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COMPARATIVE CRITIQUE ON THE PERFORMANCE EVALUATION OF A SOLAR – AIR HEATER FOR NATURAL UPDRAFT SOLAR SYSTEM

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ABSTRACT

This paper presents a mathematical model and discusses the influence of several convection heat transfer correlations and thermal efficiency schemes that have been commonly discussed in the literature regarding the performance of solar updraft system to choose the optimum forms conforming to the investigational result of the Spanish prototype. Hence, it is evident from the results of the study that the heat transfer coefficient correlation utilised by Kröger and Burger in 2004 was so close to the Manzanares experimental data. Also, the thermal efficiency equation used by Schlaich in 1996 is recommended to calculate the solar air heater collector's efficiency.

Keywords: renewable energy, solar chimney, heat transfer coefficient, mathematical modelling, and solar energy.

INTRODUCTION

The majority of the methods related to the generation of renewable energy directly involve the utilisation of solar radiation. One of the important progresses for future is the solar updraft tower power plant (SUTPP) which is comprised of the three main constituents: (1) a tower situated in the centre of the collector, (2) a solar collector (3) power conversion including one or multiple turbine generators [1]. Figure-1 shows the Schematic diagram of SUTPP.

Figure-1. Solar updraft tower power plant.

The Solar Collector (SC) is considered as the major component of the plant because of being responsible for collection and heating of air by absorbing solar radiation. The working principle of SUTPP involves generation of momentum as a result of high air temperature having low density. The momentum depends on the bouncy force towards the base of the tower, which rotates the wind turbine of the plant and generates electric power [2]. However, the ground of the plant (soil) has also been considered as the fourth part of the plant by few researchers [3].

Many researchers worked on improving and enhancing the SC, being the main part efficien and responsible for increasing the efficiency of the plant. In line with this, a thermal mathematical model was established by Mohammad [4], and Roozbeh [5] to evaluate the power output and the performance of the power plants. Based on a developed mathematical model, the simulation study by Atit [6], Fei Cao *et al.* [7], and Xinping *et al.* [8] inspected the efficiency of the power generating system. All the previous studies have incorporated different forms of correlations to calculate the convective heat transfer coefficient, while, others investigated the thermal efficiency equations of SC without comparing the mathematical results by using different equations. Thus, the objective of the present study was to develop a mathematical model of SC to analyse the effect of the different heat transfer correlations and different efficiency equations on the performance of SC. The model was intended to select the most suitable models conforming to the experimental result of the Manzanares prototype. The mass and energy balance principles within the different parts of the SC were used to establish the mathematical model.

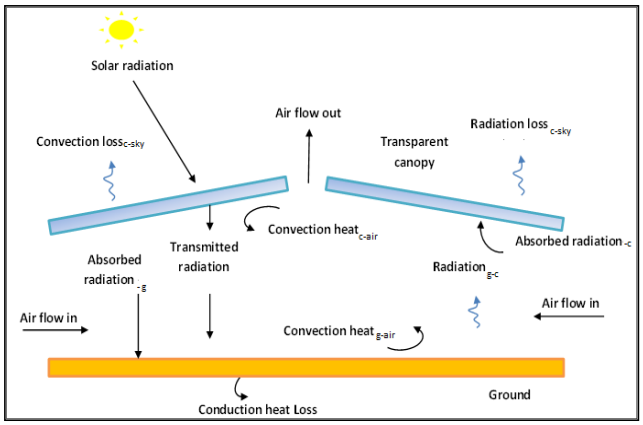

Figure-2. Solar conversion thermal network in the updraft solar collector.

MATHMATICAL MODELING

Figure-2 shows the thermal network of the solar energy conversion in the SC components, canopy and ground. By predict the performance of SC. Energy

balance, mass flow rate and the heat transfer by natural convection was calculated. Then, they are programmed to solved the predict of the canopy temperature T_c , ground temperature T_g , air temperature at the outlet of the collector $T_{air,o}$, and the air mass flow rate.

The mathematical model was developed with the following assumptions:

1- Steady state flow for the air inside the collector.
2- One dimensional flow towards the centre of the collector.
3- Temperature of the air at the inlet of the collector is equal to the ambient temperature.

Based on the thermal network and above assumptions, the energy balance equations of the SC can be presented as follows:

A. Energy balance equations for the canopy

$$S_1.A_C + h_{ra,g-c}.A_g(T_g - T_c) = h_{conv,c-a}.A_C.(T_c - T_{air}) + L_u.A_C \quad (1)$$

The solar radiation heat flux absorbed by the canopy, S_1 is given by:

$$S_1 = \alpha_C.I_r \quad (2)$$

L_u is the overall top heat loss from the canopy to the ambient. These losses are combined between the convection heat by wind $(q_{con,c-sky})$ and the radiation heat from the canopy to the sky $(q_{r,c-sky})$, it can be present as:

$$L_u = q_{con,c-sky} + q_{r,c-sky} \quad (3)$$

$$q_{con,c-sky} = A_C h_{con,c-amb}(T_c - T_{sky}) \quad (4)$$

The convective heat transfer coefficient $(h_{con,c-amb})$ due to wind can be calculated a cording by Duffine and Beckmann [9] as:

$$h_{con,c-amb} = 2.8 + 3.0\, u_w \quad (5)$$

Where,

u_w is the $Wind\ velocity\ (m/s)$ and the T_{sky} is the sky temperature given by [10] as:

$$T_{sky} = 0.0552\, T_{amb}^{1.5} \quad (6)$$

The radiation heat transfer coefficient from the outlet canopy surface to the sky $(h_{r,c-sky})$ is estimated as suggested by, Bansal *et al* [11]:

$$h_{r,c-sky} = \frac{\sigma\, \varepsilon_c (T_c + T_{sky})(T_c^2 - T_{sky}^2)(T_c - T_{sky})}{(T_c - T_{amb})} \quad (7)$$

The radiation heat transfer coefficient $(h_{ra,g-c})$ between the canopy and the ground it could estimate by Bansal [11] and Arce [12]:

$$h_{ra,g-c} = \frac{\sigma(T_g^2 + T_c^2)(T_g + T_c)}{(1/\varepsilon_g) + (1/\varepsilon_c) - 1} \quad (8)$$

The model can use various correlations depending on the previous studies to calculate the natural convective heat transfer coefficient between the canopy and the air $(h_{conv,c-a})$ inside the SC, as follows:

1- According to Kröger and Burger [13] and Pretorius [14]:

$$h_{con,c-air} = \left[\frac{0.2106 + 0.0026\, V_{in}\left(\frac{\rho\, T_{m\,c}}{\mu g (T_{air} - T_c)}\right)^{1/3}}{\left(\frac{\mu T_{m\,c}}{(T_{air} - T_c) g c_p K^2 \rho^2}\right)^{1/3}} \right] \quad (9)$$

The (T_{mc}) is the mean of temperatures of (T_c) and (T_{air}), and the (V_{in}) is the inlet air velocity into solar collector.

2- According to De Witt [15]

$$h_{con,c-air} = Nu_{c-air} \frac{K_{air}}{L_{g-c}} \quad (10)$$

The Nusselt number (Nu) calculated depending on the flow type.

For laminar flow ($R_a < 10^9$)

$$Nu_{c-air} = \frac{0.68 + (0.67\, R_a^{1/4})}{[1 + (0.492/Pr)^{9/16}]^{4/9}} \quad (11)$$

For turbulent flow ($10^9 < R_a$)

$$Nu_{c-air} = \left\{ 0.825 + \frac{0.387\, Ra_{Lg-c}^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}} \right\}^2 \quad (12)$$

Where Nu is the Nusselt number ,L_{g-c} is the spacing between the canopy and ground , R_a is the Rayleigh number given as the product of Prandtl number (Pr) and Grashof number (Gr).

$$Pr = \frac{\mu\, c_p}{K_{air}} \quad (13)$$

$$Ra = Gr.Pr = \frac{g.cos\theta.\beta(T_c - T_{air})L_{g-c}^3}{v\, \alpha_c} \quad (14)$$

Where:

β = Volumetric thermal expansion coefficient of air $= \frac{1}{T_{air}}$ $for\ ideal\ gas$

θ = Inclination angle of the canopy.

3- According to Churchill and Chu [16]

For the upper and lower horizontal heated plate surface.

$Nu_{c-air} = 0.54\, R_a^{0.25}$ for $2x10^4 < R_a < 8x10^7$ (15)

$Nu_{c-air} = 0.15\, R_a^{0.33}$ for $8x10^7 < R_a < 8x10^{11}$ (16)

And by using Equations (14), (10), we can estimate the convective heat transfer coefficient ($h_{con,c-air}$).

B. Energy balance equations for the airflow inside collector

The energy balance for the air flow between the canopy and the ground can mathematically be expressed as:

$$Q_{air} = [(h_{con,c-air} A_c(T_c - T_{air})] + [h_{con,g-air} A_g(T_g - T_{air})] \quad (17)$$

Where (Q_{air}) is the heat gain carried out by the air and it is presented as:

$$Q_{air} = \dot{m}_{air} . cp_{air} . (T_{a,o} - T_{a,in}) \quad (18)$$

The mean air temperature inside the SC (T_{air}) is compensated by 0.5 ($T_{a,o}$ +$T_{a.in}$) where is $T_{a.in} = T_{amb}$

C. Energy balance equations for the ground

$S_2.A_g = [(h_{ra,g-c} . A_g . (T_g - T_c)) + (h_{conv,g-a} . A_g(T_g - T_{air})) + (L_d . A_g . (T_g - T_{amb})]$

The solar radiation heat flux absorbed by the ground , S_2 is given by:

$$S_2 = \tau . I_r . \alpha_g . \quad (19)$$

L_d = is the overall down heat loss from the ground to the ambient , it can be present as:

$$L_d = \frac{K_g}{Z_e} \quad (20)$$

The convective heat transfer coefficient between the ground and the air($h_{conv,g-a}$), could be determined by using the Equations (9-17), but replaced the temperature and properties of canopy on the ground.

D. Mass flow rate inside the collector

The mass flow rate of the air at the collector could be estimated according to Continuity equation, as follows:

$$\rho_{in} v_{in} A_{in} = \rho_o v_o A_o \quad (21)$$

And we can find the outlet velocity from the collector ($V_{o,coll}$) by using:

$$V_{o,coll} = \frac{\dot{m}}{\rho_{air,o}\, A_o} \quad (22)$$

Where the (ρ_{air}) is the air density and calculated depended on the Volume coefficient of expansion:

$$\rho_{air} = \rho_{amb}[1 - \beta(T_{air} - T_{amb})] \quad (23)$$

$$\beta = \frac{1}{T_{air}} \quad (24)$$

E. The updraft solar collector performance

This model will be used various solar collector efficiency forms depended on the previous studies, as present as blow:

1- According to Schlaich (1996) [17], the collector efficiency defined is a ratio between the heat gain by the air to the available heat from the sun radiation , which is defined as :

$$\eta_c = \frac{Q_{air}}{A_c . I_r} \quad (25)$$

2- According to Schlaich (1996) and .Joneydi (2015), the collector efficiency based on effective absorption coefficient of the collector ($\alpha 1$) and heat loss coefficient, obtained as following:

$$\eta_c = \alpha 1 - \frac{U_t . \Delta T_O}{I_r} \quad (26)$$

It should be noted that ($\alpha 1$) in Equation. (27) it is mean the transmittance - absorption coefficient ($\tau . \alpha_g$), ΔT_O it is the stands for temperature difference between surface (heat absorbent and the ambient temperature) = ($T_g - T_{amb}$) , and U_t is heat loss Coefficient from solar collector and calculated as follows [18] [19]:

$$U_t = \frac{(L_u + L_d) \times [(h_1 \times h_2) + (h_1 \times h_{ra,g-c}) + (h_2 \times h_{ra,g-c})] + [(L_u \times L_d) \times (h_1 + h_2)]}{[(h_1 \times h_{ra,g-c}) + (h_2 \times L_u) + (h_2 \times h_{ra,g-c}) + (h_1 \times h_2)]}$$

where h_1 is the convective heat transfer coefficient between the canopy to the inside is air , and h_2 is the convective heat transfer coefficient between the ground to inside air.

3- According to Atit Koonsrisuk (2013) [20], the collector efficiency obtained as following:

$$\eta_c = \frac{Q_{air}}{A_c . q''} \quad (27)$$

and the (q'') it is the insolation (W/m^2) and defined as:

$$q'' = \alpha 1 . I_r - U_t . (T_{a,o} - T_{a,in}) \quad (28)$$

RESULTS AND DISCUSSIONS

This mathematical model illustrates the parameters which describe the performance of SC. The analysis involved the calculation of the temperature of the canopy (T_c), the temperature of the ground (T_g), the temperature of the air inside the collector at the exit (T_{air}), and mass flow rate of the air ($\dot{m}$) . the values were evaluated by arranging the model in form of a matrix,and the program , MATLAB was used. The results of the mathematical model were confirmed by comparing with

the experimental data of the Spanish prototype .The main geometrical dimensions of the Spanish prototype are listed in Table-1. The properties of the solar collector materials are given in Table-2

Table-1. Geometrical dimensions of the Manzanares prototype [21].

Area of canopy	46759.5 m^2
Diameter of the ground	244 m
Spacing between ground and canopy	2 m
Height of chimney	194.5 m
Area of chimney	78.5 m^2

Table-2. Properties of the SC parts of the Manzanares prototype [21].

Emissivity of the canopy	0.87
Emissivity of the ground	0.9
Absorptivity of the canopy	0.15
Absorptivity of the ground	0.9
Transmittance of the canopy	0.85
Thermal conductivity of ground	1.83 w/m.k

The use of different heat transfer coefficient correlations and variable thermal efficiency concepts utilised in the study are analysed as follows:

Heat transfer coefficient analysis

The results of the mathematical model in the study, which used the three different correlations to calculate the heat transfer coefficient from the ground to the air inside the collector can show in Table-3. The study calculates the heat transfer from the ground to the air, because this parameter the primary source to heat the air. It is evident from the results in Table-3 that the model presented by Kroger and Burger provides quite similar results as the manzanares readings. Hence, it is proved that, this correlation represents authentic equation to be used in the calculation of the heat transfer coefficient for the ground and canopy of the collector in SC. Morever, the parameters in the equation which affected the results and values of the heat transfer coefficient are shown in Table-3. It is evident the correlation used by De Witt and Churchil, provided different air temperature results compared with Manzanares readings and Kroger and Burger model, which give close results. Therefore, it can be concluded that different correlations provide different values of heat transfer coefficients and variant temperature difference (ΔT) between the inlet and outlet collector.

Table-3. Results of heat transfer coefficient and (ΔT) by proposed mathematical models

Correlation models	**Kroger and Burger**	**De Witt**	**Churchill and Chu**
h ($w/m^2.K$)	7.1	6.3	6.6
ΔT (K)	21.6	18.2	19
Percentage different of ΔT (%) comparing with Manzanares (ΔT=20 K)	0.5	9.89	5.26

Performance analysis for collector

In this part of study, the efficiency of the collector was calculated by using the three different equations depending on previous studies. The experimental readings of Manzanare's solar collector listed in Table-4 have been used as case study. The collector efficiency by using Schlaich's (1996) scheme (Eqn.26) was 33%. In this equation the collector efficiency is a function of the area of the collector and the solar intensity, which means that increase in the radius, area and solar intensity of the collector provides higher heat gain (Q) to the collector. Whereas, increasing (Q) means the temperature and mass flow rate of the air inside the collector will also increase and give higher collector efficiency.

Table-4. Data of Manzanres prototype for 1st September 1989 taken from [20], [21].

Solar radiation (W/m^2)	1017
Ambient temperature(℃)	18.5
Outlet air collector temperature(℃)	38
Collector absorption coefficient , $\alpha 1$	0.65
Collector loss coefficient ($W/m^2.k$),	15

After using Joneydi (2015) scheme (Equation.27), to calculate collector efficiency for same case study and collector ground (330 K), the result was found to be 9 %, depending on the temperature of collector ground, the solar intensity and the properties of the ground, and not on the plant geometry. As a result, no temperature difference between the ground and the ambient temperature or no losses were found, which shows a maximum value of the collector efficiency. However, this claim is not practically logical, because the increase in an area of solar collector means more heat

transfer storage to the collector ground, higher ground temperature and more heat gain for the air inside the collector, which increases the efficiency of the collector.

Also, the Equation.30, suggested by Atit (2013), having the same case steady showed the value of efficiency of the collector to be 95% which is not logical. In this equation, the collector efficiency is not a function of the solar heat gain, rather it is a function of collector area as mentioned by Atit [20]. Hence, it can be concluded that increase in the area of collector causes decrease in collector efficiency, but this claim is not physically possible, as the solar intensity is still affected due to the air heat gain and the value of (q''), still depends on the solar intensity as in Equation 30. Furthermore, increase in solar intensity means (q'') will be higher, and at the same time, the heat gain (Q) also increases. In Equation 30, the maximum value of (q'') is reached when the losses are equal to zero or (ΔT) equal to zero.

CONCLUSIONS

This paper investigated, analysed and compared the effect of different convective heat transfer correlations using Kroger- Burger, De Witt and Churchill heat transfer models for laminar and turbulent natural flow. Also, solar collector efficiency models presented by Schlaich, O. Joneydi and Koonsrisuk were evaluated by mathematical modelling to determine the optimum models, which yielded results suiting with the Manzanares solar collector project readings. It was concluded that the correlation results provided by Kröger and Burger model were close to the Manzanares readings for temperature difference (ΔT). Hence, this equation can be successfully recommended as an excellent correlation to be used in the mathematical models used for the calculations of SC.

Moreover, it is clear from the above analysis that the collector efficiency is a function to multiple parameters (solar intensity, collector geometry, and type of heat storage material), which means that equation used by Schlaich [17], is recommended equation to represent and calculate the thermal collector efficiency.

NOMENCLATURE

A_c,A_g	Areas of canopy and ground respectively	(m^2)
C_p	Specific heat capacity of air	$(W/m^2.k)$
k_a	Thermal conductivity of air	$(W/m.k)$
k_g	Thermal conductivity of round	$(W/m.k)$
g	Gravitational constant = 0.981	(m/sec^2)
I_r	Intensity of solar radiation	(W/m^2)
T_{amb}	Ambient temperature	(K)
L	Spacing between ground and canopy	(m)
Z	Depth of the ground	(m)

Greek symbols

μ	Dynamic viscosity	
v	Kinematic viscosity	(m^2/sec)
α_C, α_g	Absorptivity of the canopy and ground respectively	
$\varepsilon_c, \varepsilon_g$	Emissivity of the canopy and ground respectively	
σ	Stefan- Boltzmann constant	
τ	Transmittance of the canopy	

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