to 5.07 J/s and 21 to 27%, while simulated air temperatures increased from 5.01 J/s to 7.49 J/s, 1.33 J/s to 2.20 J/s, 3.66 J/s to 5.29 J/s and 27 to 29% respectively with respect to the drying air temperature range (50–70 °C). Model equations were derived from the plotted graphs to express the energy and exergy parameters as a function of drying temperature.

1.0 Introduction

Drying is an industrial preservation method widely used in which water activity of food is decreased to minimize biochemical reactions of degradation. In order to improve the control of this unit operation, it is important to use accurate models to simulate the drying operation most especially the air flow and temperature distribution in the dryer as this is responsible for the uneven drying of the products which is one of the most reoccurring situation in a static convective dryers. Odewole and Oyeniyi, (2016) reported that drying is one of the most important processes in food processing and preservation. It is also defined generally as the removal of moisture from a material to a predetermined level by different authors (Mujumdar, 2007; Kumar et al., 2012; Aviara, et al., 2014 and Odewole and Olaniyan, 2015). The use of computer programmes to predict the behaviours of drying process especially the thermal profile of drying air in the drying chamber has been advanced recently due to the advances in the development of computer software like Computational Fluid Dynamics software that has the great capacity in evaluating the drying parameters with high efficiency which gives high impact to the cost of energy being used during the operation. Amanlou and Zomorodian, (2010) wrote that evaluating drying experimentally is costly but computational work using computer programs and models are more important to use due to lower cost and acceptable accuracy with minimum error. It has been adjudged that Computational Fluid Dynamics (CFD) software reduces a lot of trial and error on experimental work because of its ability to give detailed account of the stringent parameters that cannot be observed analytically since it is capable of presenting visualized results. The advancement in the trust of CFD simulation has been enhanced by the improvement in computer codes in solving Partial Differential Equations (PDE) equations

Many dryer types have been reported to be in existence and several works have been done to improve their efficiencies but in terms of dryer simulation there are few reports on Centre Shaft (CS) rotary tray cabinet dryer in order to predict the thermal profile of the dryer system and the thermal analysis of the energy and exergy accounting of the dryer. Data obtained from the CFD analysis were used to achieve this objective.

1.2 Thermodynamics Analysis

The significance of thermodynamic principles is very important in performing the energy and exergy analyzes of the drying procedures. Reason being that the fundamental rules of thermodynamics (i.e. first and second laws) govern the accounting of the analyzes. The first law is the foundation of the heat-balance technique of assessment frequently used in the performance analysis of the engineering system and the second law includes the reversibility or irreversibility of procedures and is a very significant element in the assessment of the energy system's exergy technique (i.e. the quality of the energy available in the scheme). Dincer and Sahin (2004) wrote that thermodynamic analyzes, especially exergy analyzes, seemed to be an important instrument for designing, analyzing and optimizing thermal systems. Ozegerner and Ozegernar (2006) wrote that the available and unavailable forms consist of energy from the thermodynamic point of view. Work performed by a system is obtained from the available energy while the unavailable form of energy remains unexploited. Whereas exergy refers to the accessible type of energy of the system that is convertible to full helpful job and strikes a balance with its setting from its initial state (Coskun et al., 2009; Hou et al., 2007).

Corresponding author’s e-mail address: oyeniyisk@gmail.com

1.3 Energy and Exergy Analysis

The energy quality available for the drying method has been referred to as exergy, which is clarified by the second thermodynamic legislation (Chemmala and Dinesh, 2014). Measuring these parameters has put CFD in a pinnacle in such a manner that it has been used in various ways to forecast fluid flow behavior and also to analyze the multiphysics governing such a complicated scenario.

Exergy is said to be the highest quantity of work obtainable from a stream of matter when some matter is carried to a state of thermodynamic equilibrium with the prevalent element of the natural environment through reversible processes and is a measure of the ability of a stream to cause change as a result of not being entirely stable in relation to the reference setting (Dincer, 2002; Pandey et al., 2012; Prommas et al., 2010).

The thermodynamic analysis-energy and exergy of a thermal system is very essential to technicians in order to optimize system efficiency, minimize losses, decrease operating and capital investment expenses, and improve heat system productivity (Riviere et al., 2009). The optimal thermodynamic efficiency of a system is the proportion of helpful job to the quantity of energy provided to the system.

Drying process and dryer design have been advanced in a variety of ways based on the requirements for high-quality dried products, but the energy used during the drying process, especially for cabinet dryers, has not been spelt out in terms of the energy and exergy analyzes that were the focus of this present study. A number of research on this cogent element in dryer design have been performed on solar dryers, but few on this form of dryer. The aim of this research is therefore to conduct energy and exergy analyzes of the cabinet dryer using the information obtained during the CFD simulation and to input three inlet velocities in order to assess the impact of air velocity on the effectiveness of the CSRTC dryer.

2.0 Materials and Methods

2.1 Description of the CSRTC Dryer

The dryer used for the drying procedure is shown in Figure 1. It is basically split into three parts, the heat producing portion, the heat exchanger, and the drying chamber. The heat producing segment consists of a blower (axial cross flow fan) and a 1.8 kW red hot heating element. Here, the blower vanes turns the flow centrifugally outwards through the blades to the pipes in the heat exchanger conveying the hot air to the drying chamber. The incorporation of the pipes is to control direct heating of the products being dried in the drying chamber. The third segment is the drying chamber separated into three compartments with a center rotating shaft that assists the uniformity of warm air in the drying chamber. Three trays were stacked vertically here.

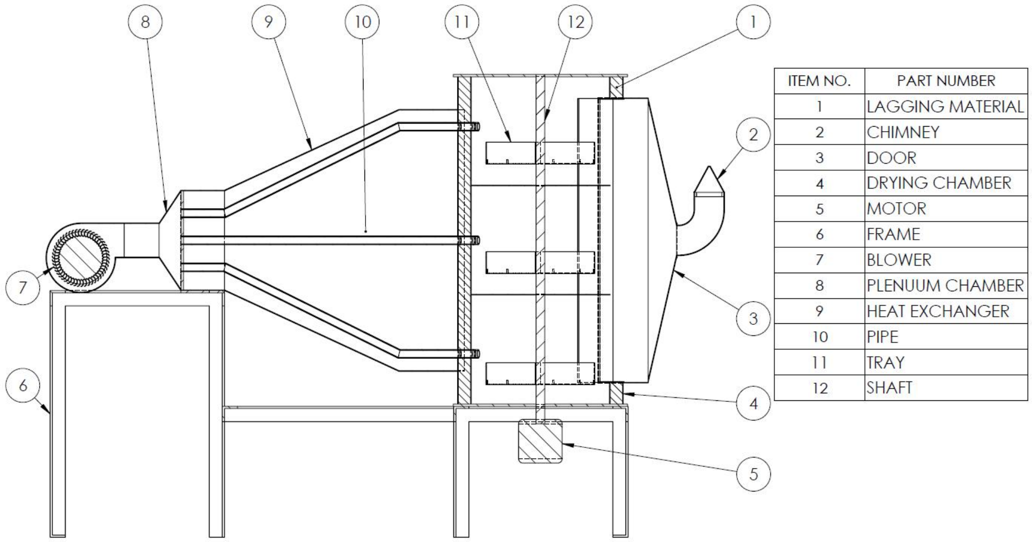

Figure 1: Schematic diagram and part list of the CSRTC tray dryer.

2.1 CFD Simulation Setup

The 2-dimensional assessment of the flow pattern in the dryer was performed in ANSYS FLUENT solver as follows: the dryer's geometry was modeled in ANSYS Design Modeler to portray the dryer in 2-dimensions ; mesh analysis was performed in ANSYS ICEM. ANSYS FLUENT CFD solver was used because it is the easiest solution to analyze CFD simulation. (Murathathunyaluk, et al., 2015). The fluid flow within the dryer was defined by using the commercial CFD code ANSYS Fluent to solve iteratively using a 2D solver with a steady state condition, mass conservation, momentum equations and energy equations. The code utilizes a pressure-based solver for velocity-pressure coupling using the SIMPLE technique. The relaxation variables were 0.3 for stress, 0.7 for momentum, 1 for density, 1 for body strength and 0.8 for turbulent kinetic energy. Gauss Seidel's smoother form was used in sophisticated solution control. The velocity and temperature fields were discretized with a second order upwind system, while the pressure field was discretized with a PRESTO (Pressure Discretization Schemes) scheme. The convergence criteria for continuity and momentum equations residuals were 10-4 and 10-6 for standard model power and radiation equations with improved wall temperature therapy. The initialization of all the boundary conditions was done in order for the software to solve the numerical equations. The amount of iterations and the reporting rates were set at 600 and 10, respectively, in which the outcome converged at the 470th iteration. The airflow distribution and heat transfer inside the cabinet tray dryer were then plotted with contour lines to better represent the flow patterns. For the exergy assessment, the inlet and outlet temperatures calculated from the ANSYS FLUENT solver were used.

In this research, a thorough thermodynamic inquiry is performed through energy and exergy analyzes to evaluate the performance of a center shaft rotary tray cabinet tray dryer during the drying phase of okra and to study how its working conditions and effectiveness can be further enhanced by varying the drying air temperature. Some of the data used in the calculations were taken from computer-generated results during the CFD simulation conducted with ANSYS FLUENT 14.5 by varying the drying air temperature in order to converse materials and energy needed to run the experiments in replicates.

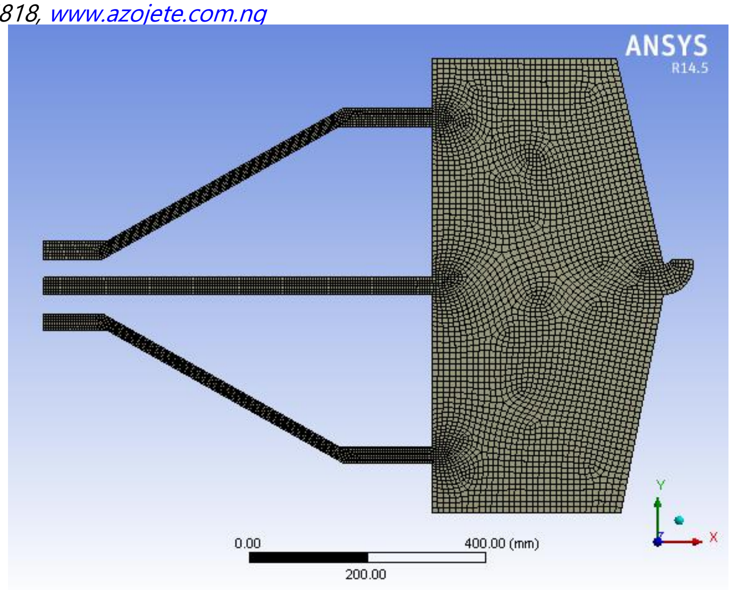

Figure 2: Discretization of the 2D model of the CSRTC Dryer

2.2 Energy usage by the dryer

The following equations (1-3) govern the detailed account of the drying air behavior and the energy change with respect to the components in the dryer. It was presumed that the stream is steady-state with regard to the first thermodynamics law. The three equations describing Navier-Stokes equations are as follows (Dincer, 2002):

The conservation of mass for dry air

$$\sum \dot{m}_{hi} = \sum \dot{m}_{ho} \tag{1}$$

The conservation of mass for humidity is given by:

$$\sum (\dot{m}_{hi} + \dot{m}_{hi}) = \sum \dot{m}_{ho} \tag{2}$$

The conservation of energy

$$\dot{Q} - \dot{W} = \sum \dot{m}_o \left(h_0 + \frac{{V_o}^2}{2}\right) - \sum \dot{m}_i \left(h_i + \frac{{V_i}^2}{2}\right) \tag{3}$$

Where

$\dot{m}_i$ and $\dot{m}_o$ are the mass flow rate at the inlet and outlet respectively (kg/s)

Q - is heat energy inflow (kJ/s),

W - is rate of mechanical work output in (J/s),

h_i and h_O are the enthalpies of air at the dryer inlet and outlet temperature (J/kg)

V_i and V_O are air velocities at dryer inlet and outlet respectively (m/s)

Since there is no resultant motion in the drying process, the momentum components $\frac{{V_o}^2}{2}$ and $\frac{{V_i}^2}{2}$ were eliminated and the equation is reduced to (4)

$$\dot{Q} = \sum \dot{m}_o h_0 - \sum \dot{m}_i h_i \tag{4}$$

Considering the mass flow rate of the air to be uniform (i.e. $\dot{m}_a \doteq m_i = \dot{m}_o$), equation (4) is reduced to

$$\dot{Q} = \dot{m}_a(h_0 - h_i) \tag{5}$$

Corresponding author's e-mail address: oyeniyisk@gmail.com

2.3 Thermodynamic parameters

Wet air is considered a one-phase homogeneous system with only two components governed by the ideal gas law for fluid mixtures.

2.3.1 *Relative humidity*

It is defined as the ratio between the partial vapour pressure of water in the mixture at a given temperature (Pv,T), and the saturated vapour pressure at the same temperature (Psat,T):

$$\emptyset = \frac{P_{v@T}}{P_{sat@T}} \times 100\ \% \tag{6}$$

$\emptyset$ is the relative humidity (%), $P_{v@T}$ vapour pressure at time, T (Pa) and $P_{sat@T}$ saturated vapour pressure at time, T (Pa)

2.3.2 *Specific humidity*

This is defined as the water vapour mass per drying air unit mass

$$w = \frac{m_v}{m_a} = 0.622\frac{P_{v@T}}{P - P_{v@T}} \tag{7}$$

w is the specific humidity (kg/kg Dry air), m_v mass of vapour (kg/s), m_a mass of air (kg/s), $P_{v@T}$ vapour pressure at temperature, T (Pa) and P is the total pressure (Pa)

2.3.3 *Enthalpy (of the drying air)*

$$h_{da} = c_{pda}T_{da} + wh_{sat@T} \tag{8}$$

h_{da} is the enthalpy of the drying air (kJ/kg), c_{pda} is the specific heat of drying air (kJ/kg.K), T_{da} is the drying air temperature (K), w is the specific humidity (kg/kg Dry air) and $h_{sat@T}$ is the enthalpy of the saturated vapour (kJ/kg).

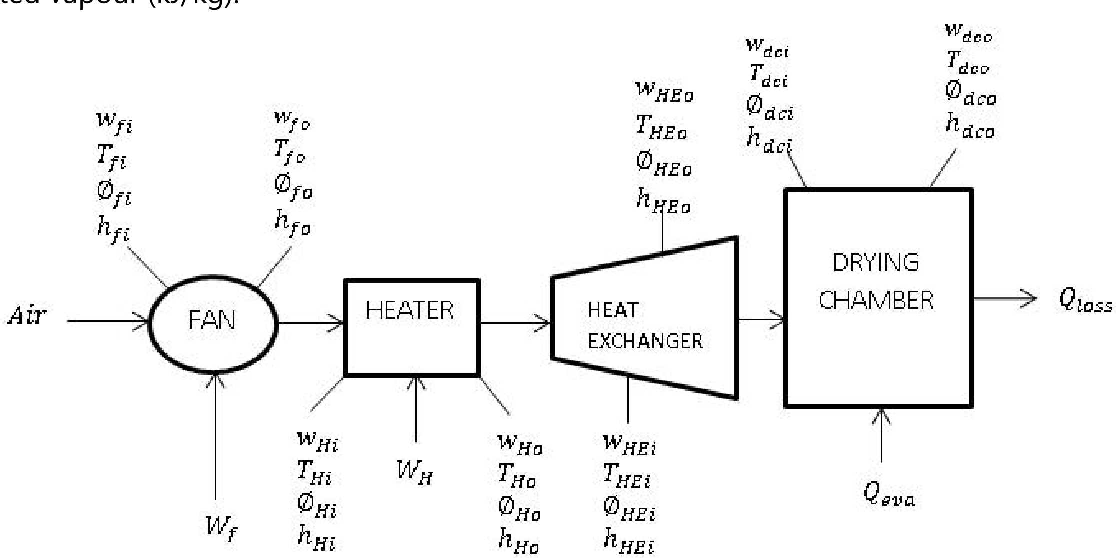

Figure 3: The mass-energy model for the drying experiment in respect to the components

Where:

w = specific humidity (kg/kg Dry air)

T = air temperature (K)

$\emptyset$ = relative humidity (%)

h = enthalpy of air (kJ/kg)

fi and fo are the air conditions at the fan inlet and outlet

Hi and Ho are the air conditions at the inlet and outlet of the heater

HEi and HEo are the air conditions at the inlet and outlet of the heat exchanger

dci and dco are the air conditions at the inlet and outlet of the drying chamber

Q_{eva} = thermal power obtained from evaporation (kJ/s)

Q_{loss} = thermal power loss (kJ/s)

2.3.4 Determination of fan outlet conditions

$$\dot{Q} - \dot{W}_f = \sum \dot{m}_{da}\left[(h_{fo} - h_{fi}) + \left(\frac{{V_{fo}}^2 - {V_{fi}}^2}{2 \times 1000}\right)\right] \tag{9}$$

$\dot{Q} = 0$ since there is no heat transfer and also since we are considering only the outlet conditions, $V_{fi} = 0$

Thus:

$$h_{fo} = \left[\left(\dot{W}_f - \frac{{V_{fo}}^2}{2 \times 1000}\right)\left(\frac{1}{\dot{m}_{da}}\right)\right] + h_{fi} \tag{10}$$

Where:

h_{fi} and h_{fo} are the enthalpies of air at the inlet and outlet of the fan (kJ/kg)

V_{fi} and V_{fo} are the air velocities at the inlet and outlet of the fan (m/s)

$\dot{W}_f$ is the power of the fan (kJ/s)

$\dot{m}_{da}$ is the mass flow rate of dry air (kg/s)

2.3.5 Determination of heater inlet and outlet conditions

$$Q_{usable} = \dot{m}_{da} c_{pda}(T_{Hi} - T_{Ho}) \tag{11}$$

2.3.6 Determination of Energy Utilization (EU)

EU (Energy utilization) was determined by applying the first law of thermodynamics as expressed by Equation (5) and transformed in Equation (12)

$$EU = \dot{m}_a(h_0 - h_i) \tag{12}$$

2.3.7 Determination of Energy Utilization Ratio (EUR)

The energy usage ratio during the drying process was found from the equation given by (Akpinar, 2004)

$$EUR_{dc} = \frac{\dot{m}_a(h_{dci\,@\,T} - h_{dco\,@\,T})}{\dot{m}_a(h_{dci\,@\,T} - h_{a\infty})} \tag{13}$$

Where:

EUR_{dc} = the Energy Usage Ratio for the drying chamber

m_{da} = mass of dry air

$h_{dci\,@\,T}$ = enthalpy of air at the inlet of the drying chamber at temperature, T (kJ/kg)

$h_{dco\,@\,T}$ = enthalpy of air at the outlet of the drying chamber at temperature, T (kJ/kg)

$h_{a\infty}$ = enthalpy of air at ambient temperature, T (kJ/kg)

2.3.8 Energy Efficiency

This was evaluated as the ratio of the energy used and the input energy

$$\eta_E = \frac{E_i - E_o}{E_i} = \frac{\dot{m}_a(h_{dci\,@\,T} - h_{dco\,@\,T})}{\dot{m}_a h_{dci\,@\,T}} \times 100\ \% \tag{14}$$

Where η_E is the energy efficiency in %, E_i and E_o are the input and output energy respectively in kJ/s

2.4 Exergy analysis

The helpful notion of exergy in the assessment of thermal systems is implemented by the second law of thermodynamics which is a measure of the energy quality in the thermal system. Total exergy of in stream, outflow and drying chamber losses were estimated in the scope of the second law assessment of thermodynamics. The fundamental method for chamber exergy assessment is to determine the exergy values at steady-state points and the reason for the process's exergy variation. The exergy values are calculated using the features of the working medium from the energy equilibrium of the first law. For this purpose, the mathematical formulations used to perform the exergy balance are as shown below as shown by (Ahern, 1980).

$$\begin{aligned}\text{Exergy} = &\underbrace{(u - u_\infty)}_{\text{I}} - \underbrace{T_\infty(S - S_\infty)}_{\text{II}} + \underbrace{\frac{P_\infty}{J}(v - v_\infty)}_{\text{III}} + \underbrace{\frac{V^2}{2gJ}}_{\text{IV}} + \underbrace{(z - z_\infty)\frac{g}{g_c J}}_{\text{V}} \\ &+ \underbrace{E_c(\mu_c - \mu_\infty)N_c + E_i A_i F_i\left(3T^4 - {T_\infty}^4 - 4T_\infty T^3\right) + \ldots}_{\text{VI}}\end{aligned} \tag{15}$$

i – internal energy

ii – entropy

iii – work

iv – momentum

v – gravity

vi – chemical radiation emission

u - specific internal energy, (kJ/kg)

T - temperature, (K)

s - specific entropy, (kJ/kg K)

P - pressure, (kPa)

J - joule constant

v - specific volume, (m3/s)

V - velocity, (m/s)

g - gravitational acceleration, m/s^2

z - altitude coordinate, (m)

gc - constant in Newton's law

h - enthalpy, (kJ/kg)

μ_c – kinematic viscosity(m^2/s)

N - number of species

The subscript indicates the terms of reference. Only some of the conditions shown in Equation 15 are used in the exergy analyzes of many structures, but not all. Since exergy is energy accessible from any source, it can be created using materials ' electrical present flow, magnetic fields, and diffusion flow. One popular simplification is to replace enthalpy with internal energy and PV conditions relevant to steady-flow systems. Equation 15 is often used under circumstances where the terms of gravity and momentum are ignored. In addition, the stress changes in the scheme are also overlooked because of $v \cong v_{\infty}$,

The general exergy equation is being derived from the above equation as follows:

$$\dot{E}x = \dot{m}_{da} c_{pda} \left[(T - T_{\infty}) - T_{\infty} \ \mathrm{In} \ \frac{T}{T_{\infty}} \right] \tag{16}$$

The inlet and outlet exergies were determined according to the drying chamber inlet and outlet temperatures.

$$Ex_{dci} = c_{pda} \left[(T_{dci} - T_a) - T_a \ln \frac{T_{dci}}{T_a} \right] \tag{17}$$

$$Ex_{dco} = c_{pda} \left[(T_{dco} - T_a) - T_a \ln \frac{T_{dco}}{T_a} \right] \tag{18}$$

Where:

Ex_{dci} and Ex_{dco} are the exergy at the inlet and outlet of the drying chamber respectively (kJ/s)

2.4.1 Exergy efficiency

Akbulut and Durmus, (2010) explained exergy efficiency as the ratio of the exergy use in the drying of the product exergy to energy inflow from the drying chamber. Midilli and Kukuk, (2003) gave the general form of the exergetic efficiency as follows:

$$Ex_{loss} = Ex_{dci} - Ex_{dco} \tag{19}$$

$$\eta_{Ex} = \frac{Ex_{dco}}{Ex_{dci}} \tag{20}$$

Where:

Ex_{dci} and Ex_{dco} are the exergy at the drying chamber inlet and outlet respectively

c_{pa} is the specific heat of the air

T_{dci} and T_{dco} are the temperature at the air inlet and outlet of the drying chamber respectively

T_a is the temperature of the environment

η_{Ex} is the exergy loss

2.4.2 Exergetic Improvement Potential

The equation given by Hammond and Stapleton, (2001) was used to determine the exergetic improvement potential of the drying process. This is expressed as:

$$\dot{I}P = (1 - \eta_{Ex})(\dot{E}x_I - \dot{E}x_0) \tag{21}$$

3.0 Results and Discussions

Energy Utilization (EU) was varied with the drying air temperature during the simulation process of okra (Abelmoschus esculentus) drying and the relationship is presented in Table 1.

Table 1: Temperature and Enthalpies of air obtained from the CFD simulation

Drying temp. (°C)	$T_{dci@T}$		$T_{dco@T}$		$h_{dci@T}$		$h_{dci@T}$		Energy Utilization		Energy Utilization Ratio		Energetic Efficiency	
	EXP	SIM	EXP	SIM	EXP	SIM	EXP	SIM	EXP	SIM	EXP	SIM	EXP	SIM
50.00	322.38	328.56	294.82	301.00	152.34	186.12	123.56	137.33	14.10	23.91	0.15	0.19	18.89	26.21
55.00	328.04	331.51	299.04	302.51	219.00	218.96	150.73	159.21	33.45	29.28	0.21	0.19	31.17	27.29
60.00	332.06	332.45	303.64	304.03	259.67	251.79	187.93	181.08	35.15	34.65	0.18	0.18	27.63	28.08
65.00	337.35	337.15	306.73	306.53	304.11	302.60	208.43	207.89	46.88	46.41	0.19	0.19	31.46	31.30
70.00	339.27	341.85	306.44	309.02	347.89	352.41	229.66	234.70	57.93	57.68	0.20	0.20	33.98	33.40

3.1 Energy Utilization

It can be said that the energy utilization increased from 14.10 to 57.93 J/s for the experimental data while 23.91 to 57.68 J/s for the simulation as the temperature of the drying air increased from 50 to 70°C this trend is in good agreement with Aviara et al. (2014) in the drying of cassava starches that energy utilization increased from 1.93 to 5.51 J/s with varied drying air temperature of 40 to 60 °C. Similar have been recorded by Erbay and Icier (2011) on the drying of olive leaves in a tray dryer and Motevali and Minaei (2012) on the drying of sour pomegranate arils in a microwave dryer. Also, Odewole and Oyeniyi (2016) in the drying of green bell pepper in a convective cabinet tray dryer that energy utilization increased with increase in the drying temperature from 50 to 60 °C. The current research shows that the relationship that exist between EU (energy utilization) and drying air temperature is polynomial of the third order and can be represented by the following equation:

$$EU_{SIM} = -0.0003T^3 + 0.1106T^2 - 8.0345T + 189.97, \qquad R^2 = 0.9966 \qquad (22)$$

$$EU_{EXP} = 0.0113T^3 - 2.0547T^2 + 125.47T - 2535.8, \qquad R^2 = 0.9804 \qquad (23)$$

Where, EU is the energy utilization, T is the dry air temperature and R^2 is the coefficient of determination.

3.2 Energy efficiency

The energy efficiency of okra (Abelmoschus esculentus) drying in a Center Shaft rotary tray cabinet (CSRTC) dryer increased from 18.87 to 33.98 percent for the experiment and from 26.21 to 33.40 for the simulation as the drying air temperature increased from 50 ° C to 70 ° C. The energy efficiency of okra drying in a cabinet dryer was discovered to have a third-order connection with drying air temperature and this connection was expressed with the following equation:

$$\eta_{E_{SIM}} = -0.0006T^3 + 0.1126T^2 - 7.108T + 169.52, \qquad R^2 = 0.9845 \qquad (24)$$

$$\eta_{E_{EXP}} = 0.0097T^3 + 1.7763T^2 + 108.44T - 2171.6, \qquad R^2 = 0.8954 \qquad (25)$$

Where, $\eta_{E_{SIM}}$ is the simulated energy efficiency, $\eta_{E_{EXP}}$ is the experimental energy efficiency T is the dry air temperature and R^2 is the coefficient of determination.

3.3 Energy Utilization Ratio

It can be deduced from Table 1 that the energy utilization ratio increased from 0.15 to 0.20 for the experimental data while the simulated EUR increased from 0.19 to 0.20 as the drying air temperature increased from 50 to 70 ° C this trend is advantageous because the energy used is minimal. Current study demonstrates that the connection between EUR (power usage ratio) and

Corresponding author's e-mail address: oyeniyisk@gmail.com

drying air temperature is third-order polynomial and can be represented by the following equation:

$$EUR_{SIM} = -4 \times 10^{-6}T^3 + 0.0009T^2 - 0.0597T + 1.4767, \qquad R^2 = 0.9127 \qquad (26)$$
$$EUR_{EXP} = 6 \times 10^{-5}T^3 - 0.0104T^2 + 0.6265T - 12.359, \qquad R^2 = 0.7161 \qquad (27)$$

3.4 Variation of exergy inflow, exergy outflow and exergy loss with drying air temperature

Figure 4 below demonstrates the variety of exergy inflow, exergy outflow and exergy loss with drying air temperature in okra processing (Abelmoschus esculentus). Experimental exergy inflow, outflow and losses decreased from 4.01 to 6.98, 0.83 to 1.90 and 3.18 to 5.07 J / s respectively as the drying air temperature rose (50-70 ° C), while simulated exergy inflow, outflow and losses increased from 5.01 to 7.49, 1.35 to 2.20 and 3.66 to 5.29 respectively. This outcome gave a nice representation of the drying method since the drying method used energy. In addition, the quality of the energy used which was characterized as exergy improved with the drying temperature; this means that the drying temperature also improved the energy quality required for the drying procedure. Aviara, et al. (2014) reported similar results in drying cassava starch using 40 to 60 ° C air temperature. Akpinar et al. (2005) also noted the same trend in drying coroba slice in a convective type dryer connection between exergy inflow, outflow and exergy loss in drying air. Colak et al. (2008) observed that the loss of exergy increased with an rise in temperature when drying mint leaves using a heat pump dryer.

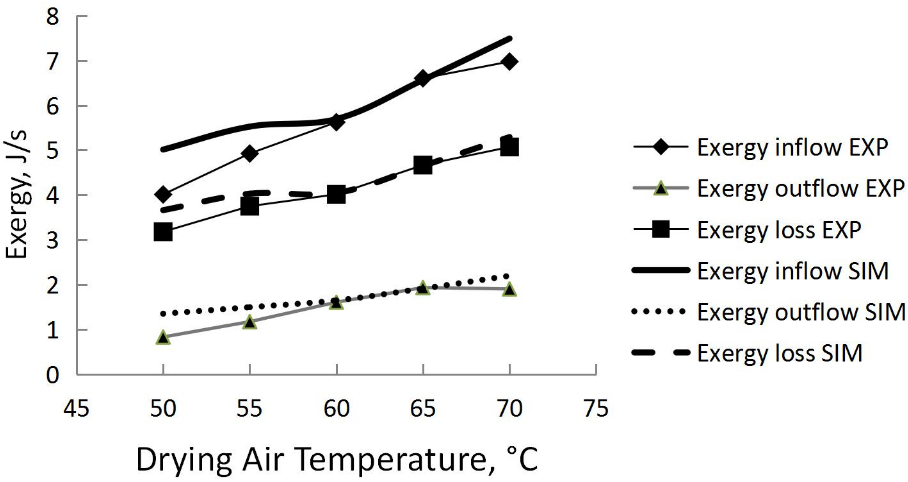

Figure 4: Effect of drying temperature on the exergy inflow, outflow and loss.

The connections between exergy inflow, outflow and exergy loss and drying air temperature were discovered to be second-order polynomial. The connection is demonstrated in the following equations:

$$Ex_{in\_SIM} = 0.0044T^2 - 0.4046T + 14.379, \qquad R^2 = 0.985 \qquad (28)$$
$$Ex_{in\_EXP} = -0.0023T^2 + 0.4262T - 11.609, \qquad R^2 = 0.9927 \qquad (29)$$
$$Ex_{out\_SIM} = 0.0011T^2 - 0.092T + 3.1587, \qquad R^2 = 0.9987 \qquad (30)$$
$$Ex_{out\_EXP} = -0.0024T^2 + 0.3505T - 10.645, \qquad R^2 = 0.9778 \qquad (31)$$
$$Ex_{loss\_SIM} = 0.0033T^2 - 0.3126T + 11.32, \qquad R^2 = 0.9715 \qquad (32)$$
$$Ex_{loss\_EXP} = 0.0002T^2 + 0.0757T - 0.9636, \qquad R^2 = 0.9885 \qquad (33)$$

Where, Ex_{in_EXP}, Ex_{out_EXP} and Ex_{loss_EXP} are the experimental exergy inflow, outflow and exergy loss respectively, Ex_{in_SIM}, Ex_{out_SIM} and Ex_{loss_SIM} are the simulated exergy inflow, outflow and exergy loss respectively T is the dry air temperature and R^2 is the coefficient of determination.

3.5 Variation of exergy inflow, exergy outflow and exergy loss with energy utilization

Figure 5 shows that exergy inflow, outflow and losses varied with energy usage in a way comparable to their drying air temperature variability. Each of them improved with enhanced energy usage and had a linear connection with energy usage. The relationships with the following equations were articulated:

$$Ex_{in\_SIM} = 0.0572EU + 3.9113, \quad R^2 = 0.9214 \quad (34)$$
$$Ex_{in\_EXP} = 0.072EU + 2.9278, \quad R^2 = 0.9479 \quad (35)$$
$$Ex_{out\_SIM} = 0.0374EU + 2.9322, \quad R^2 = 0.9126 \quad (36)$$
$$Ex_{out\_EXP} = 0.0449EU + 2.4503, \quad R^2 = 0.9718 \quad (37)$$
$$Ex_{loss\_SIM} = 0.0198EU + 0.9792, \quad R^2 = 0.9221 \quad (38)$$
$$Ex_{loss\_EXP} = 0.027EU + 0.4775, \quad R^2 = 0.856 \quad (39)$$

Where, EU is the energy utilization

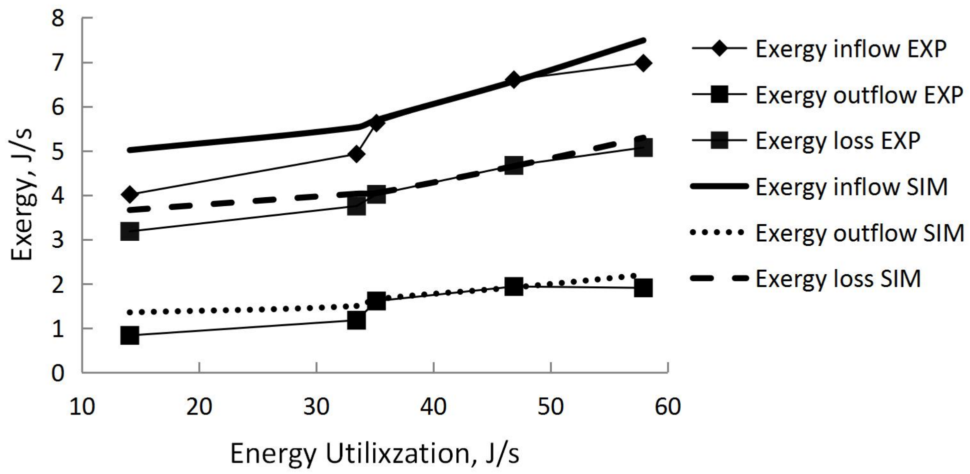

Figure 5: Effect of energy utilization (EU) on the exergy inflow, outflow and loss.

3.6 Relationship between the Exergetic efficiency with drying temperature

Figure 6 demonstrates differences in the center shaft rotary tray cabinet dryer's exertion effectiveness with drying air temperature. Exergetic efficiency increased from 0.21 to increased drying air temperature (50–70 ° C). Similar results were reported on the drying of cassava starches (Aviara et al., 2014), eggplant slices (Akpinar et al., 2005), green olive (Colak and Hepbasli, 2007), mint leaves (Colak et al., 2008), jackfruit leather (Chowdhury et al., 2011) and sour pomegranate arils (Motevali and Minaei, 2012). With regard to the temperature range used, the exergetic effectiveness differs inversely with the conduct of the energy efficiency. The connections between experimental exergetic effectiveness and drying air temperature were discovered to be second-class polynomial while third-class polynomial was discovered for simulated exergetic effectiveness. The connection is demonstrated in the following equations:

$$\eta_{Ex\_SIM} = -1 \times 10^{-5}T^3 + 0.0021T^2 - 0.1187T + 2.5125, \quad R^2 = 0.9333 \quad (40)$$
$$\eta_{Ex\_EXP} = 0.0024T^2 - 0.2409T + 8.7156, \quad R^2 = 0.9498 \quad (41)$$

Where, η_{Ex_SIM}, η_{Ex_EXP} are the simulated and experimental exergetic efficiency, T is the dry air temperature and R^2 is the coefficient of determination

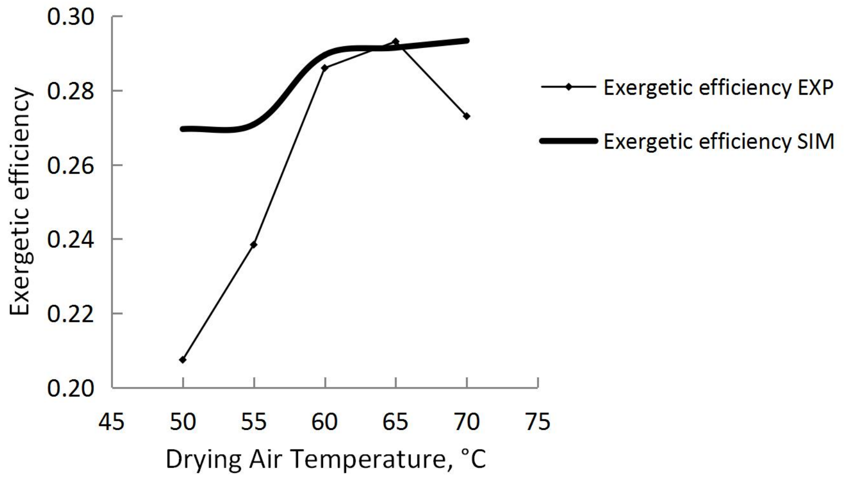

Figure 6: Variation of Exergetic efficiency with drying temperature

3.7 Exergetic Improvement Potential

Figure 7 shows the impact of drying air temperature on the enhancement potential of drying okra (Abelmoschus esculentus) in a center shaft rotary tray cabinet dryer. The figure shows that the potential for exertional enhancement increased with an rise in drying air temperature, which is in excellent agreement with the results obtained from the drying of cassava starches. Aviara et al., (2014). Erbay and Icier, (2011) and Aghbashlo et al., (2013) reported similar trend on the drying of olive leaves and fish oil encapsulation, respectively. The relationship existing between improvement potential and drying air temperature was depicted by the following equation:

$$IP_{SIM} = 0.0024T^2 - 0.2409T + 8.7156, \quad R^2 = 0.9498 \quad (42)$$

$$IP_{EXP} = 0.0015T^2 - 0.1241T + 5.0283, \quad R^2 = 0.9646 \quad (43)$$

Where: IP_{SIM} and IP_{EXP} are the simulated and experimental improvement potential in J/s respectively, T is drying air temperature in °C and R^2 is coefficient of determination.

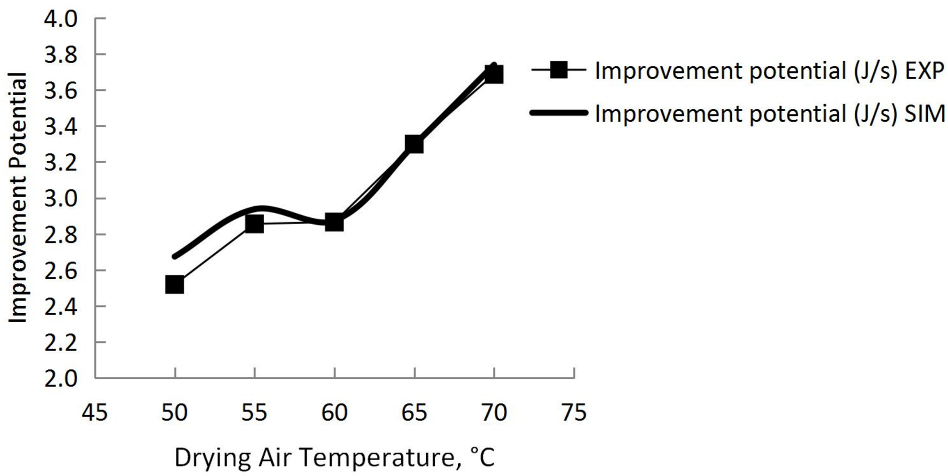

Figure 7: Variation of exergetic improvement potential with drying temperature.

4.0 Conclusion

Computational fluid dynamics simulation using ANSYS 14.5 Fluent CFD Solver was used to explore the energy and exergy analysis of a center shaft rotary tray cabinet dryer.

The relationship between the Energy Utilization (EU) and the drying air temperature is a polynomial with third order.

Second-order polynomial can best be used to define the trend between the energy utilization ratio and the drying air temperature.

Drying air temperature is directly proportional to exergy inflow, exergy outflow and exergy loss since an increase in the drying air temperature brings about increase in the exergy analysis.

The energy utilization has the same trend with the drying air temperature when varied with the exergy inflow, exergy outflow and exergy loss.

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