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Experimental Investigation on the Effects of Wall Chamber Insulation on Nusselt Number and Heat Loss Reduction in a Natural Convection Oven

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ABSTRACT

The main aim of this study was to investigate the effect of oven wall chamber insulation on the Nusselt number and heat loss reduction in a direct-fired natural convection oven. Three sets of experiments, namely uninsulated and insulation with each of the two insulation materials were performed and a data acquisition system was used to continuously monitor temperature and environmental changes from the oven's four sides, ambient, and inside using a heat temperature sensor and a thermocouple. The temperature profiles, the Nusselt number, the Rayleigh number and the heat transfer coefficient were used to assess the heat transfer characteristics of the oven. Within the range of the Raleigh numbers studied, the average Nusselt number is highest for the uninsulated and higher for the insulated with aluminum-reinforced fiber and least for the insulated with fiberglass. Based on the findings, installing thermal insulation materials in oven walls could reduce heat loss, appreciably. The average percentage reductions in heat loss as a result of insulation were 3.8% and 16.7% for the fiber reinforced aluminum and fiberglass, respectively. The least-square correlation of the Nusselt numbers and the Rayleigh numbers gave high values of adjusted R-square values, which are recommended for optimization of natural convective heat transfer for uninsulated and insulated ovens.

1. INTRODUCTION

The oven is a critical energy consumption component in the bread-baking process that demands consideration for energy use reduction [1]. The rising energy cost, environmental and climate change impact due to increasing demand for energy consumption call for innovative solutions to reduce energy utilization in bread baking ovens. Several research in the field have highlighted already the fact that the energy demands of bread-baking ovens generally are high [1, 2]. However, it is reported that quite a sizeable portion of the energy input for bread baking gets lost to the ambient environment. According to reference [3], ovens typically consume five or more times as much energy as it is thermodynamically required to bake products. High percentage of the input energy is lost as heat through the oven walls. It is estimated that about 47% of input energy to a typical oven is absorbed by the oven walls, 25% get lost through the oven walls through convection and radiation and about 15% get lost due to evaporative moisture [4-8] also presented that only 13% of supplied power of oven is used for cooking and 50% of energy lost to heating structure as well as air leakage and ventilation losses.

Heat transfer in ovens generally occurs in all the three modes, namely, convection, conduction and radiation. Muthukumar and Shyamkumar [9] highlighted that heat is conducted through the shells and the other units and then transferred to the surroundings by convection and radiation. However, according to Pask et al. [10], combustion in conventional burners is characterized by free flame, in that situation convection becomes the predominant mode of heat transfer since gases have poor thermal conductivity and low opacity, and therefore the contributions of conduction and radiation modes of heat transfer from the burned to unburned gases are negligible. A practical approach to reducing energy loss in the form of heat lost is by insulating or covering the surface. Thermal energy loss reduction through the use of insulation has been the subject of researchers over several decades [9-12]. According to Sari et al. [13], theoretical, experimental and numerical approaches have been used to investigate the heat transfer phenomena in insulating gap; involving several geometries and configurations. Burlon [14] conducted a study on the effects of insulation parameters on the energy consumption in domestic ovens. They indicated that not only computational studies, but also experimental studies are important in understanding the heat and fluid flow in domestic ovens. Among other things the study focused on deciding optimum insulation design, for decreasing thermal bridges over itself. Their findings indicated that the effect of insulation in enclosures should not be underestimated in order to keep the heat in the center of the cavity. Their experimental studies showed that energy consumption of the domestic oven decreased 4.5% with the new insulation design. Sari et al. [13] proposed a method for finding the transient temperature variation in an insulated cooking device. They also reported on a means of optimizing the thickness of insulation. The method developed for finding the transient variation in temperature was tested on two cooking device volumes: 120 and 700 liters. Using the optimized parameters, a reduction in heat loss of 22% and 30%, respectively, were observed. A correlation was also developed to predict the heat transfer coefficient as a function of Rayleigh number and height to width ratio of their experimental device. According to Muthukumar and Shyamkumar [9], the quantity of the thermal mass and frequency that thermal mass is heated, to a large extend influence the quantity of the heat lost. Literature on energy saving in the field of process heating is common [11, 12, 15, 16]. According to Therkelsen et al. [3], energy optimization can be used to reduce the cost of heating ovens.

Technically, two main types of ovens are identified in the commercial baking industries: in direct fired force convection and direct fired natural convection ovens [1]. According to Davidson [17] and Adiutori [18], the main difference between them is that, an indirect fired oven makes use of heat exchanger concept to transfer energy from the burner to the product while a direct fired oven has the energy source placed inside the baking chamber [19]. The heat transfer coefficient is a constant coefficient; therefore, its value is independent of the temperature difference and the heat flux. Because heat transfer coefficient is a constant coefficient, correlations in the usual form of the Nusselt number as a function of Reynolds number and Prandtl number can be solved directly for the heat flux and for boundary layer temperature difference for direct fired force convection [20-22]. However, according to Kumar et al. [23], with natural convection, the heat flux is a nonlinear function of boundary layer temperature difference, and therefore heat transfer coefficient is a variable coefficient. Its value is therefore dependent on boundary layer temperature difference. Due to the fact that heat transfer coefficient is a variable coefficient, correlations in the form of Nusselt number as a function of the Rayleigh number cannot be solved directly for boundary layer temperature difference. They must be solved using an indirect method such as iteration or trial-and-error or experimentally [24]. Thermal energy losses can vary depending on the oven design, the oven technology as well as the energy source. Optimization of the energy consumption will lead to reduction in greenhouse gas emission and carbon footprints comparison of energy losses through area walls.

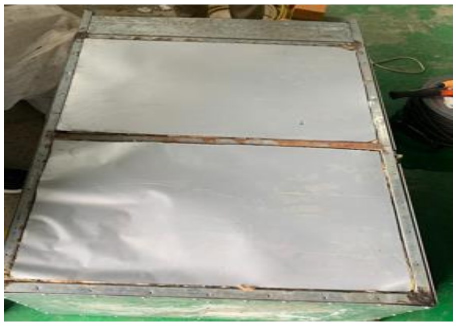
Earlier literature on thermal insulation in ovens have emphasized the effect of insulation materials on energy reduction [7, 15] with minimal emphasis on the effect on Nusselt number, Rayleigh number and heat transfer coefficient, which are important natural convective heat transfer performance parameters. In this study, a direct-fired oven was used to investigate the effect of insulation on the Nusselt number, Raleigh number as well as on reduction in convective heat transfer.

2. MATERIALS AND METHOD

The set-up consisted of a specially fabricated rectangular gas-fired oven (Figure 1). The walls, base and top of the oven were made of 1.25 mm thick galvanized steel plate. The oven was constructed such that insulation material could be inserted in between the outer metal plate (760×940×440 mm) and inner metal sheet plate (725×885×385 mm) as well as the base and the top sides of the oven (760×440 mm). A see-through double-glazed window made of tempered glass (220×570 mm) was inserted at the oven door to enable the checking of the baking process. The gas burner, with connection to the supply source of the liquified petroleum gas (LPG), was inserted inside at the bottom of the oven. A gas-regulator was fitted to the outlet of the gas cylinder to regulate the flow of the gas into the burner. The gas cylinder, which is the fuel supply source to the burner, was placed on a weighing balance scale to measure the gas consumption for the process. Temperature sensors, model REX-C700-FK07-M-AN; range 1300℃; output relay, were inserted inside the insulating wall of the top (P1), the side (P2), bottom (P3), back (P4) and inside in the burning chamber to measure the temperature while performing the experiment.

Figure 1. Experimental set-up of the direct gas-fired oven

(a) No insulation



(b) Fiber reinforced aluminum

(c) Fiber glass insulation

Figure 2. Experimental models of the direct gas-fired oven

Table 1. Thermal properties of the insulation materials

S/N	Properties	$\boldsymbol{K}$ [W/mK]	$\boldsymbol{C_p}$ [kJ/kgK]	$\boldsymbol{R_\theta}$ [mm²K/W]	$\boldsymbol{\alpha}$ [mm²/s]
1	Fiber glass	0.5-1.5	0.8-1.2	10-50	0.2– 0.5
2	Fiber reinforced aluminum	100-200	0.8-1.1	10-50	50-70

Two insulating materials, namely fiber-glass and aluminum reinforced-fiber were used for the experiment (Figure 2). The selection of the insulating materials was based on the availability of the materials in the market as well as their specifications as having high thermal insulation properties. Several research has reported on the thermal resistance of the two materials [25-27]. The thermal properties of the fiberglass and fiber-reinforced aluminum are indicated in Table 1. Three sets of experiments were performed. Initially, the experiment was performed without insulation material. With the gas cylinder placed on the weighing balance, the oven was turned on for preheating for 5 minutes. A data acquisition system comprising of a heat temperature sensor (REX-C700-FK07-M-AN) and a thermocouple (OM-HL-EH-TC) were used to continuously capture the temperature changes for 5 minutes from the four points P1, P2, P3 and P4 on the oven within 20 minutes interval. To reduce the margin of error, the process of data acquisition was repeated successively three times and the average was recorded. The weight of the LPG gas was continuously measured every 5 minutes to verify the gas consumption rate during the process. Subsequently, the first insulating material was installed in the hollow between the metal sheets and the experiment was performed. Lastly, the second insulating material was also installed and the experiment was repeated. The experiment was performed in the Mechanical Engineering Workshop of Koforidua Technical University.

2.1 Thermal performance analysis

The thermal performance of the oven was assessed by using the following dimensionless parameters, namely the Nusselt number, the Rayleigh number (Ra), and the heat transfer coefficient (h), which have been used by earlier researchers in assessing heat transfer characteristics [28, 29]. The Rayleigh number is the criterion for the ratio of buoyancy and inertia. Its value depends on the temperature difference between heated surfaces and fluids and characteristic height for a closed space. The Rayleigh number (Ra) which is defined as the product of the Grashof number (Gr) and Prandtl number (Pr) determines whether the buoyancy-driven natural convection plays an important role in heat transfer.

$$Ra = GrPr = \frac{g\beta(T_s - T_a)\delta^3}{\vartheta^2}Pr \tag{1}$$

where, $\beta = \frac{1}{(T_w + T_a)}$, is the volumetric thermal expansion coefficient, $[K^{-1}]$, g is acceleration due to gravity, $[ms^{-2}]$, $(T_w - T_a)$ is the temperature difference between the measuring point on the wall and the ambient, $[K]$, ϑ is the kinematic viscosity of the fluid, $[m^2s^{-1}]$. A constant Prandtl number of 0.7 was used.

The Nusselt number (Nu) provides a measure of the convection heat transfer at the surface in relation to the conductive heat transfer. It is defined as:

$$Nu = \frac{h\delta}{k} = cRa^n \tag{2}$$

The constants c and n depend on the geometry of the surface and the heat flow dynamics, they are determined by regression analysis and h is the convective heat transfer coefficient, $[W/m^2K]$.

2.2 Estimation of energy savings by reducing energy lost to the environment

Reducing the quantity of energy lost to the environment in a process heating system saves energy. The energy loss in the form of heat savings were calculated as:

$$Q_s = Q_{un} - Q_{in} \tag{3}$$

where, Q_s is the heat savings as a result of adding insulation material, Q_{un} and Q_{in} are the heat losses from the oven surface without insulation and with insulation, respectively. For heat transfer systems, the modes of heat transfer almost never occur individually, they involve series and parallel heat flow paths and combinations of the three modes of heat transfer. However, earlier studies by Khatir et al. [1] indicate that the proportion of heat transfer attributed to convection is approximately 93% as compared to only 7% attributed to radiation. Therefore, in the analysis of heat loss to the ambient, heat transfer due to radiation was neglected and only heat transfer due to convection was used. The heat loss to the ambient environment was estimated as:

$$Q = hA(T_w - T_a) \tag{4}$$

where, A is area of the convection surface, [m^2]. The required input variables were measured and taken from standard engineering manuals. Temperature, geometry of the hot surface, the temperature of the surrounding air and the inside temperature of the oven are the input variables required. The convective heat transfer coefficient depends on whether the air movement over the hot surface is natural due to buoyancy effect as a result of the density of the air close to the hot surface in comparison to that of the ambient air. The value of the convective heat transfer coefficient is also dependent on the geometry of the surface as well as its orientation. It also depends on the variation of temperature on the surface, the thermophysical properties of the fluid.

2.3 Uncertainty analysis

Taking into consideration the measurement of the primary data, uncertainty (error) analysis was performed for the calculation of the heat transfer performance parameters, such as the Nusselt number, the Raleigh number and the heat transfer coefficient, by adopting the derivative method detailed in the study of Moffat [30] which has been used by Basak et al. [31] and Zhang et al. [32], and several others. Neglecting the uncertainties for the acceleration due to gravity, the kinematic viscosity, the Prandtl number, the uncertainties for R_a, N_u and h were analyzed as:

$$\left(\frac{\delta R_a}{R_a}\right)^2 = \left(\frac{\delta\beta}{\beta}\right)^2 + \left(\frac{\delta T_s}{\Delta T}\right)^2 + \left(\frac{\delta T_a}{\Delta T}\right)^2 + \left(\frac{3\delta L}{L}\right)^2 \tag{5}$$

$$\left(\frac{\delta N_u}{N_u}\right)^2 = \left(\frac{\delta Q}{Q}\right)^2 + \left(\frac{\delta L}{L}\right)^2 + \left(\frac{\delta w}{w}\right)^2 + \left(\frac{2\delta T_a}{\Delta T}\right)^2 \quad (6)$$

$$\left(\frac{\delta h}{h}\right)^2 = \left(\frac{\delta Q}{Q}\right)^2 + \left(\frac{\delta L}{L}\right)^2 + \left(\frac{\delta w}{w}\right)^2 + 2\left(\frac{\delta T}{\Delta T}\right)^2 \quad (7)$$

respectively, resulting in relative uncertainty as ±6.4%, ±4.1% and ±4.1%, for the Raleigh number, Nusselt number and the convective heat transfer coefficient, respectively.

3. RESULTS AND DISCUSSION

The heat transfer characteristics of the insulated and uninsulated ovens were studied. Fiber glass and Fiber reinforced aluminum were used as insulation materials. The temperature profiles, the Nusselt number, the Rayleigh number and the heat transfer coefficient were used to assess the heat transfer characteristics of the insulated and uninsulated oven to ascertain the heat loss reduction potential of the insulation materials.

Figure 3 shows the temperature profiles for the inside of the oven, the uninsulated oven and the insulated oven with Fiber reinforced aluminum as insulation. In general, the uninsulated temperature profiles are higher, ranging between 52.2-62.3°C and closer to the inside oven temperature profile, 137.5-150°C than the insulated temperature profiles, 48-59.2°C, during the heating period.

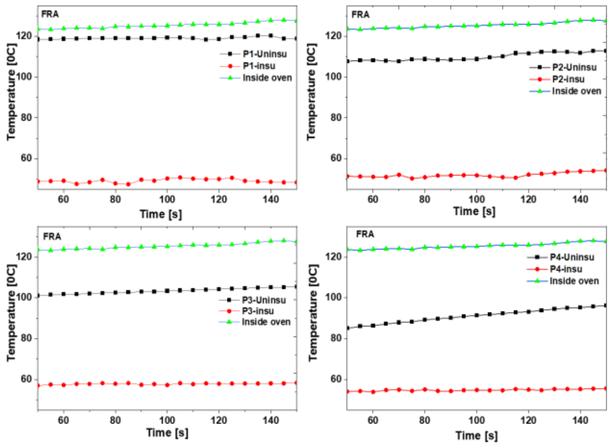


Figure 3. Temperature profile for the uninsulated, inside oven and insulated with Fiber reinforced aluminum (FRA)

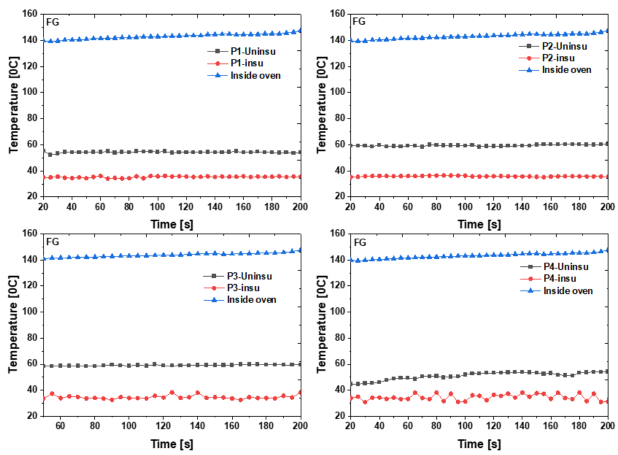
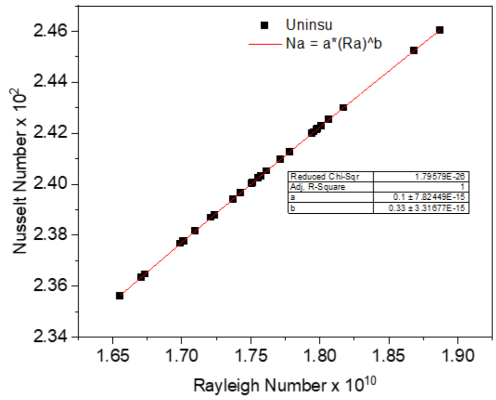
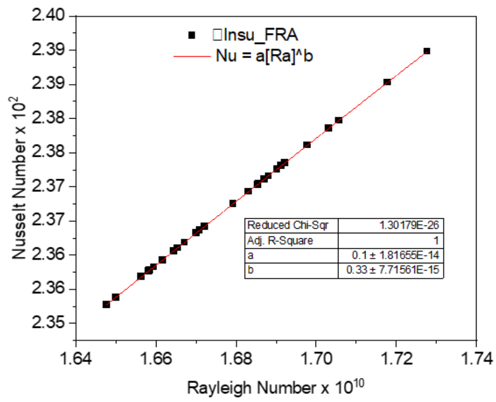


Figure 4. Temperature profile for the uninsulated, inside oven and insulated with Fiber glass (FG)

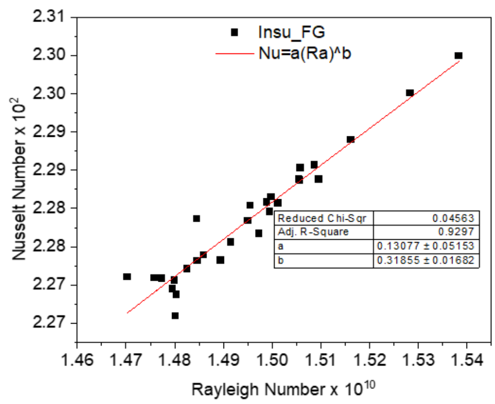
Figure 4 shows the temperature profiles for the inside of the oven, the uninsulated oven and the insulated oven with FG as insulation. It can also be seen that the trend in Figure 4 is similar to the trend in Figure 3, as the temperature profiles of the insulated oven with fiberglass ranging between 34-36.2°C are lower than that of the uninsulated oven temperature profiles 44.5-62.3°C at all four measuring points. The profiles show the effect of the insulation materials on the thermal characteristics of the oven.



(a) uninsulated



(b) insulated with fiber reinforced aluminum



(c) insulated with fiber glass

Figure 5. Nusselt number verses the Rayleigh number together with their corresponding regression correlations

The least square optimization of the values followed the power function as $Nu = a(Ra)^b$, which gave very high adjusted R-square values and the corresponding coefficients as follows: a=0.1±7.824 E-15, b=0.33±3.317 E-15 adjusted R-square value of 1; a=0.1±1.817 E-14, b=0.33±7.716 E-15 with adjusted R-square value of 1 and a=0.13077±0.05153, b=0.318±0.0168 with adjusted R-square value of 0.9297 for the uninsulated, insulated with fiber reinforced aluminum and insulated with fiber glass, respectively (refer to Figure 5). The high correlation coefficient shows good fit between the experimental and computed data. The above correlation functions are quite consistent with the *Nu* correlation with the Raleigh number [33] which also followed the power law such that $Nu = 1.621(Ra)^{0.145}$ for $Pr = 0.7$ heating was uniform, and $Nu = 0.293(Ra)^{0.249}$ for $Pr = 0.7$ for non-uniform heating.

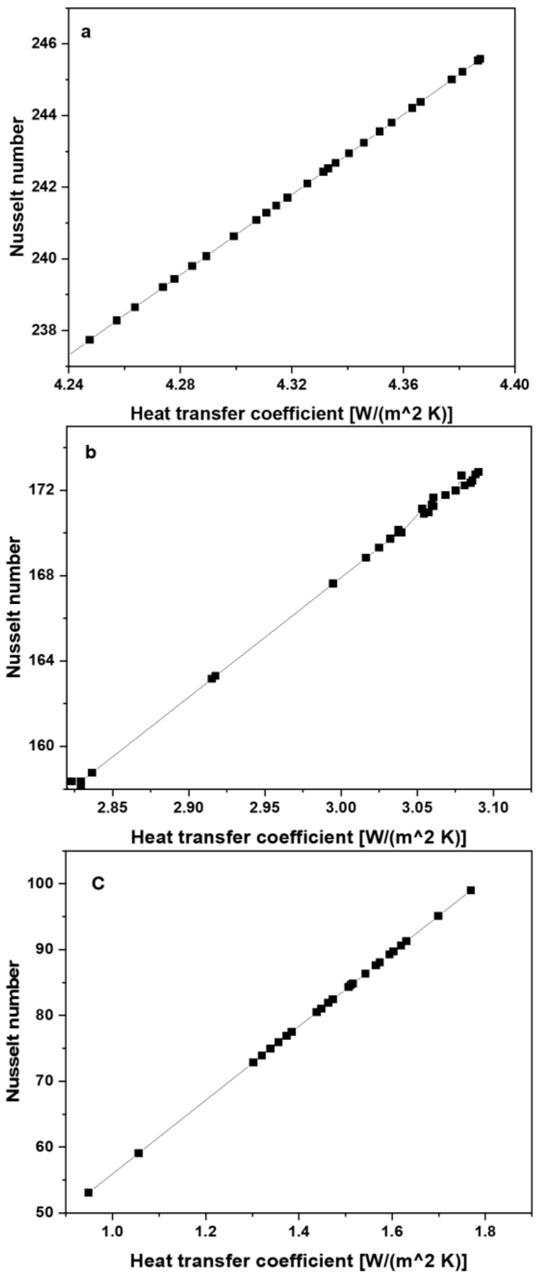
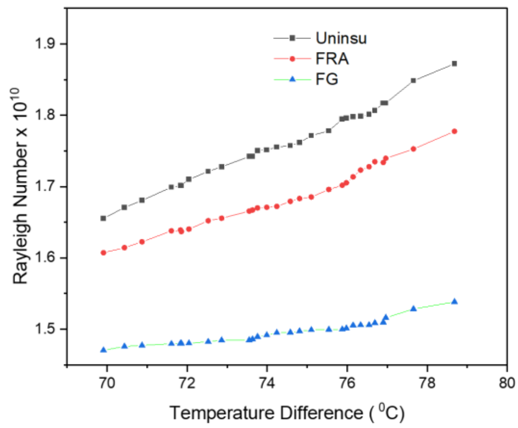


Figure 6. Relationships between Nusselt number and heat transfer coefficient for (a) Uninsulated oven, (b) Oven insulated with fiber reinforced aluminum, and (c) Oven insulated with fiber glass

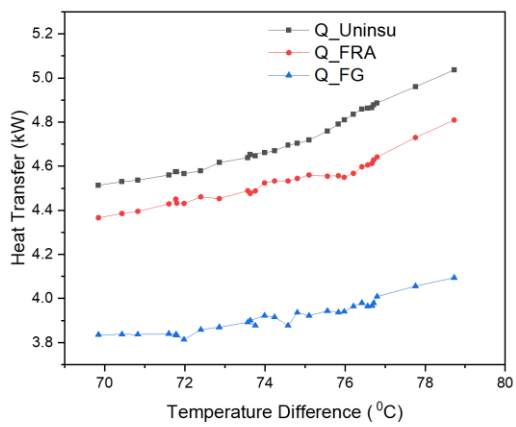
In this study, the Nusselt number is greater for the uninsulated compared to the insulated with fiber reinforced aluminum and fiber glass. Specifically, the range of values of the Nusselt numbers are 2.35 to 2.39; 2.33 to 2.38; and 2.271 to 2.299 for the uninsulated, insulated with fiber reinforced aluminum and insulated with fiber glass, respectively. The above observation is very significant. A lower Nusselt number is desirable as heat convection away into the ambient is reduced, assuming all other parameters are constant. The high Nusselt values for the uninsulated oven are attributed to the higher heat convection flux from the shell walls of the uninsulated oven compared to the insulated oven.

Another observation of interest is that, the Nusselt number for the insulation with fiber reinforced aluminum compared to the insulation with fiber glass. Over the Rayleigh number range studied, the Nusselt number is higher for the insulated with fiber reinforced aluminum than that of the insulated with fiber glass. Higher values of Nusselt number indicate that the heat transfer induced by convection from the oven walls is enhanced. Based on the Nusselt numbers of the two insulation materials, it can be deduced that the fiber glass has a higher resistance to heat transfer induced by convection and therefore, conversely, it has a higher heat loss reduction potential than the fiber reinforced aluminum.

The effect of the variation of insulation materials on the Nusselt number with heat transfer coefficient is shown in Figure 6. The Nusselt number varied linearly with the heat transfer coefficient for all the three cases, with the heat transfer coefficient ranging from 4.25-4.39, 2.83-3.09, and 0.95-1.76 with their corresponding Nusselt number from 237.7-245.5, 158.4-172.9, and 53.1-99.0 for the uninsulated, insulated with fiber reinforced aluminum and insulated with fiber glass, respectively. The heat transfer coefficient values are least for the insulated with fiber glass, higher for the insulated with fiber reinforced aluminum and highest for the uninsulated oven.



(a)



(b)

Figure 7. Relationships between temperature difference and (a) Rayleigh number, (b) Heat transfer for uninsulated oven, and ovens insulated with fiber reinforced aluminum and fiber glass

Figure 7(a) illustrates the relationship between the Rayleigh number and the temperature difference for scenarios using no insulation, fiber-reinforced aluminum, and fiberglass insulation, respectively. It is evident that within the examined range of temperature differences, the Rayleigh number is directly proportional to the temperature difference. This finding aligns with existing theoretical and empirical observations. An increase in temperature difference results in a higher Rayleigh number, indicating that buoyancy forces surpass viscous forces. This leads to convective heat currents becoming more dominant at higher Rayleigh numbers in the sequence of uninsulated, insulated with fiber-reinforced aluminum, and insulated with fiberglass.

Figure 7(b) on the other hand shows the variation of heat transfer with temperature difference for the uninsulated and insulated with fiber reinforced aluminum and fiber glass, respectively. In all the three cases, as temperature difference between the oven walls and the ambient temperature increased, the heat transfer induced by convection also increased. However, the heat transfer induced by natural convection increased from 4.514 kW to 5.037 kW, 4.366 kW to 4.809 kW and 3.836 kW to 4.09 kW for the uninsulated, insulated with fiber reinforced aluminum and insulated with fiber glass, respectively. The range of heat transfer induced by convective flux was highest for the uninsulated, followed by the insulated with fiber reinforced aluminum and lastly for the insulated with FG fiber glass. The influence of the insulation materials in retaining heat in the oven can be clearly seen in Figure 6(b). The gap between the uninsulated and the insulated graphs shows the extent of heat loss reduction by introducing the insulation material. The average percentage reduction in heat loss as a result of insulation are 3.8% and 16.7% for the fiber reinforced aluminum and fiber glass, respectively. Exclusive of heat transfer by radiation and conduction, the heat loss reductions are quite consistent with the findings of earlier research [4, 5, 7].

4. DISCUSSION

The temperature profiles show the effect of the insulation materials on the thermal characteristics of the oven. The effects of the insulation material on heat loss reduction are demonstrated by the differences in the temperature profiles of the uninsulated and insulated ovens. For comparison purposes, the average temperature profiles of the insulated with fiber reinforced aluminum and fiber glass were found to be 10.9% and 13.5% lower than the uninsulated, respectively. The above findings are consistent with findings of earlier research as indicated in the literature [7]. It is important to indicate that a rectangular oven was used for this study and not a perfect square. That means the surface area of the four measuring sides are not all the same. Therefore, variation in heat concentration within the cavities due to different surface areas at the four measuring sides of the oven may introduce some discrepancies in the results. Normalization of the temperature values with surface area is therefore recommended.

Considering the Nusselt numbers of the two insulation materials, it can be deduced that the fiber glass has a higher resistance to heat transfer induced by convection and therefore, it has a higher heat loss reduction potential than the fiber reinforced aluminum. That implies that the energy saving potential in using fiber glass as an insulation material is higher compared to fiber reinforced aluminum. In this study, a constant thickness of 30 mm was used for both insulation materials. The dynamics of heat loss reduction with varying insulation material thickness were not considered to ascertain minimum material thickness for optimum insulation. However, theoretically, the relationship between insulation thickness and heat loss is governed by Fourier's law of heat conduction. Further research will be necessary to establish minimum material thickness for optimum insulation of the fiber glass.

5. CONCLUSION

The effect of wall insulation on the Nusselt number and on heat loss reduction in a natural convection direct-fired oven has been investigated.

1. The Nusselt number was found to be greater for the uninsulated oven compared to the insulated ovens due to the higher heat convection flux from the shell walls.
2. Within the range of the Raleigh numbers studied, the Nusselt number is higher for the insulated with aluminum reinforced fiber than that of the insulated with fiber glass.
3. The heat transfer coefficient values are least for the insulated with fiber glass (0.95-1.76), higher for the insulated with fiber reinforced aluminum (2.83-3.09) and highest for the uninsulated oven (4.25-4.39).
4. The average percentage reduction in heat loss as a result of insulation are 3.8% and 16.7% for the aluminum reinforced fiber and fiber glass, respectively.
5. For the uninsulated, the insulated with fiber reinforced aluminum and the insulated with fiber glass, the heat transfer induced by natural convection increased from 4.514 kW to 5.037 kW, 4.366 kW to 4.809 kW and 3.836 kW to 4.09 kW, respectively.
6. Least square correlation of the Nusselt number with the Rayleigh number was found to follow a power law variation with high values of adjusted R-square values, which can be used for optimization of natural convective heat transfer for direct-fired ovens.
7. Variation in heat concentration within the cavities due to different surface areas at the four measuring sides of the oven may introduce some discrepancies in the results. Normalization of the temperature values with surface area is therefore recommended.

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NOMENCLATURE

A	surface area, m^2
C_P	specific heat capacity, J. kg^{-1}. K^{-1}
g	gravitational acceleration, m.s^{-2}
k	thermal conductivity, W.m^{-1}. K^{-1}
h	heat transfer coefficient
Gr	Grashof number
Nu	Nusselt number
Pr	Prandtl number
Q	heat flux, Wj/m^2
Ra	Rayleigh number
T	temperature, K
T_w	Wall temperature
T_a	Ambient temperature

Greek symbols

α	thermal diffusivity, m^2. s-^{1}
β	thermal expansion coefficient, K^{-1}
ϑ	kinematic viscosity of the fluid, m^2s^{-1}

Subscripts

i	insulated
u	Un-insulated