Parametric Study of Thermal Performance of Cylindrical Parabolic Trough Solar Collector in Ogbomoso Environs.

Emmanuel O. Sangotayo, M.Tech.[1*]; Waheed M. Adekojo, Ph.D.[2]; and Jelili O. Alamu, Ph.D.[3]

[1]Department of Mechanical Engineering, Ladoke Akintola University of Technology, Ogbomoso, Nigeria.
[2]Department of Mechanical Engineering, College of Engineering, University of Agriculture, Abeokuta, Nigeria.
[3]Department of Mechanical Engineering, Osun State University, Osogbo, Nigeria.

E-mail: olemsangotayo@yahoo.com*

ABSTRACT

This paper presents a numerical investigation on the enhancement of thermal performance of solar air heater having cylindrical parabolic trough solar collector with twisted tape in Ogbomoso weather conditions (lat.8^{o}01$^{'}$, long. 4^{o}11^1). The parametric studies were conducted to investigate the effects of all the operating parameters on the system performance in order to obtain the optimum performance of the system.

The energy equation for heat transfer of two dimensional fully developed fluid flow of the cylindrical parabolic trough collector have been considered and subsequently collector efficiency factor, F', collector heat removal factor, Fr and collector overall heat loss coefficient, U_L are used to analyze the thermal performance and to study the effect of mass flow rate, $\dot{m}$ from 0 to 1.0 kg/s for the fixed value of incident solar absorbed flux, I_b of 186 w/m^2.

The results revealed that the optimum design parameters are length, 1.30m, mass flow rate, 0.036kg/s, concentrator aperture width, 0.6 m, concentration ratio, 3.667, absorbed flux, 96.39, tilt angle, 8.2, fluid temperature, 0.05053 K and instantaneous efficiency, 47.4%. It is observed that performance of the cylindrical parabolic trough collector with twisted tape was enhanced appreciably. These results can guild practicing engineers and designers in the evaluation of the existing real systems and design of future system.

(Keywords: thermal performance, cylindrical parabolic solar collector)

INTRODUCTION

Flat-plate solar collectors are vastly employed in low temperature energy technology and have drawn the attention of a large number of investigators. Several designs of solar air heaters have been industrialized over the years in order to improve their system thermal performance.

The thermal performance of solar air heater is generally low because of low value of the convective heat transfer coefficient between the absorber plate and the air, leading to high absorber plate temperature and greater heat losses to the surroundings. It has been found that the main thermal resistance to the heat transfer is due to the formation of a laminar sub-layer on the heat transferring surface, (Prasad and Mullic, 1983; Momin, 2001).

Efforts of improving the heat transfer rate have been directed towards artificially destroying the sub-layer. The suitable design of solar air heaters for high temperatures applications have been the subject of many theoretical and experimental investigations. Recently, researchers have employed wire screen matrices, expanded metal mesh, finned, corrugated, and packed bed as absorbing porous media for directly incident solar radiation in the solar air heater due to improve its performance because as the surface area and turbulence producing air flow path through the bed increase, heat transfer rate increases.

The high heat transfer area to volume ratio for the air flowing through these matrices enhances the heat transfer capability. Bhagoria, et al. [2002]; Han [1988]; Taslim [1996]; Lau [1991]; Park [1988]; Mital [2005]; Saini and Saini [1996] have carried out investigation on the performance of solar air heater. These studies have indicated that such air heaters have superior performance as compared to that of flat plate collectors.

Recently, much attention has been given to concentrating solar collectors, which are capable of reaching higher temperatures compared to flat plate collectors. Hong [1976]; Kumar and Prasad [2000], Sharma [2003], and Togrul and Pehlivan [2005] have carried out investigations on the twisted tape; it was reported for enhancement in performance. These works need investigations of the heat and fluid flow phenomena in the absorbing tube by evaluating friction factors, pumping power and convective heat transfer coefficients and comparisons for the alternative cases and may bring into light the factors to construct an economically viewable solar air collector for high temperature applications.

This paper aims at investigating the optimal values of the key design parameters and effect of twist tape factor on the system thermal performance in Ogbomoso weather conditions.

ANALYSIS OF CYLINDRICAL PARABOLIC CONCENTRATING COLLECTOR

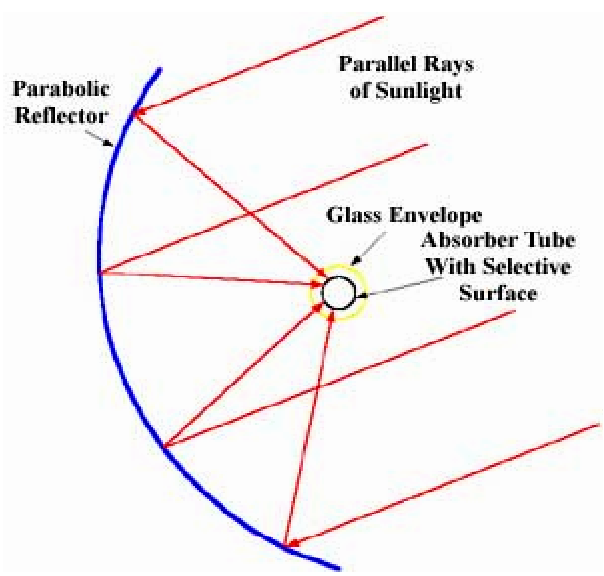

Figure1: Cross Section of Cylindrical parabolic Concentrating Collector.

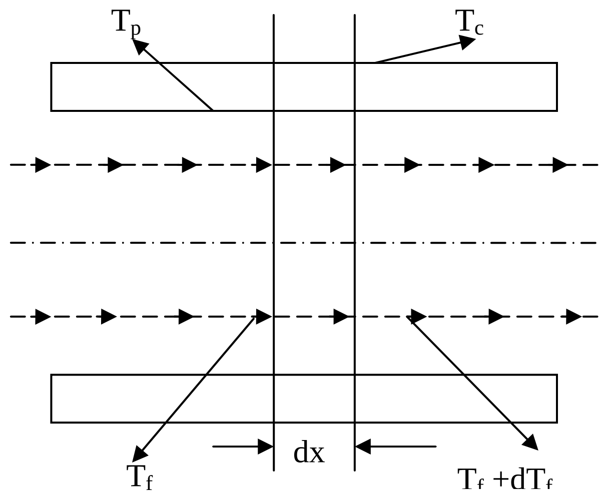

Figure 2: Energy Flow Diagram of Cylindrical Parabolic Concentrating Collector (Sukhatme, 2007).

PERFORMANCE ANALYSES

The structure of a cylindrical parabolic concentrating collector of glass envelope absorber tube with selective surface of such system having concentrator width aperture 'W' and length 'L' as shown in Figure 1. The performance of a cylindrical parabolic concentrating collector is assumed for the same radiation flux all along the length and the negligible temperature drop across the absorber tube and the glass cover. The energy balance under steady state conditions is shown in Figure 2 and the total heat gain rate can be expressed as:

$$dq_u = \begin{bmatrix} I_b r_b (W - D_o)\rho\gamma(\tau a)_b \\ + I_b r_b D_O (\tau a)_b \\ - U_L \pi D_o (T_p - T_a) \end{bmatrix} dx \qquad \textbf{(1)}$$

The left side term represents the useful heat gain rate for a length, dx. The first term on the right side represents the incident beam radiation absorbed in the absorber tube after reflection, while the second term represents the absorbed incident beam radiation which falls directly on the absorber tube and the third term represents the loss by convention and radiation. Also ργ(τα) represents the optical properties of the system. The intercept factor, 'γ' is defined as the reflected fraction of the incident radiation on the absorbing surface of the receiver. τ is the transitivity of transparent cover. α and ρ are the absorptivity and the reflectivity of the absorber and concentrator, respectively.

The absorbed flux 'S' can be given as:

$$S = I_b r_b \rho\gamma(\tau a)b + I_b r_b D_O (\tau a) b\left(\frac{D_O}{W - D_o}\right) \tag{2}$$

Equation (1) thus becomes:

$$dq_u = \left[S - \frac{U_L}{C}\left(T_p - T_a\right)\right](W - D_o)dx \tag{3}$$

where C is the concentration ratio of the collector which is defined as the ratio of the effective aperture area and absorber tube area and is written as:

$$C = \frac{(W - D_o)L}{\pi D_O L} = \frac{(W - D_o)}{\pi D_O} \tag{4}$$

The useful heat gain rate dq_u can be given as:

$$dq_u = h_f \pi D_i (Tp - T_f)dx \tag{5}$$

where h_f and T_f are the heat transfer coefficient on the inside surface of the tube and the local fluid temperature, combining Equations (3) and (5) to eliminate the absorber tube temperature (T_p), it yields:

$$dq_u = F'\left[S - \frac{U_L}{C}(T_f - T_a)\right](W - D_o)dx \tag{6}$$

where F' is the collector efficiency factor and written as:

$$F' = \frac{1}{U_L}\left[\frac{1}{U_L} + \frac{D_O}{D_i h_f}\right] \tag{7}$$

Therefore,

$$\frac{dT_f}{dx} = \frac{F'\pi D_O U_L}{mC_p}\left[\frac{CS}{U_L} - \left(T_f - T_a\right)\right] \tag{8}$$

The inlet fluid temperature (*T_{fi}*) and outlet fluid temperature (*T_{fo}*) are established by applying boundary conditions at inlet (x = 0. $T_f = T_{fi}$) and at outlet ($x=L$, $T_f = T_{fo}$). They are written as:

$$T_{fi} = \exp\left\{\frac{-F'\pi D_O U_L x}{mC_p}\right\} \tag{9}$$

$$T_{fo} = 1 - \exp\left\{\frac{-F' D_O U_L x}{mC_p}\right\} \tag{10}$$

Thus the useful heat gain rate can be expressed as:

$$Q_u = mC_p\left(T_{fo} - T_{fi}\right) = F_r (W - D_O)L\left[S - \frac{U_L}{C}\left(T_{fi} - T_a\right)\right] \tag{11}$$

where Fr, the heat removal factor is expressed as:

$$F_r = \frac{mC_p}{\pi D_O U_L L}\left[1 - \exp\left\{\frac{-F'\pi D_O U_L L}{mC_p}\right\}\right] \tag{12}$$

Equation (11) is the equivalent of the Hotel-Whillier-Bliss equation for the flat plate collector, (Sukhatme, 2007). The instantaneous collector Efficiency (h_i) is given by this equation:

$$\eta_i = \frac{Q_u}{(I_b r_b + I_d r_d)WL} \tag{13}$$

The instantaneous efficiency can also be determined on the basis of beam radiation alone, if the ground reflected radiation is neglected and given in Equation (14):

$$\eta_i = \frac{Q_u}{I_b r_b WL} \tag{14}$$

HEAT TRANSFER COEFFICIENTS

In order to determine the performance of cylindrical parabolic concentrating solar collector, correlations are required for calculating the values of convective heat transfer coefficient between the absorbed tube and cover, outside surface of the cover and inside surface of the absorbed tube, respectively.

The following correlations are used to calculate the natural convection heat transfer coefficient ($h_{p\text{-}c}$) for the enclosed annular surface between the horizontal absorber tube and the concentric cover.

$$K_{eff} / K = 0.317(Ra)^{1/4} \quad \textbf{(15)}$$

where K_{eff} is the effective thermal conductivity.

The convective heat transfer coefficient (h_w) on the outside surface of the cover may be calculated by this Equation (16):

$$Nu = C_1 \operatorname{Re}^n \quad \textbf{(16)}$$

The convective heat transfer coefficient (h_f) on the inside surface absorbed tube can be calculated using Dittus-Boelter equation:

$$Nu = 0.023 \operatorname{Re}^{0.8} \operatorname{Pr}^{0.4} \quad \textbf{(17)}$$

The heat transfer coefficient for twisted tape may be determined using the correlation given by Hong and Burgles equation.

$$Nu = 5.172\left[1 + 0.005484\left\{\operatorname{Pr}(\operatorname{Re}/X)^{1.78}\right\}^{0.7}\right]^{0.5} \quad \textbf{(18)}$$

where X is the tape twist ratio, it is expressed as:

$$X = \frac{H}{Dt} \quad \textbf{(19)}$$

A simple and flexible program, writing in C++ language was developed based on models above (Equations 1-19) to obtain results under different design and operating conditions. The flowchart for implementing the program is shown in Figure 3.

RESULTS AND DISCUSSIONS

The results obtained from the program developed are presented as profiles in Figures 4 – 17. The effect of variable mass flow rate ($\dot{m}$) ranges from 0.1 to 1.0 kg/s on the system thermal performance parameters were investigated for different values of tape twist ratio, X (1, 3, 6, 9, 12, and 15) for a fixed constant value of incident beam solar flux (I_b = 186 W/m^2) in Ogbomoso climatic conditions.

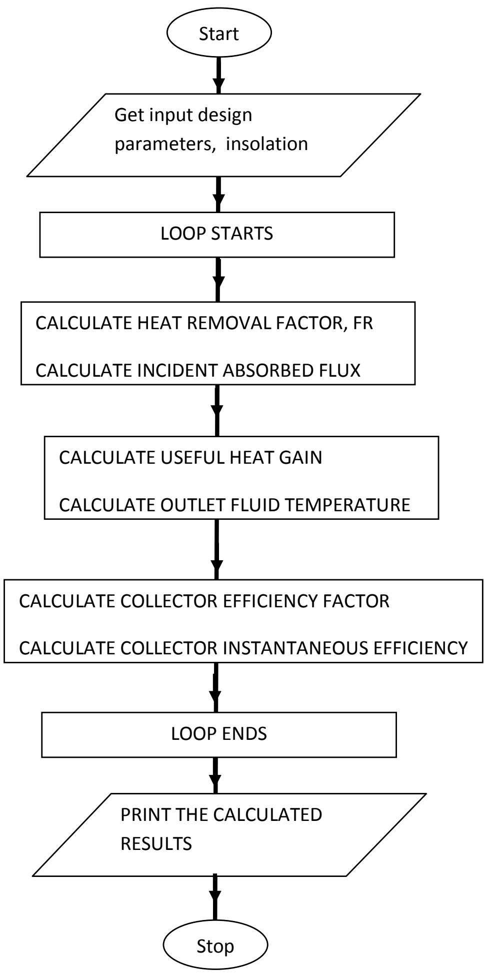

Figure 3: Flow Chart for Solar Collector Calculations.

Figures 4 – 7 show the plots of the collector instantaneous efficiency as a function of concentration ratio (C), concentrator aperture width (W), tilt factor (r_b) and length (L). Figures 8 and 9 show the plots of fluid inlet and outlet temperature distribution as a function of mass flow rate ($\dot{m}$) and length (L). It is deduced that the optimum design parameters are: length (L) is 1.30 m, mass flow rate is 0.036 kg/s, outlet and inlet fluid temperature is 0.505K with instantaneous collector efficiency of 47.38%.

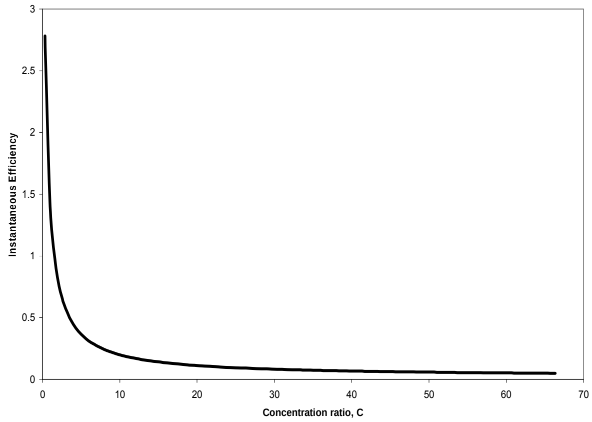

Figure 4: Effect of Concentration Ratio, C on Instantaneous Efficiency.

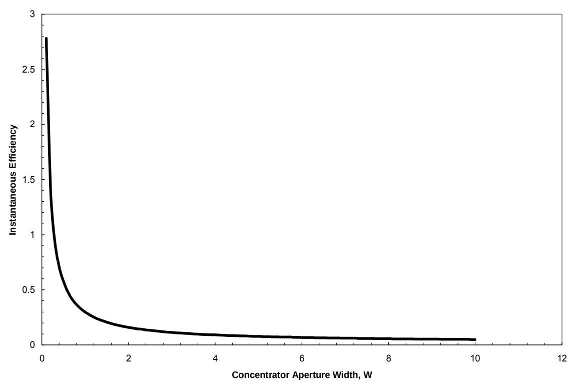

Figure 5: Effect of Concentrator Aperture Width, W on Instantaneous Efficiency.

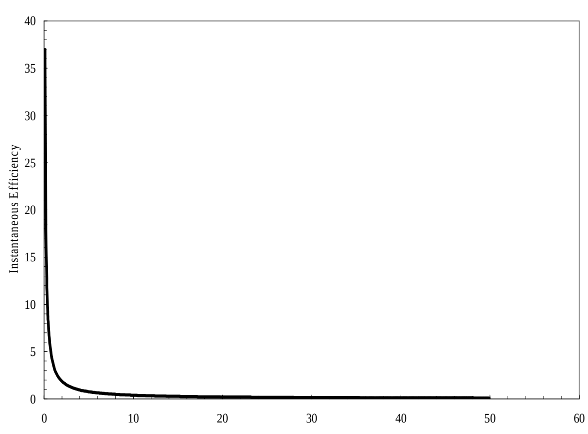

Figure 6: Effect of Tilt Factor r_b on Instantaneous Efficiency

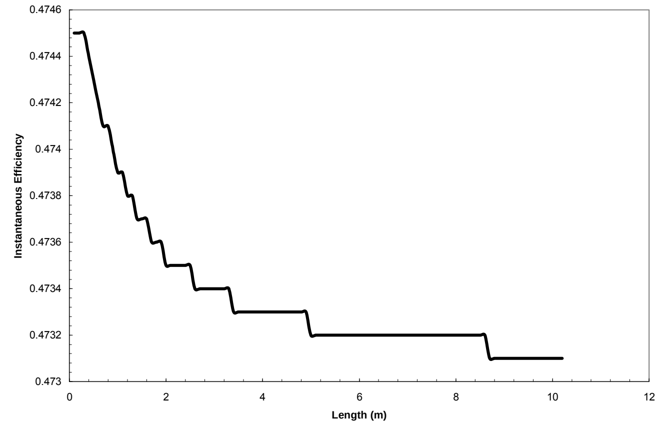

Figure 7: Effect of Variation of Length, L on Instantaneous Efficiency.

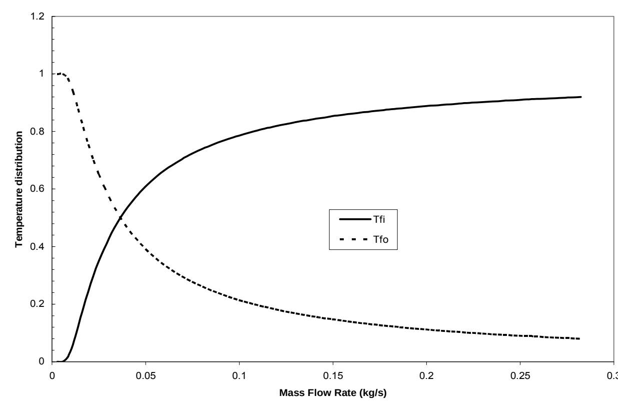

Figure 8: Effect of Mass Flow Rate, m on Fluid Inlet and Outlet Temperature.

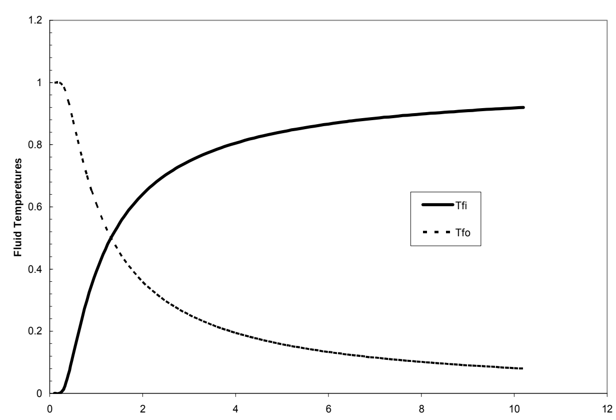

Figure 9: Effect of Length, L on Fluid Inlet and Outlet Temperature.

Figures 10, 11, and 12 present the effect of mass flow rate with variable tape twist factor, X on inlet fluid temperature distribution (T_{fi}), collector efficiency factor (F^1) and collector heat removal factor (Fr). It was found that the system losses reduce as the twist tape factor inserted in the absorber increases.

Figures 13, 14, 15, and 16 show the plots of fluid outlet temperature distribution, T_{fo} and collector overall heat loss coefficient (U_L), useful heat gain rate (Q_u) and instantaneous efficiency as a function of mass flow rate ($\dot{m}$) with variable tape twist factor, X. It was found that the system thermal performance parameters increases as the twist tape factor inserted in the absorber increases. Hence it enhances the thermal performance of the system.

Figure 17 show the plot of Nusselt number, Nu as a function of Reynolds number, Re with variable tape twist factor, X and Prandtl number for air, Pr = 0.71. It is evident that, higher value of Nusselt number is obtained with corresponding lower value of tape twist ratio, because high twisted tape increases the friction factor and pressure drop leading to higher pumping power.

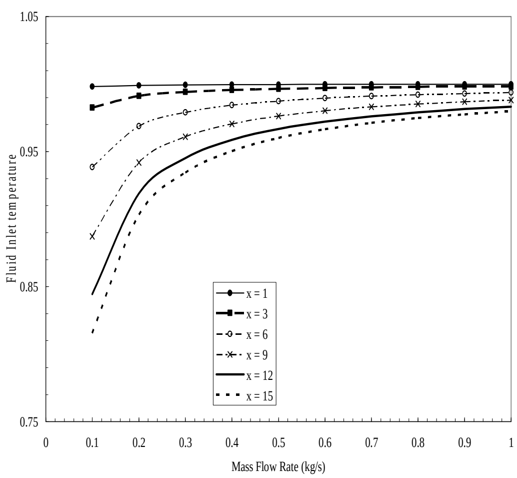

Figure 10: Effect of Mass Flow Rate Coupled with Twisted Tape Factor on Inlet Fluid Temperature

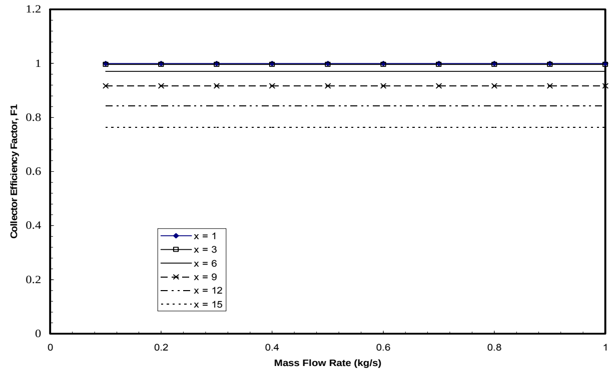

Figure 11: Effect of Mass Flow Rate Coupled with Twisted Tape Factor on Collector Efficiency Factor.

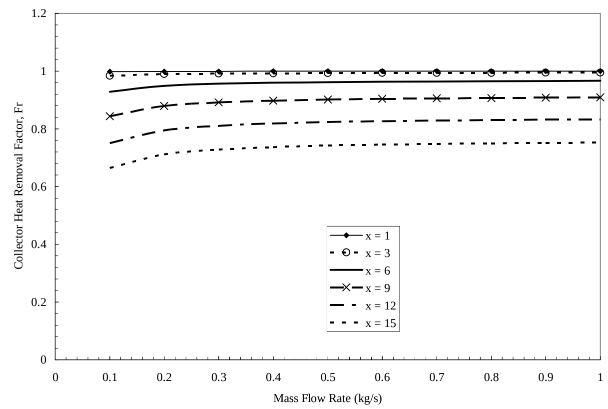

Figure 12: Effect of Mass Flow Rate Coupled with Twisted Tape Factor on Collector Heat Removal Factor.

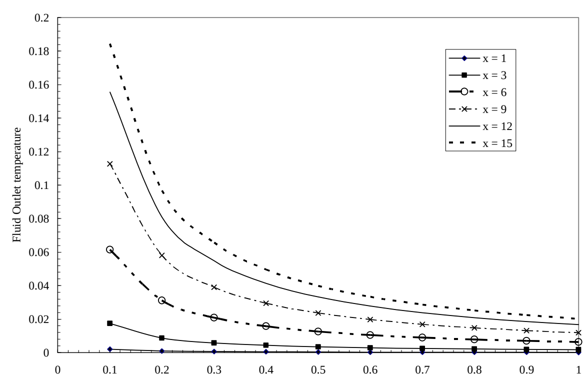

Figure 13: Effect of Mass Flow Rate Coupled with Twisted Tape Factor on Fluid Outlet Temperature.

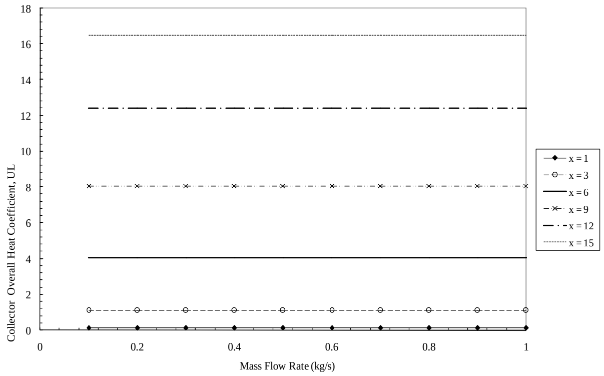

Figure 14: Effect of Mass Flow rate coupled with twisted tape factor on Collector Overall Heat Coefficient.

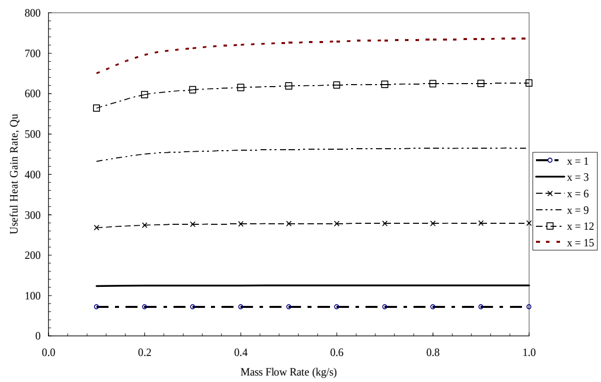

Figure 15: Effect of Mass Flow Rate Coupled with Twisted Tape Factor on Useful Heat Gain Rate.

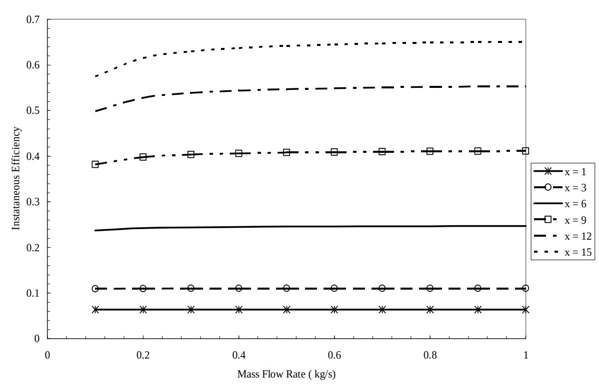

Figure 16: Effect of Mass Flow Rate Coupled with Twisted Tape Factor on Instantaneous Efficiency.

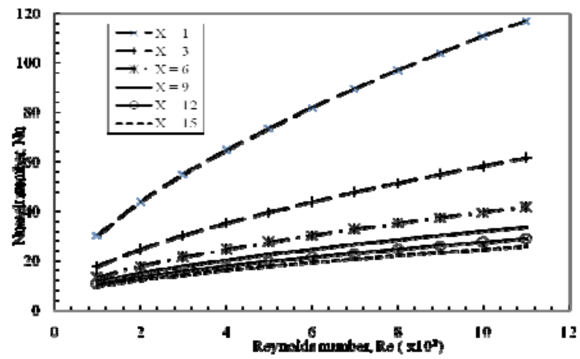

Figure 17: Effect of Reynolds Number Coupled with Twisted Tape Factor on Nusselt number, Pr = 0.71.

CONCLUSIONS

The following conclusions are drawn from the parametric studies conducted investigating the effect of design and operating conditions on the system performance. The optimal values of the key design parameters were established for length is 1.30 m and mass flow rate is 0.036 kg/s with collector instantaneous efficiency of 46.47%.

The insertion of twisted tape in the absorber tube reduces the losses in the system thermal performance and it increases the heat transfer leading to increase in the system thermal performance. The significant increase in heat transfer coefficient is observed with high value of Nusselt number.

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NOMENCLATURE

Fr	Collector Heat Removal factor
h_{p-c}	Heat transfer coefficient between absorber and cover (W/m^3 K)
hf	Convention heat transfer coefficient on the inside surface of the tube (W/m^3 - K)
X	Tape twist ratio
I_b	Incident beam radiation (W/m^3 - K)
U_L	Overall heat transfer coefficient (W/m^3 - K)
Re	Reynolds Number
Nu	Nusselt Number
Ra	Rayleigh Number
η	Instantaneous Collector efficiency
δ	Stefan-Boltzmann constant (W/m^3 - K)
τ	Transmissivity of glass
α	Reflectivity of glass
ρ	Absortivity of glass
ε	Emissivity of the plate
γ	Intercept Factor

ABOUT THE AUTHORS

Emmanuel O. Sangotayo, is a Lecturer at the Department of Mechanical Engineering. He is a registered Engineer (COREN) and is a member of the Nigerian Society of Engineer, (NSE). He holds a Master of Technology (M.Tech.) in Mechanical Engineering from the Ladoke Akintola University of Technology, Ogbomoso, Nigeria. He is a Ph.D. degree student at Federal University of Agriculture, Abeokuta, Nigeria. His research interests are in thermofluid/energy studies.

Dr. Waheed, M. Adekojo, is a Professor in the Department of Mechanical Engineering, Federal University of Agriculture, Abeokuta, Nigeria. He is a registered Engineer (COREN) and is a member of the Nigerian Society of Engineer, (NSE). He holds a Ph.D. degree in Mechanical Engineering and currently serves as the Director, Academic Planning, FUNAAB. His research interests are in the areas of thermofluid/energy studies.

Dr. Jelili O. Alamu, is a Reader in the Department of Mechanical Engineering, Osun State University, Osogbo, Nigeria. He is a registered Engineer (COREN) and is a member of the Nigerian Society of Engineer, (NSE). He holds a Ph.D. degree in Mechanical Engineering and currently serves as the Director, academic planning, Osun State University. His research interests are in the areas thermofluid/energy studies.

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