Heat Transfer Coefficientfora Glazed TranspiredSolarCollectorin Natural Convection Mode

Abstract(The abstract section is a direct copy of the conclusion part. It is better if this section is enriched and comprehensive)

A glazed transpired solar collector (GTC)was set up to operate in natural convection mode. Collector air temperatures and velocities were measured at specific points in the collector. Solar radiation, ambient temperature, collector cover glass and absorber temperatureswere also measured. Using dimensional analysis and the measured data,a heat transfer correlationwas obtained for the collector which will enable the heat transfer coefficient to be determined. The GTC heat transfer correlation based on influence of pitch is $Nu_p = 0.44\left(\frac{P}{D}\right)^{-1.6} Ra^{0.2}$. The GTC absorber plate hole diameterwas 3.0 mm and the pitches ranged from 15 mm to 35 mm. The Nusselt number correlation for influence of pitch, P for$15\, mm \leq P \leq 35\, mm$ predicted all Nusselt numbers well with maximum error of 5.2%. This correlation is valuable for future studies on optimization of pitch for a given hole size in the application of the GTC for air heating and crop drying.

Keywords: Transpired solar collector,Natural convection, Heat transfer correlation, Heat transfer coefficient, Pitch, Nusselt number.

Introduction(Put full stop at the end of sentences)

Although flat plate glazed non transpired solar collector (GNTC) and unglazed transpired solar collector (UTC)used as air heaters for crop drying and space heating exist they have major disadvantages. One such disadvantage of the UTC is high convection heat loss due to wind effects when the air pressure across the collector absorber is insufficient. It then follows thatthe UTC would have little or no practical value when operated in natural convection mode due to the relatively low collector pressure involved in natural convection systems. Another disadvantage of the UTC is that it requires electricity to power the fan used for its operation, limiting its usefulness to places that have reliable and adequate electricity supply.A major disadvantage of the GNTC is that its absorber is non transpired andair transpiration through absorber plate of solar collectors enhance the performance of the collectors, Walker *et al* [1].These disadvantages of the GNTC and UTC reduce their efficiency for crop drying in natural convection mode. Furthermore, although the theories of GNTC and UTC abound in the literature, they cannot be applied to the GTC in natural convection mode. This is because the UTC operates in forced convection mode, and performance variables and the method of correlating them are different from those in natural convectionwhile the geometry of the GNTC is significantly different from that of the GTC. As a solution to the problems of GNTC and UTC in natural convection mode operation, a glazed transpired collector (GTC) was experimentally studied. The GTC addresses the disadvantages of the GNTC and the UTC in natural convection mode by incorporating glazing over the absorber to reduce convection heat loss due to wind effect that would occur with the UTC in natural convection mode and by providing air transpirationthrough the absorber plate to enhance the performance of the GTC over that of the GNTC of the same dimensions used in natural convection mode.

Besides performance indices like thermal efficiency and heat exchange effectiveness, it is often useful to determine the quantity of heat that a transpired absorber flat plate is capable of transferring to ambient air. To do that the heat transfer coefficient has to be evaluated first and this is not usually straight forward. A more direct approach would be to develop correlation equations which give Nusselt number as a function of Rayleigh number, and the absorber or collector geometry.The Nusselt number is then determined from such equations after which the heat transfer coefficient is determined from the relationship between the Nusselt number and the heat transfer coefficient. The quantity of heat transferred by the absorber plate is then easily determined from Newton's cooling law.

Correlation equations for natural convection heat transfer from flat plates have been reported by several researchers. Natural convection heat transfer coefficients for isolated vertical and horizontal plates from 1949 to 1988 have been reviewed [2]. Most of the correlations were reported in the standard classical form $Nu = C(GrPr)^n$ with $n = 1/4$ for laminar regime and $n = 1/3$ for turbulent regime. However, most recent investigations report different forms of the equations as well as different values for the index n.The constant C was found to vary between0.516 and 0.726 for convective heat transfer from vertical surfaces in the laminar regime. For the turbulent regime the constant was found to vary between 0.056 and 0.130. Most of the correlations for horizontal surfaces were reported in the $h\ vs\ \Delta T$form. Most of the flat plates of the collectors studied which were presented in the review were not transpired.

Studies have been done on transpired solar collectors and heat transfer correlations have been obtained for the collectors. The heat transfer correlation given by Equation (1) was obtained for air flow through 6.35 mm thick perforated plates having square arrays of holes with different hole diameters and pitches [3]. The range of values of the variables for which Equation (1) was obtained were $500 < Re_D \leq 43{,}000$, $1.9 < P/D < 22$, $0.8 < t/D < 9.9$ and the suction mass flow rate was from 0.1 to 1.7 $Kg/m^2\ s$

$$Nu_D = 2.44(P/D)^{-1.43} {Re_D}^{0.57} Pr^{0.33} \quad (1)$$

Sparrow and Ortiz [4] performed experiments to determine heat transfer coefficients on an upstream-facing surface having triangular array of holes. Calculations showed that per-hole Reynolds number Re as large as 20,000 could be obtained for P/D ratios of 2.0 and 2.5 and the corresponding hole sizes were 1.27 cm and 1.17 cm respectively. The hole pitch-to-diameter ratio $P/D$and the per-hole Reynolds number Re were varied in the experiments and the results were correlated as given by Equation (2)

$$Nu_{L^*} = 0.881 Re^{0.476} Pr^{1/3} \quad (2)$$

Kutscher[5] experimentally studied heat exchange through perforated plates for a range of plate porosities from 0.1 to 5%, hole Reynolds numbers from 100 to 2000 and wind speeds from $0\ to\ 4\ m/s$. The tests plates had diameters from 0.5 mm to 3.5 mm, hole pitches from 5 mm to 30 mm and the range of suction mass flow rate through the holes in the plates were from $0.02\ to\ 0.07\ kg/m^2\ s$. The heat transfer correlation Equation (3) was developed from the study for normal flow (i.e. without cross wind).

$$Nu_D = 2.75(P/D)^{-1.2} {Re_D}^{0.43} \quad (3)$$

Gawlik and Kutscher [6]numerically and experimentally studied heat transfer from a corrugated transpired plate with suction to air flowing perpendicular to the corrugations. In the study heat loss to the air stream over the plate was determined as a function of wind speed, suction velocity and plate geometry. The heat transfer correlation for the attached flow was given by Equation (4) while Equation (5) gave the correlation for separated flow.

$$Nu_{att} = Nu_{flat}\{1 + 0.81(A/P)^{0.5}\} \quad (4)$$

$$Nu_{sep} = 2.05(A/P)^{1.40}Re^{1.63} \quad (5)$$

Wang et al [7]did an experimental study on heating characteristics and performance optimization of transpired solar collectors. They developed a heat transfer correlation given by Equation (6)

$$Nu_{wi} = 0.664Re_2^{0.5}Pr_2^{0.33} \quad (6)$$

The correlations given by Equations (1) to (6) were developed in forced convection mode.

Bokor et al [8] developed heat transfer correlation for nocturnal passive cooling by unglazed transpired solar collectors given by Equation (7)

$$Nu_D = 1.05Re_D^{0.82} \quad (7)$$

Rani and Triparty [9] did an experimental study on heat transfer performance of a solar collector with baffles and semicircular loops fins under varied mass flow rates and developed a correlation given by Equation (8). The collector was glazed but the absorber of the collector was not perforated.

$$Nu_{p,g} = 1 + 1.44[1 - 1708(sin1.80\theta)^{1.6}/Racos\theta][1 - 1708/Racos\theta]^+ + [(Racos\theta/5830) - 1]^+ \quad (8)$$

Ekoja et al [10]carried out an experimentalstudy on a glazed transpired solar collector in natural convection mode and developed a heat transfer correlation given by Equation (9)

$$Nu_D = 32.88(P/D)^{-2.8}Ra^{0.02} \quad (9)$$

The diameter of the holes in the absorber plates of the collector ranged from 1.0 mm to 3.0 mm while the pitch was fixed at 25.0 mm. The corresponding Rayleigh number ranged from 3080 to 18388. The characteristic length of the Nusselt number was the hole diameter. The authors reported that Equation (9) predicted Nusselt number for plates with hole diameters less than 2.0 mm with very high errors due to the difficulty of measuring the air velocity accurately for plates with hole diameters less than 2.0 mm. The authors also reported that the collector with the best performance had absorber plates with 3.0 mm diameter holes.Since there were inaccuracies in the measurement of the collector velocities when absorber plates with hole diameter less than 2.0 mm were used it follows that there were high errors associated with the Rayleigh numbers when those plates were used and the accuracies ofNusselt numbers predicted from Equation (9) were affectedby those errors.

A characteristic length which is a geometric dimension is required in order to develop a Nusselt number heat transfer correlation. The choice of the characteristic length could be any appropriate component of the plate geometry, Kutscher [5].In this work the Nusselt number correlation is developed using the pitch as the characteristic length. The GTC of the current work had an absorber plate with 3.0 mm diameter holes and had the same dimensions as that studied by Ekoja et al[10], to enable a comparison with the result of Nusselt number prediction from their model.The authors reported that the GTC had the best performance with the absorber plate that had 3.0 mm holes.Using the pitch as characteristic length is particularly useful if it is required to determine the pitch that gives optimum heat transfer from an absorber plate of a given hole diameter.

It can be shown that thegeneral heat transfer correlation equation for the GTC using dimensional analysisis given byEquation (10),Ekoja et al [10].

$$\frac{hD}{k} = f\left[\frac{\mu C_p}{k}, \frac{P}{D}, \frac{(\beta g\,\Delta t)D^3}{\nu^2}\right]. \tag{10}$$

In order to take the inclination of the collector into account,g which is a vector quantity was replaced by $gCos\,\alpha$, where α is the angle that the collector absorber plate makes with the horizontal. Equation (10) implied that the Nusselt number was a function of the Prandtl and Grashof numbers as well as the ratio of pitch to diameter of the holes in the plate as given in Equation (11).

$$Nu = f\left(Pr, Gr, \frac{P}{D}\right) \tag{11}$$

The product of Prandtl number Pr, and Grashoff number Gr, is the Rayleigh number Ra which is similar to Reynold number Re in forced convection. The general form of the correlation equation for heat transfer for the flat plate glazed transpired solar collector was therefore given byEkoja et al [10] as

$$Nu = C\left(\frac{P}{D}\right)^n (Ra)^m \tag{12}$$

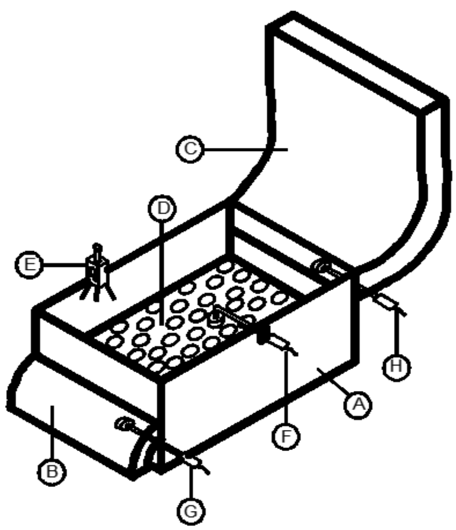
The values of the leading coefficient C, and the exponentsn and m had to be determined from experiments. The convection heat transfer coefficient h can then be determined once the Nusselt number Nu is found from the specific form of Equation (12) and hence, the rate of heat transfer from the GTC to ambient aircan be evaluated easily.

Experimental Setup

The experimental rig shown in Figure 1 and the sketch of the set-up of theglazed transpired collector showing the positions of instruments used for measurement of various variables shown in Figure 2, were the same as the ones used by Ekoja et al [10]. The experimental rig was setup at the Federal University of Agriculture, Makurdi in June 2016. The measured variables were solar radiation incident on the glass cover, absorber plate temperature, cover glass temperature, inlet air temperature, heated air outlet temperature and velocity. The air inlet hood was installed to minimize the effect of occasional spurts of high wind speeds into the collector. Prior to the installation of the hood, the spurts of wind caused a significant increase in air velocity through the holes in the collector absorberplate resulting in the cooling of the plate,such that low absorber plate temperatures were measured even at periods when the solar radiation values were relatively high.

1. Air inlet hood; 2. Transpired glazed collector; 3. Duct; 4.Chamber; 5.Chimney; 6.Support.

Fig.1. General features of the experimental rig [10]



A. Collector; B. Air inlet hood;
C. Duct (a part of chimney); D. Transpired absorber;
E. Solar meter; F. Thermocouple);
G. Hot wire Anemometer 1; H. Hot wire Anemometer 2

Fig.2. Sketch of the solar collector showing locations of instruments [10]

The absorber test plates of this work have a fixed diameter of 3.0 mm and the pitches ranged from 15 mm to 35 mm. The holes in the absorber test plates shown in Figure 3 were on a triangular pitch patternbecause absorber plates that have circular holes on triangular pitch pattern perform better than plates having circular holes on a square pitch pattern or slots [11].

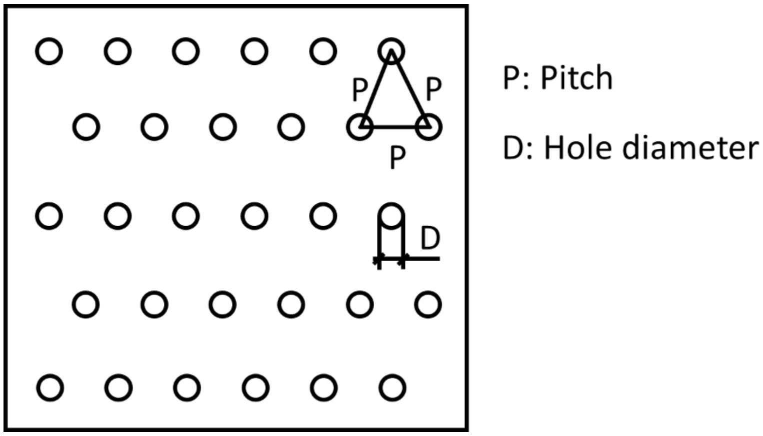


Fig. 3. Absorber test plate with holes on triangular pitch pattern

Solar radiation data were collected with a solar radiation meter PCE-SPM 1 made by PCE Group. It is a precision instrument made for measurement of solar radiation in the field and it is cosine corrected. It measured solar power in W/m^2 and solar transmittance and had data logging capability. The meter had an accuracy of $\pm 10W/m^2$ and a resolution of $0.1W/m^2$. Solar radiation values needed for practical applications of solar air heaters are usually in hundreds of watts per meter squared, therefore the accuracy of the solar meter was acceptable. The meter had a sampling rate of 4 times per second. The instrument was placed on the glazing with the sensor positioned to face the sun directly and it measured the global radiation (beam or direct radiation plus diffuse radiation mainly from the sky).

The surface temperatures of the absorber test plates were measured with 472A-1 Dual input Thermocouple Thermometer manufactured by Dwyer. A K-Type thermocouple having an L-bend probe suitable for solid surface temperature measurement was installed on the meter for the temperature measurements. The meter has an accuracy of $\pm 0.1\%$ and a resolution of 0.1^oC.

The temperature and velocity of the heated air were measured with AVM440 Velometer-a brand of hot wire anemometer made by Alnor TSI. The velocity measurement of the instrument has an accuracy of $\pm 0.015 m/s$ and a resolution of 0.01m/s. The accuracy of temperature measurement is $\pm 0.3^o C$ and a resolution of 0.1^oC. The instrument has data logging capability. Ambient temperatures were measured with AM4201A THERMO ANEMOMETER manufactured by Lutron. The accuracy of temperature measurement was 2.0^oC and a resolution of 0.1^oC. The optimal tilt angle for harnessing solar energy has been shown to range from 0^o- 42^o throughout the year in Nigeria [12]. The optimum angle was found to be 0^o during rainy seasons (April to August) in all locations while the optimum angle increased in the range of 5^o-42^o during the dry season (September-March), reaching the maximum value in the month of December in all locations. For this study a nominal collector tilt angle of 17.7^o was chosen for Makurdi in Benue state of Nigeria which is located on Latitude 7.7^oN.The collector tilt angle of 17.7^o was achieved with the aid of Wixey Digital Angle Gauge and the adjustment mechanism of the collector stand.

Results and Discussion

The results of the influence of absorber hole pitch on the performance of the solar air heater showed that the Nusselt number observed increased with decrease in absorber plate hole pitch (Figure 4). The Nusselt number observedrefers to Nusselt number computed using the standard method in the literature given by Equation (15). The explanation for this result is that although the test plates have the same hole diameter, a reduction of pitch meant having more holes, which in turn reduces resistance to air flow through the plate thereby enabling heat transfer to a greater mass of air and hence the increase in Nusselt number.Similarly, test plates that have larger pitches have fewer holes and present greater resistance to air flow with a consequent reduction in thermal performance. This explains why the plate having 35 mm hole pitch had the least Nusselt number and the plate having 15 mm hole pitch had the highest Nusselt number.

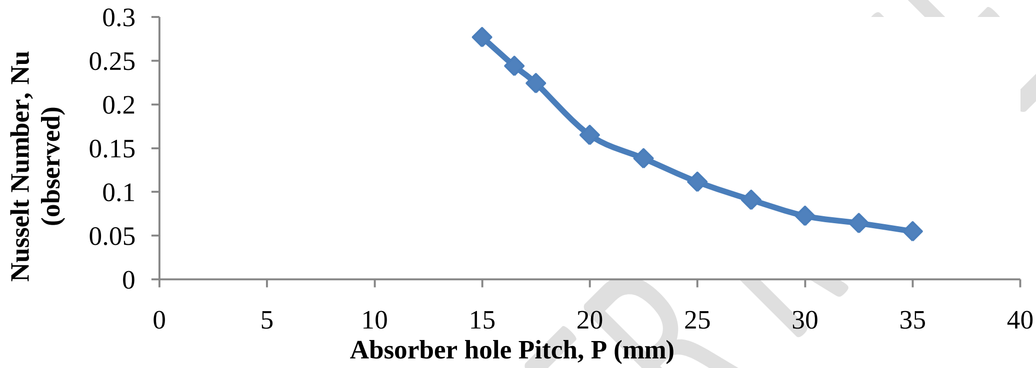


Fig. 4. Plot of Variation of Nusselt Number with Absorber hole Pitch

A plot of Nusselt number against the dimensionless ratio of pitch to hole diameter is shown by Figure 5. Figure 5 showed that the Nusselt number observed increased with decrease in pitch to diameter ratio. Here the diameter was held constant so a decrease in pitch to diameter ratio implied a larger open area in the plate resulting in increased air suction and higher heat transfer determined by the Nusselt number. The strength of the relationship between P/D and Nusselt number was tested by correlation analysis which yielded a coefficient of R= – 0.961. It followed that there was a strong relationship between P/D and Nusselt number. The negative sign in the value of R indicated an inverse relationship between them. Similarly, the correlation coefficient for the relationship between Rayleigh and Nusselt numbers was R= 0.994, the positive sign indicating a direct relationship between Rayleigh and Nusselt numbers. The plot of Nusselt number observed against the Rayleigh number is given by Figure 6. The Figure showed that Nusselt number observed increased with increase in Rayleigh number. This result is similar to that reported by [3-4, 13]. The correlation coefficients obtained in the relationship between P/D and Nusselt number Nu on one hand, and the relationship between Rayleigh number Ra and Nu on the other hand, suggested a relationship amongst the three variables of the general form given by Equation (13).

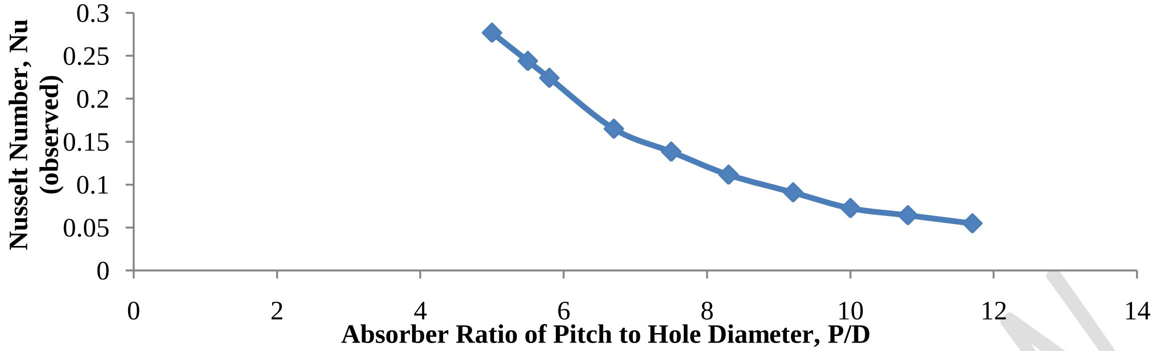


Fig. 5. Plot of Variation of Nusselt Number with Absorber Pitch to Hole Diameter Ratio

$$Nu_{D_{eq}} = K\left(\frac{P}{D}\right)^{-s} Ra^{e} \tag{13}$$

Equation (13) has three unknown parameters K, s and e to be determined from a single equation. A non-linear regression analysis gave the values of the parameters that best fit the data as $K = 0.44$; $s = -1.6$; $e = 0.2$ with a coefficient of determination, $R^2 = 0.9989$. Equation (14) was therefore obtained as the specific form of Equation (13).

$$Nu_P = 0.44\left(\frac{P}{D}\right)^{-1.6} Ra^{0.2} \tag{14}$$

$$Nu_D = hD/\kappa \tag{15}$$

Validation of model

To validate Equation (14) data obtained from the experimental rig shown in Figure 1 was used. Nusselt number computed using Equation (14) referred to as Nusselt number (predicted) was compared with Nusselt number computed using the standard method in the literaturereferred to as Nusselt number (observed) given by Equation (15). There was a close agreement between the observed and predicted Nusselt numbers. If Equation (14)was valid for predicting the Nusselt number, then a plot of Nusselt number observed against Nusselt number predicted should yield a straight line passing through the origin [14-16]. Also, if Equation (14)was valid then a plot of Nusselt number against $\left(\frac{P}{D}\right)^{-1.6} Ra^{0.2}$ (the Nusselt number factor, Nu^*) should also yield a straight line with a gradient equal to 0.44, [17]. Figure 7 is a plot of Nusselt number observed against the Nusselt number factor. The plot is a straight line with gradient of 0.45 suggesting that Equation (14) is valid since the difference between the expected gradient of 0.44 and the gradient of 0.45 obtained from the plot is only 0.01 (2.2%). A further validation was done by plotting Nusselt number observed and Nusselt number predicted against the Rayleigh numbers computed from experimental data and the physical properties of air (Figure 8).Figure 8 shows a good agreement between the two plots which implied that Equation (14) is valid for predicting the Nusselt number.

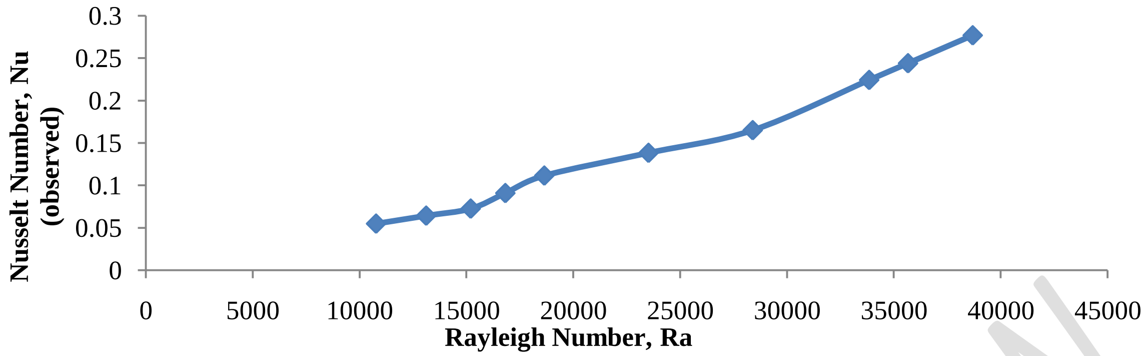


Fig. 6. Variation of Nusselt Number with Rayleigh Number

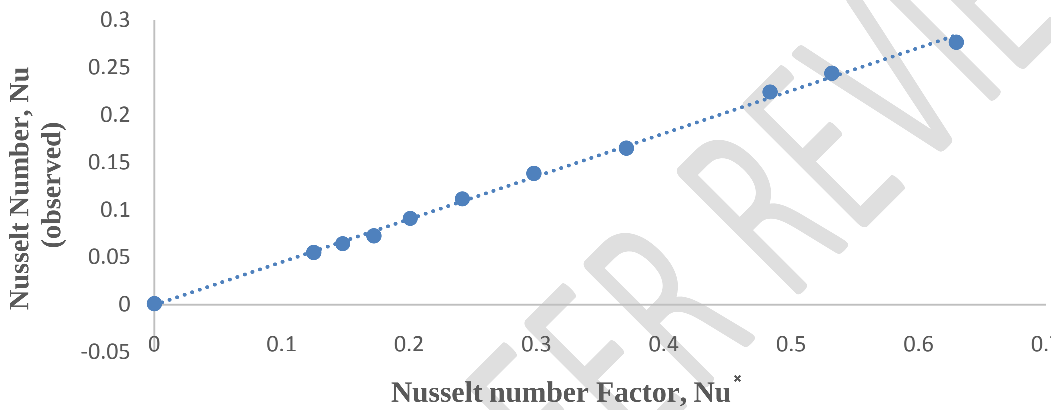


Fig. 7. Plot of Observed Nusselt Number against Nusselt number Factor

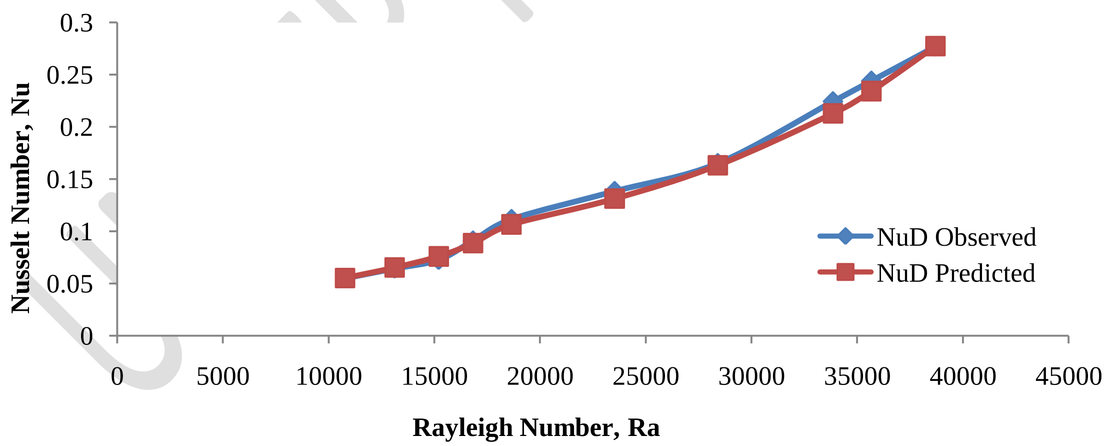


Fig. 8. Plot of Observed and Predicted Nusselt Numbers against Rayleigh Number

Comparison of models

The data presented in Table 1 was obtained from the experimental rig shown in Figure 1 and the data was used to compare the prediction of Nusselt number of the GTC using the correlation of this work with the one developed by Ekoja et al [10]

Table 1. Experimental Data for comparison of correlations

D_{eq} (m)	T_{amb}(^oC)	T_o(^oC)	T_p(^oC)	A_{ex}(m^2)	A_{col} (m^2)	V_{ex} (m/s)
0.068	31.9	39.2	41.0	0.052	0.277	0.25

The following values of the physical properties of air were obtained from literature at ambient

(Cite the references) temperature$T_{amb}$and used in computations requiring them since the difference between collector air temperature at outlet T_o and ambient air temperature is small: $\rho = 1.141(Kg/m^3)$; $C_P = 1.006(KJ/Kg\text{K})$; $\kappa = 0.0272(J/ms\text{K})$.

The "Observed" Nusselt number was computed from the standard method in the literature as

$$Nu_D = hD_{eq}/\kappa,$$

where the convective heat transfer coefficient was obtained from Ekoja et al [10] as given by Equation (16)

$$h = (T_o - T_{amb}/T_P - T_{amb})\, \dot{m}C_P/A_{col} \tag{16}$$

The predicted Nusselt number using the correlation given by Equation (14) which is based on the influence of holes' pitches is compared with the predicted Nusselt number using the correlation developed by Ekoja et al [10]which was developed based on the influence of the holes' diameters given by Equation (9)

$$Nu_D = 32.88\left(\frac{P}{D}\right)^{-2.8} Ra^{0.02} \tag{9}$$

The computed values for various parameters are presented in Table 2

Table 2. Computed parameters

h	Ra	Nu (Observed)	Nu_D (predicted)	Nu_P (predicted)	% Difference Nu_D	% Difference Nu_P
0.0432	18644	0.10803	0.10569	0.10574	2.17	2.12

It can be seen from Table 2 that the difference between the Nusselt number computed using the standard method in the literature referred to here as "Nu Observed" and the predicted Nusselt number Nu_P(predicted)computed from the developed correlation in this work based on the influence of absorber hole pitch is only 2.12% which shows close agreement between them. Also, the values of Nusselt numbers obtained from the correlation of Ekoja et al [10]based on influence of hole diameter shown in Table 2 as Nu_D(predicted) and that obtained from the correlation developed in this work based on influence of absorber hole pitch referred to as

Nu_P(predicted) are very close. This means that the Nusselt number correlation of this work is a good alternative for computing Nusselt number for a GTC and consequently determining the convective heat transfer coefficient. Note that the difference (in percent) between Nu_P(predicted) and Nu(observed) is slightly lower than that between Nu_D(predicted) and Nu(observed) indicating that the accuracy of predicted Nusselt number from the correlation of this work is slightly higher than that of Ekoja et al [10].**[Check grammar of this sentence**] The limitation of the Nusselt number correlation of this work is that the diameter of the absorber plate of a GTC must be fixed and new correlations must be developed for different absorber plate hole diameters for a GTC of the same dimensions.However, the Nusselt number correlation of this work provides a great tool for determining the best pitch between holes of a given diameter in the absorber plate of a GTC to optimize convective heat transfer.

Conclusion

A heat transfer correlation equation for a glazed transpired solar collectorused for heating ambient air in natural convection mode has been developed. The GTC heat transfer correlation based on influence of pitch is$Nu_P = 0.44\left(\frac{P}{D}\right)^{-1.6} Ra^{0.2}$. The Nusselt number correlation for influence of pitch for $15\ mm \leq P \leq 35\ mm$ predicted all measured Nusselt numbers wellwith maximum error less than 5%. The Nusselt number correlation of this work which is based on the influence of absorber plate hole pitch provides a good alternative to a correlation model based on the influence of absorber plate hole diameter. Furthermore, the correlation of this work could provide a great tool for determining the best pitch between holes of a given diameter in the absorber plate of a GTC to optimize convective heat transfer. The heat transfer coefficient can be easily computed once the Nusselt number is determined from the Nusselt number correlation of this work.

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Nomenclature

A_{col}= Area of collector (m^2)
C_p = Specific heat capacity of air at constant pressure (kJ/kgK)
D= Absorber Plate hole diameter (m)
g = Acceleration due togravity (m/s^2)
Gr = Grashof number
h = Convective heat transfer coefficient($J/m^2s.K$)
I= Solar radiation incident on collector (J/m^2s)
L or P = Pitch (m)

L_c= Characteristic length (m)
$\dot{m}_o$= Air suction mass flow rate (kg/s).
Nu = Nusselt Number
Nu_D = Nusselt Number based on influence of absorber plate holediameter
Nu_P = Nusselt Number based on influence of absorber plate hole pitch
P = Pitch of holes in the absorber plate (m)
Pr = Prandtl Number
Ra = Rayleigh number
t = Absorber plate thickness (m)
T_o =Temperature of heated air exiting the collector at the reverse end (K)
T_p = Absorber plate temperature(K)
T_{amb} = Ambient air temperature (K)

Greek

β = Volume expansion coefficient (1/K)
κ= Thermal conductivity (J/smK)
ρ = Density (kg/m^2)
ϑ = Kinematic viscousity (m^2/s)