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Effect and Sensitivity of Solar Collector Parameters on the Heat Removal Factor

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ABSTRACT: *Satisfactory performance and reliability of a solar collector requires maximum collection of solar energy by the collector and improvement on the heat removal factor through proper configuration and adequate sizing of its components. This research work presents a parametric study to determine the sensitivity of the heat removal factor and solar energy received on the collector to the collector design parameters such as; tube spacing, internal tube diameter, and absorber plate thickness and collector tilt angle respectively. Computer programme codes developed using Matlab based on the appropriate equations and system characteristics were used for the study. The results reveals that the solar energy received on the collector surface is significantly affected by the choice of the tilt angle of the collector and for maximum energy collection the collector orientation for the considered location should be at a tilt angle of 12^o from the horizontal tilted toward the south. The result also shows significant improvement in the heat removal factor as the tube spacing is varied. Maximum heat removal factor occurred at a tube spacing of 15cm for tube diameter of 2cm. However for the tube diameter greater than 2cm, the maximum heat removal factor occurred at tube spacing greater than 15cm. The study also shows that the heat removal factor does not respond significantly to changes in the absorber plate thickness.*

Keywords: *Heat removal factor, Tube spacing, Tilt angle, Solar radiation, Sensitivity*

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I. INTRODUCTION

Satisfactory performance and reliability of a solar collector requires proper configuration and adequate sizing of its components so as to improve on the collector heat removal factor. Generally poor collector performance is an indication of an inherently low heat removal factor [1] . The collector heat removal factor is analogous to the effectiveness of a heat exchanger. Maximum heat transfer to fluid circulating in the collector occurs when the heat removal factor is maximized. To improve on the heat removal factor and hence the energy conversion efficiency of the flat plate collector, its components must be properly sized and arranged in such a way to maintain the temperature of the solar collector at temperature almost equal to the inlet fluid temperature [2]. The energy conversion efficiency of the flat plate solar collector has been found to depend on the heat removal factor. An improvement of the heat removal factor of the collector implies a corresponding improvement on the useful energy gain of the collector and therefore a better system thermal performance. [3] . This research is a parametric study to determine the effect and sensitivity of solar collector components sizes on the heat removal factor through a programme code developed in Matlab [4].

Many researchers have worked in this area. Yeh and Lin [5] investigated theoretically as well as experimentally the effect of collector aspect ratio on the energy collection efficiency of flat-plate solar air heaters for a constant collector area and different flow rates. They found that energy collection efficiency increases with mass flow rate and collector aspect ratio, and theoretical predictions agree reasonably well with the experimental results. Yeh and Lin [6] investigated the effect of parallel barriers on the energy collection efficiency of flat-plate solar air heaters. The barriers divide the air channel into parallel sub-channels or sub-collectors connected in series, and air flows through them in sequentially reversed directions. Thus the effect of increasing the number of parallel barriers is equivalent to increasing the collector aspect ratio. Manfrida [7], pointed out that to obtain higher exergetic or rational efficiency, the difference between the collector inlet and outlet temperatures should be small for low-performance collectors and higher for selective coated, evacuated or focusing collectors. Kar [8] proved that for maximum exergy output for the flat-plate solar collector for a particular mass flow rate, there is an optimum inlet temperature.

Lampert [9] , discussed in detail the properties of optical materials and coatings, which would predominantly increase the performance of the solar thermal system. He highlighted that for reducing the radiative loss of the top surface of the flat plate collector to the atmosphere, a low emittance coating material for glazing should be selected, which reduces the heat loss. Kaushika and Rulanantham, [10], investigated the transmittance-absorptance product $\tau\alpha$ of solar glazing. They developed a method to determine the transmittance-absorptance product for solar radiation using individual transmittance of cellular array and encapsulating cover. The developed method coincided with the value, calculated theoretically. Maatouk, [11] reported the effect of glass cover thickness at low and high temperature, and the radiative and conductive heat transfer for one and two glasses. He found that with the increase in thickness, the heat flux through the glass decreases, at high temperature. And at low temperature of the absorber, the steady heat flux through the single glass cover is higher that obtained with double glazing. At high temperature of the absorber, the double glazing is more suitable rather than single glass. At high temperature of the absorber the emissive power emitting from the black absorber is too high. Therefore, the double glazing reduces better the heat loss from the absorber to the surroundings than the single glass cover

II. THEORETICAL BACKGROUND

For the purposes of solar process design and performance calculations, it is necessary to know the radiation on the plane of the collector from measurements or estimates of radiation on horizontal surfaces. Models developed by Reindl *et al.*,[12] ; and Young [13] , take into account the circumsolar diffuse and horizontal brightening components on a tilted surface, giving a model referred to as the Hay–Davies–Klucher–Reindl (HDKR) diffuse model which estimates hourly total radiation on tilted surface as :

$$I_T = (I_b + I_d A_i)R_b + I_d(1 - A_i)\left(\frac{1+Cos\beta}{2}\right)\left[1 + f\sin^3\frac{\beta}{2}\right] + I\left(\frac{1-Cos\beta}{2}\right) \qquad (1)$$

A_i is an anisotropic index which is a function of the transmittance of the atmosphere for beam radiation expressed as:

$$A_i = \frac{I_b}{I_o} \qquad (2)$$

f is the square root of ratio of beam to total radiation expressed as:

$$f = \left(\frac{I_b}{I}\right)^{1/2} \qquad (3)$$

The geometric factor R_b which is the ratio of the average daily beam radiation on tilted surface to that on horizontal surface is a function of transmittance of the atmosphere, but Liu and Jordan ; Duffie and Beckman [1], suggested that it can be estimated by assuming that it has value which would be obtained if there were no atmosphere. For surfaces that are sloped toward the equator in the northern hemisphere, i.e. surfaces with surface azimuth angle =0^o, the geometric factor is expressed as [1] .

$$R_b = \frac{\cos(\emptyset+\beta)\cos\delta\sin\omega_s{}^1 + (\frac{\pi}{180})\omega_s{}^1\sin(\emptyset+\beta)\sin\delta}{\cos\emptyset\cos\delta\sin\omega_s + \frac{\pi}{180}\omega_s\sin\emptyset\sin\delta} \qquad (4)$$

Where: $$\omega_s{}^1 = min\begin{pmatrix}\cos^{-1}(-\tan\emptyset\tan\delta) \\ \cos^{-1}(-\tan(\emptyset+\beta)\tan\delta)\end{pmatrix} \qquad (5)$$

Where “min” means the smaller of the two in the bracket.

If I is the intensity of solar radiation, in W/m^2, incident on the aperture plane of the tilted solar collector having a collector surface area of **A** in m^2, then the amount of solar energy received by the collector is (:

$$Q_i = I.A \quad [14] \qquad (6)$$

However, Fabio, [3] has shown that a part of the incident radiation I is reflected back to the sky, another component is absorbed by the glazing and the rest is transmitted through the glazing and reaches the absorber plate as short wave radiation. Therefore a conversion factor $\tau\alpha$ only indicates the percentage of the solar rays penetrating the transparent cover of the collector and the percentage being absorbed. Basically, it is the product of the rate of transmission of the cover and the absorption rate of the absorber. Thus,

$$Q_i = I(\tau\alpha)A \qquad (7)$$

As the collector absorbs heat its temperature is getting higher than that of the surrounding and heat is lost to the atmosphere by convection and radiation. The rate of heat loss (Q_o) depends on the collector overall heat transfer coefficient U_L and the collector temperature.

$$Q_o = AU_L(T_c - T_a) \tag{8}$$

Thus, the rate of useful energy extracted by the collector (Qu), expressed as a rate of extraction under steady state conditions, is proportional to the rate of useful energy absorbed by the collector, less the amount lost by the collector to its surroundings expressed as [3].

$$Q_u = Q_i - Q_o = I(\tau\alpha)A - AU_L(T_c - T_a) \tag{9}$$

It is also known that the rate of extraction of heat from the collector may be measured by means of the amount of heat carried away in the fluid passed through it, as [1] :

$$Q_u = mC_p(T_c - T_i) \tag{10}$$

Equation 5 proves to be somewhat inconvenient because of the difficulty in defining the collector average temperature. It is convenient to define a quantity that relates the actual useful energy gain of a collector to the useful gain if the whole collector surface were at the fluid inlet temperature. This quantity is known as "the collector heat removal factor (F_R)" and is expressed as:

$$F_R = \frac{\dot{m}C_p}{AU_L}\left[1 - exp\left(-\frac{AU_LF'}{\dot{m}C_p}\right)\right] \tag{11}$$

Where *F'* is the collector efficiency factor expressed as [1] .

$$F' = \frac{1/U_L}{W\left[\frac{1}{U_L[D_i+(W-D_i)F]}+\frac{1}{C_b}+\frac{1}{\pi D_i h_{fi}}\right]} \tag{12}$$

h_{fi} $(W/m^2.K)$ is the internal fluid heat transfer coefficient and F is the standard fin efficiency for straight fins with rectangular profile, given as [1]

$$F = \frac{tan\,h\left[\frac{m(W-D_i)}{2}\right]}{\frac{m(W-D_i)}{2}} \tag{13}$$

Where :

$$m = \sqrt{\frac{U_L}{K\delta}} \tag{14}$$

An approximate relation for collector top loss coefficient (U_{top}) is given by [1] :

$$U_{top} = \frac{1}{\frac{N_G}{\frac{C}{T_{pm}}\left[\frac{T_{pm}-T_a}{N_G+f}\right]^e+\frac{1}{h_w}}} + \frac{[\sigma(T_{pm}{}^2+T_a{}^2)][T_a+T_{pm}]}{\frac{1}{\varepsilon_p+0.00591\,N_G h_w}+\frac{2N_G+f-1+0.133\,\varepsilon_p}{\varepsilon_g}-N_G} \tag{15}$$

Where:

$$f = \left(1 + 0.089h_w - 0.1166h_w\varepsilon_p\right)(1 + 0.07866N_G) \tag{16}$$

$$C = 520[1 - 0.000051\beta^2] \tag{17}$$

$$e = 0.430\left(1 - \frac{100}{T_{pm}}\right) \tag{18}$$

The energy loss through the bottom of the collector is made up of conductive loss to heat flow through the insulation and convection and radiation resistance to the environment. The magnitude of the conduction and radiation loss compared to the radiation loss is such that the radiation is negligible [1] . Thus the back loss coefficient is estimated as [1]

$$U_b = \frac{K_{bi}}{x_{bi}} \tag{19}$$

The edge loss coefficient is the ratio of the thermal conductivity of the insulation at edge to it thickness; times the ratio of the area of edge to the collector effective aperture area. Rai [15] recommends edge insulation of about the same thickness as the bottom insulation. The edge loss estimated by assuming one-dimensional sideway heat flow around the perimeter of the collector system is expressed as:

$$U_e = \frac{K_{ei}}{x_{ei}} \frac{A_e}{A} \tag{20}$$

The overall heat loss coefficient for a flat plate collector is composed of the top loss coefficient, the edge loss coefficient and the back loss coefficient. A relation for collector overall heat loss coefficient U_L is expressed as [1] :

$$U_L = U_t + U_e + U_b \tag{21}$$

A measure of a flat plate collector performance is the collector efficiency (η) defined as the ratio of the useful energy gain (Qu) to the incident solar energy over a particular time period:

$$\eta_C = F_R \tau\alpha - F_R U_L \left(\frac{T_i - T_a}{I}\right) \tag{22}$$

III. SYSTEM DESCRIPTION AND METHOD

The collector under consideration consists of a flat- plate absorber plate painted black, a transparent cover using glass of 4mm thickness to reduce top heat-losses from the absorber plate, tubes for the flow of the heat transfer fluid (water) to remove heat from the absorber plate, a heat insulating support to reduce heat loss from the sides and bottom of the collector, and a protective casing made from wood to prevent sides heat lost and to ensure the components are free from dust and moisture. The collectors design parameters and characteristics adopted here (table 1) for the parametric studies were calculated based on the typical meteorological solar data of Zaria (latitude 11.2^{o}N) as input in previous research work [16].

Table 1: System design parameters and characteristics used for the parametric studies.

Parameter	Description	Values
A_C	Collector area (m^2)	2.20
τ	Glass transmittance	0.93
U_L	Collector total lost coefficient (W/.m^2.K).	7.46
β	Collector slope (degrees)	12.0
N_R	Number of parallel collector risers	12
k	Thermal conductivity of absorber plate (W/m.K).	211.0
h_{fi}	Internal fluid heat transfer coefficient (W/m^2.K).	113.2
α	Plate absorptance	0.90

Source: Zwalnan *etal 2014*

Parametric studies to determine the sensitivity of the objective functions (heat removal factor and solar energy on tilted collector) to variation in the collector design parameters such as; tube spacing, number of glass covers, internal tube diameter, tilt angle, and absorber plate thickness were conducted based on the collector design parameters and characteristics of table 1.

To study the effect and sensitivity of the collector tilt angle β on the amount of solar energy I_T received on the collector surface, equations (1) to (5) are coded using Matlab programming language in such a manner that represent the flow of information to calculate the solar energy on the tilted collector varying the tilt angle from 0^o to 60^o. The effect is visualized through a plot between the tilt angle and the amount of solar energy received on the collector surface.

Similarly, programme codes were also developed using Matlab based on the appropriate equations of section 2 and system characteristics of table 1 to study the effects and sensitivity of variation in tube spacing, number of glass covers, internal tube diameter, and absorber plate thickness on the heat removal factor.. The results are all shown in the next section.

IV. RESULTS AND DISCUSSION

4,1 Collector tilt angle (β)

Figure 1 shows the effect of varying the collector tilt angle on the average daily solar radiation received on the collector surface, assuming a constant average daily horizontal solar radiation value 12.56MJ/m^2 for typical recommended day for the month of August (Zwalnan etal., 2014). From the figure, the solar radiation increases as the tilt angle increase from zero, reaching the maximum amount of 16.28 MJ/m^{2}day at a collector tilt angle of 12^o. Further increases in the tilt angle resulted in a decrease in amount of solar radiation received on the collector surface. the changes in the amount of energy as the tilt angle varies in Figure 1 reveals the solar energy received on the collector surface is significantly affected by the changes in the tilt angle of the collector and for maximum energy collection in the month of August, the collector orientation should be at an angle of 12^o from the horizontal tilted toward the south.

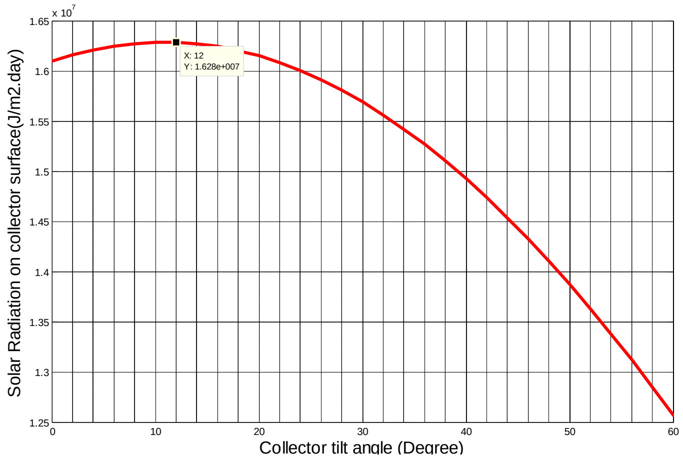

Figure1: Effect of varying collector tilt angle on the amount of solar energy received on the collector surface.

4.2 Collector tube diameter tube spacing.

Figure 2 shows the effect and sensitivity of varying the collector tube distance on the heat removal factor for different tube diameters (2.0cm to 3.5cm). The graph shows that the heat removal factor for all the range of tube diameters adopted for this study increases as the tube distance increases reaching a maximum and then decreases as the tube distance continue to increase. The graph also reveals that there is no significant difference in the heat removal factor for the tube diameters at tube spacing greater than 0.6m. The figure shows that the most significant improvement on the heat removal factor for tube diameters 0.02m, 0.025m, 0.03 and 0.035m adopted for the research occurs at 0.15m, 0.2m, 0.25m and 0.30m respectively. The graph also shows that the heat removal factor show significant improvement as the tube diameter decreases from 0.03m to 0.02m.

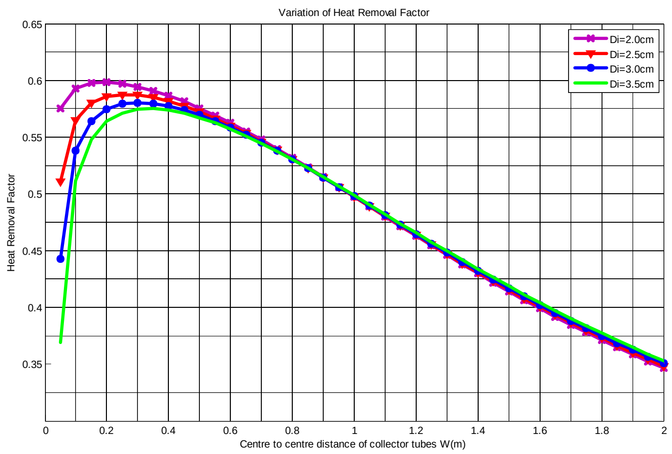

Figure 2 : Effect of variation of collector tube spacing (W) on the heat removal factor (F_R) for a range of tube internal diameter (Di).

4.3 Collector absorber plate thickness

Figure 3 shows the effect of varying the absorber plate thickness on the heat removal factor of a single glazing flat plate solar collector for given collector characteristics. The figure shows that the heat removal factor increases from a value of 0.6178 to 0.6195 as the plate thickness increases from 0.003m to 0.1m. This increase in F_R represents an increase of 0.31% in heat removal factor as the thickness of the plate is increased from 0.03m to 0.1m. This increase is considered very insignificant considering system cost in relation to the amount (0.31%) of improvement.

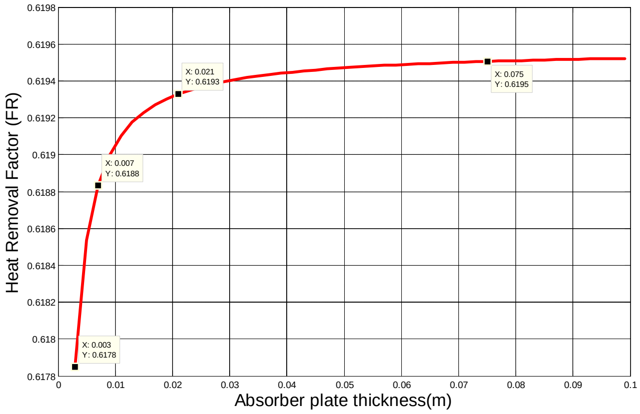

Figure 3: Effect of the variation of collector absorber plate thickness on heat removal factor for a given collector characteristics

4.4 Collector number of glazing

The effect of varying the number of glazing keeping other design parameters constant on the heat removal factor of the solar collector is shown in Figure 4. The figures reveal that as the number of glazing increases, the heat removal factor increases. The figure shows an improvement in the heat removal factor of the collector from 0.67 to 0.94 as the number of glazing increases from 1 to 20. When the collector glazing is increased from 1 to 2, the heat removal factor increased from 0.67 to 0.77, representing a significant increase of 15.44%. Additional increase in the number of glazing from 2 to 3 increased the heat removal factor from 0.77 to 0.82. This again represents a slight improvement of 6.45%. The improvement in F_R as the number of glass cover N_g increases can be explained from the fact that the overall heat loss coefficient U_L decreases with increase in N_g . The result from figure 4 implies that for the most significant and economical improvement, two number of glass cover should be adopted for most design.

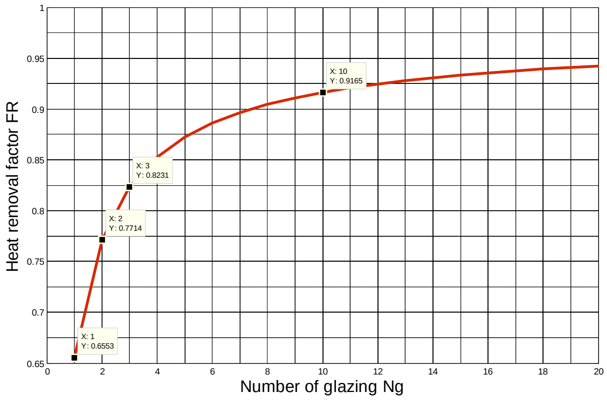

Figure 4: Effect of the variation of collector number of glass on heat removal factor (F_R) for a given collector characteristics

V. CONCLUSION

Parametric studies to determine the collector heat removal factor sensitivity to variation in its components sizes and configuration was conducted. The following are the conclusions drawn based on the results obtained and their analysis.

1. The solar energy received on the collector surface is significantly affected by the value of the tilt angle of the collector and for maximum energy collection in the month of August for this location; the collector orientation should be at an angle of 12^o from the horizontal tilted toward the south.
2. The tube spacing and tube diameter of the collector have great influence on the heat removal factor. For better performance of a solar collector the tube diameter and tube spacing should be kept at minimum values possible.
3. The heat removal factor is improved as the absorber plate thickness is increased, however the improvement is grossly insignificant for increase in thickness in the range of 0.01 to 0.1m

Nomenclature

A Collector area (m^2)
A_e Edge insulation area (m^2)
C_b Contact resistance (W/m.K).
C_p Fluid specific heat (KJ/kgK)
Di Inner tube diameter (m)
f Square root of ratio of beam to total radiation
F_R Heat removal factor
F' Collector efficiency factor
F Standard fin efficiency for straight fins with rectangular profile
F'' Collector flow factor
h_{fi} Internal fluid heat transfer coefficient ($W/m^2.K$).
h_w Wind heat transfer coefficient ($W/m^2.K$).
I Intensity of solar radiation, W/m^2
k Plate conductivity (W/ m ^oK)
K_{ei} Thermal conductivity of edge insulation materials (W/m ^oK)
N_g Number of glass covers
m Collector fluid mass flow rate ($kg/hr.m^2$)
Q_u Useful energy gain per unit time W
Q_i Collector heat input, W
Q_o Collector heat lost
T_{pm} Mean plate temperature (K)
Ta Ambient temperatures (K)
T_i Collector inlet fluid temperature (K)
T_C Collector average temperature
U_e Loss coefficient edge of collector per unit aperture area ($W/.m^2.K$).
U_o Loss coefficient of collector outlet and inlet pipe plus insulation ($W/.m^2.K$).
U_b Back loss coefficient ($W/.m^2.K$).
U_L Overall collector heat lost coefficient ($W/.m^2.K$).
U_t Collector top heat lost coefficient ($W/.m^2.K$).
W Tube spacing (m)
x_{ei} Insulation thickness at the edge (m)

Greek Symbols

$\overline{\tau\alpha}$ Transmittance-absorptance product
ρ_d Reflectance of the cover system for diffuse radiation
β Collector tilt angle
ε_g Emittance of glass (0.88)
ε_P Emittance of plate
σ Boltzmann constant
η_C Collector efficiency over a specified time horizon

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APPENDIX A

A PARAMETRIC STUDY PROGRAMME CODES WRITTEN IN MATLAB TO STUDY THE EFFECT AND SENSITIVITY OF COLLECTOR TILT ANGLE ON THE TOTAL RADIATION RECEIVED ON COLLECTOR SURFACE.

```
Clear all
Gsc=1367
LAT=11.2
%recomended days of the months
n1=[228]' %recommended Average day of the year
for i=1:length(n1)
beta=0:2:60
P=pi/180                              %factor for the conversion from radian to degree.
DLTA=23.45*sin(2*pi*((n1(i)+284)/365))            %formula for monthly declination
w_s=(acos(-tan(LAT*P).*tan(DLTA*P)))*1/P %monthly sunset angle
w_sb=(acos(-tan((LAT+beta)*P).*tan(DLTA*P)))*1/P
w_s1=min(w_s,w_sb) %minimum of line 8 and 9.
A1=86400*Gsc/pi;
A2=(1+0.033*cos(360*n1(i)*P/365));
A3=cos(LAT*P).*cos(DLTA*P).*sin(P*w_s)+(pi.*w_s*sin(P*LAT).*sin(P*DLTA)/180);
Ho=A1.*A2.*A3 %annual average extrateresterial solar radiation of location
H=[12.6584]*1e6 % monthly average solar radiation for the month of August
KT= H./Ho
g=0
Hd1=0.775+0.00606*(w_s-90)
Hd2=(0.505+0.0045*(w_s-90)).*cos((115*KT-103)*P)
Hd=(Hd1-Hd2).*H
Hb=H-Hd
Rb1=cos((LAT+beta)*P).*cos(DLTA*P).*sin(w_s1*P)
Rb2=(pi/180)*w_s1.*sin((LAT+beta)*P)*sin(DLTA*P)
Rb3=cos(LAT*P)*cos(DLTA*P)*sin(w_s*P)
Rb4=(pi/180)*w_s*sin(LAT*P)*sin(DLTA*P)
Rb=(Rb1+Rb2)/(Rb3+Rb4)
HT1=(H.*(1-(Hd./H)).*Rb+Hd.*(1+cos(P.*beta))/2)'
HT2=(H.*0.2.*(1-cos(P*beta))/2)'
HT=(HT1+HT2)'
plot(beta,HT,'linewidth',2)
xlabel('Collector tilt angle (Degree)','fontsize',12)
ylabel('Solar Radiation on collector surface(J/m2.day)','fontsize',12)
title('Variation of Solar radiation on collector surface','fontsize',12)
holdon
end
holdoff
```

APPENDIX B

A PARAMETRIC STUDY PROGRAMME CODES WRITTEN IN MATLAB TO STUDY THE EFFECT AND SENSITIVITY OF COLLECTOR TUBE DIAMETER AND TUBE CENTRE TO CENTRE DISTANCE ON THE HEAT REMOVAL FACTOR

```
Clear all
k_p=211
u_L=7.4
delta_p=6/1000
 Di=[0.5/100:0.5/100:4/100]'
H_fi=113.13
```

```
mdot=0.0035
 AC=(2.2)
for i=1:length(Di)
 W=(5/100:5/100:200/100)'
 m1=((u_L./(k_p*delta_p))).^0.5
 f1=(tanh(m1*(W-Di(i))/2)./(m1.*(W-Di(i))/2))
 f_prime1=1./u_L
 f_prime2=(W.*(1./((u_L.*(Di(i)+f1).*(W-Di(i)))+1./(pi.*Di(i).* H_fi ))))
f_prime=f_prime1./f_prime2
f_R1=(mdot.*Cp)./(AC.*u_L )
f_R2=(exp(-(AC.* u_L.*f_prime)./(mdot.*Cp)))
f_R=(f_R1.*(1-f_R2))
plot(W,f_R,'r','linewidth',2)
xlabel('Center to centre distance of collector tubes (M)')
ylabel('Heat Removal Factor')
title('Variation of Heat Removal Factor')
holdon
end
g=0
```

APPENDIX C

A PARAMETRIC STUDY PROGRAMME CODES WRITTEN IN MATLAB TO STUDY THE EFFECT AND SENSITIVITY OF ABSORBER PLATE THICKNESS ON THE HEAT REMOVAL FACTOR

```
k_p=211
u_L=7.4
  Di=0.5/100
H_fi=113.13
mdot=0.0035
 AC=(2.2)
delta_p=(5/1000:5/1000:35/1000)'
for i=1:length(delta_p)
 W=(5/100:5/100:200/100)'
 m1=((u_L./(k_p*delta_p(i)))).^0.5
 f1=(tanh(m1*(W-Di)/2)./(m1.*(W-Di)/2))
 f_prime1=1./u_L
 f_prime2=(W.*(1./((u_L.*(Di+f1).*(W-Di))+1./(pi.*Di.* H_fi ))))
f_prime=f_prime1./f_prime2
f_R1=(mdot.*Cp)./(AC.*u_L )
f_R2=(exp(-(AC.* u_L.*f_prime)./(mdot.*Cp)))
f_R=(f_R1.*(1-f_R2))
plot(W,f_R,'r','linewidth',2)
xlabel('Center to centre distance of collector tubes.W(m)')
ylabel('Heat Removal Factor')
title('Variation of Heat Removal Factor')
holdon
end
```

APPENDIX D

A PARAMETRIC STUDY PROGRAMME CODES WRITTEN IN MATLAB TO STUDY THE EFFECT AND SENSITIVITY OF NUMBER OF GLASS COVERS ON THE HEAT REMOVAL FACTOR

```
k_p=211
Di=1.5/100
hfi5=113.13
mdot=0.0035
AC=(2.2)
Ae5=0.511
abst=0.90
rho_g=0.2
```

```
beta5=11.2
hw5=10.4
V5=2.4
Tpm5=70+273
Ta5=311
Ep=0.95
zegma=5.67e-8
W=10/100
Ng=(1:1:10)'
f5=(1+0.089*hw5-0.1166*hw5*Ep )*(1+0.07866*Ng)
C =520*(1-0.000051*beta5^2)
e=0.430*(1-100/Tpm5)
U_top51=((C./Tpm5).*((Tpm5-Ta5)./(Ng+f5)).^e)+1./hw5
U_top52=(Ng./U_top51).^-1
U_top53=zegma*(Ta5.^2+Tpm5.^2).*(Ta5+Tpm5)
U_top5=(1./(Ep+0.00591*Ng*hw5))+((2*Ng+f5-1+0.133*Ep)/Eg)-Ng
U_topp=U_top52+U_top53./U_top5
U_back=Kb/tb
U_e5=Ke*Ae5./(t_e*AC)
 U_L5=U_topp+U_e5+U_back
 m5=((U_L5./(K_p*Delta_p))).^0.5
 F5=tanh(m5.*(W-Di)./2)./(m5.*(W-Di)./2) %fin efficiency
 F_prime51=1./U_L5
 F_prime52=W.*(1./((U_L5.*(Di+(W-Di).*F5))+1./(pi.*Di* hfi5 )))
 F_prime5=F_prime51./F_prime52
 F_R51=(mdot.*Cp)./(AC.*U_L5 )
F_R52=exp(-(AC.* U_L5.*F_prime5)./(mdot.*Cp))
F_R5=F_R51.*(1-F_R52) % heat removal factor
plot(Ng,F_R5,'linewidth',3)
xlabel('Number of glazing Ng','fontsize',18)
ylabel('Heat removal factor FR','fontsize',18)
g=0
```

Z.S Johnson “Effect and Sensitivity of Solar Collector Parameters on the Heat Removal Factor” American Journal of Engineering Research (AJER), vol. 6, no. 10, 2017, pp. 249-258.