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MATHEMATICAL MODELING TO EVALUATE THE PERFORMANCE ENHANCEMENT OF SOLAR UPDRAFT POWER PLANT BY EXTERNAL HEAT SOURCE

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ABSTRACT

Solar Chimney Power Plant (SCPP) is considered one of the stunts applications for the use of solar energy. The problem lies in this type of power plant in the low rate of power generation on cloudy days and at night. In this study, a mathematical model has been introduced, of a hybrid solar chimney power plant (HSCPP) generating power using solar energy and external heat source. The external heat source is a hot flue gases flows in channels called "Thermal Enhancing Channels (TECh)" located underneath the canopy. Air, as working fluid in the collector, extracts heat from the channel surface, during the night. This study also proposed a mathematical model to evaluate the performance of HSCPP. The results showed that at solar intensity 1000 W/m^2, the temperatures of air inside the collector increased 5.88 % after using the external heat source when the wall temperature of the TECh was 100°C. Also the power generation has been enhanced by 23.0%.

Keywords: energy recapture, SCPP, solar collector, hybrid energy system, flue gas.

INTRODUCTION

Use of renewable energy has become essential these days, being clean and inexhaustible energy .The solar chimney power plant (SCPP) system (Figure-1) represents a large-scale of power plant which used solar energy. These types of plants are uncomplicated and contain key parts simple free of complexity. These parts are: collector (green house), chimney and turbine. The working principle in this type of plants depends on the heating air inside the collector by the influence of solar radiation energy where the air acquires kinetic energy which leads to a push toward collector center and then to the chimney, who in turn contains a turbine. Hot air energy transformed into mechanical energy operates to rotate the turbine and generate electricity. Began thinking about the design and implementation of this type of plant by Schlaich [1] who proposed the design and implemented by 1982 in Manzanares in Spain with the help of German and Spanish governments, the plant continued to work until 1989. Remains the biggest challenge in using this type of plants is low-lying efficiency and low processing power sudden at night.

One of the main important parts at the (SCPP) is the collector being responsible for the processing the plant hot air impact of solar radiation. So a lot of researchers try to focusing on the development of this part and try to enhance it to make the system operates with high efficiency and for the duration of the night. Bernardes and Zhou [2] analysed the sensible heat storage materials when used a water bugs in the collector of the SCPP. They carried out a simulation to study the thermal stratification effect. In this study they mentioned that the thicker water bags are more suitable regarding heat exchanges than the thinner one. Zuo *et al.* [3] proposed a solar chimney power generation compound with synthetic of sea water desalinization system. After analysed the system theoretically, the proved that the proposed system can improve the solar energy utilization. Ming *et al.* [4] analysed numerically the air flow and heat transfer characteristics for SCPP with energy storage layers. The results of this work showed that the efficiency of the plant is improved when the solar radiation increased and in the same time the temperature gradient of the storage media increased and that caused increase in thermal heat losses by the storage layers.

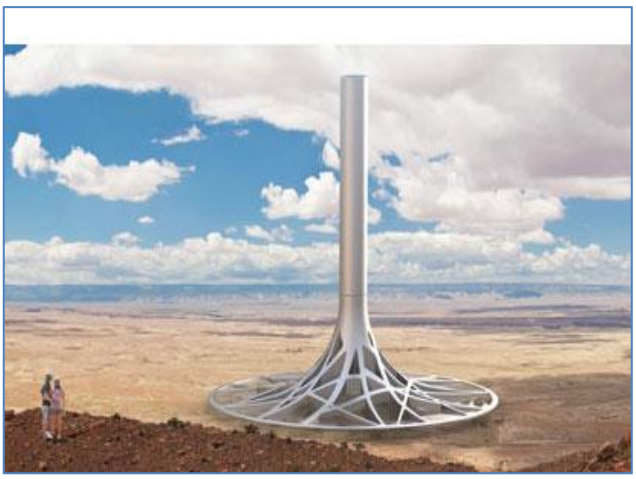

Figure-1. Solar chimney power plant.

Pretorius and Kroger [5], employed models of SCPP with Limestone, Sandstone and Granite as heat storage materials, also used better quality of the glass for the collector to increase the solar intensity. They showed that Limestone, Sandstone and Granite have same energy storage performance. Li *et al* [6] proposed a comprehensive mathematical model to evaluate the performance of the SCPP and validated the model with experimental work. The important results in this study

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were the limitation in collector radius may exist while such this is not with the chimney.

Zheng *et al* [7] analysed the performance of the SCPP by carried out numerical simulations. Different heat storage materials were used to study its effect on the power output of the plant with different solar radiations. In this study the researcher showed that the soil and gravel were good heat capacity can store more energy in sunny day. Al-Kayeim *et al* [8], showed how can benefit from the waste heat or thermal energy which came from the flue gases to enhance the performance of the SCPP. These gases contain more than 50% of the fuel thermal energy as Al-Kayeim told. They designed and fabricated solar chimney model. They achieved air channel with absorber plate and flue gas channel to enhance the flow inside the air channel. Chikere *et al* [9] suggested how can be enhanced heat transfer and the performance of the SCPP by using flue gas waste heat. They used a new model which contains of a flue gas channel, absorber plates, green house air flow channel bounded at the bottom with the absorber plate and on top with transparent cover. Flue gas exit chimney and air exit chimney also carried out.

ENHANCING THE COLLECTOR OF SCPP BY EXTERNAL HEAT SOURCE

One of the weak points in this type of plants is not possible to sustain the generation of electrical energy with high efficiency during the night, cloudy or rainy days. To overcome this problem, we introduced (hybrid solar chimney power plant) as a proposed model, because it would not only increase the power plant efficiency during the day, rather it would provide power even during no sun or at night. The technique in this model is used the flue gases or exhaust gases from a nearby gas turbine power plant or from any external waste heat source and convert the waste thermal energy in this flues or exhaust to a thermal energy in the solar chimney power plant (SCPP) and this consider a new form of waste heat energy recapture and perfect exploitation way. In this way, we will increase the thermal energy of the air flow in the solar collector of the plant by using hollow rectangular channels, which called (Thermal Enhancing Channels, (TECh)) below the collector cover. The exhaust or flue gases will push through the channels by means the pipe holes and these gases which passing through the channels causing, heat transfer from walls of the channels to the air flowing below the collector (green house) and that mean increasing in the outlet collector air temperature, which cause increasing in performance of the plant (see Figure-2).

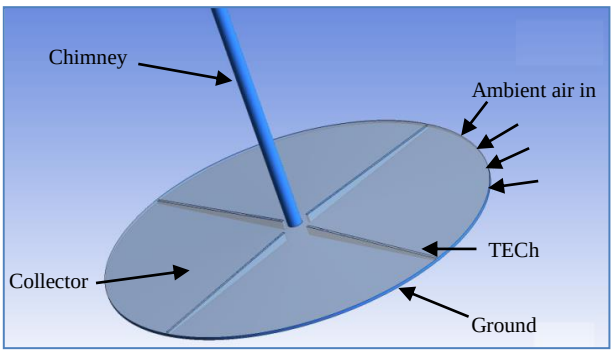

Figure-2. Proposed model dimensions.

The proposed model in the present work provided by Perspex glass sheet as a canopy with 4 mm thickness, PVC plastic pipe as a chimney and the ground is assumed black pebble to increase the radiation absorption. Four channels which made by galvanized steel installed inside the collector. The objective of this study is presented a theoretical model to predict the temperature distribution at the canopy, ground and the air inside the collector with assumed a constant and equalized wall temperature for channels. This prediction is very important to evaluate the performance of the proposed model.

MATHEMATICAL MODELING FOR PROPOSED MODEL

Mathematical modelling for the proposed model (HSCPP) can be achieved by energy balance as shown in Figure.3.

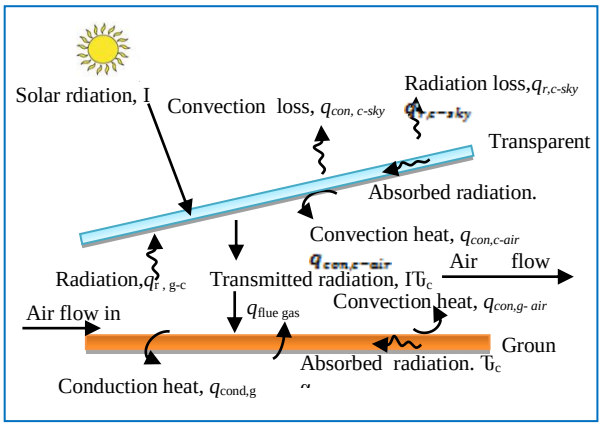

Figure-3. Energy balance for the collector of (HSCPP).

The mathematical model has been developed with the following assumptions:

1- Steady state flow for the air inside the collector of HSCPP.
2- One dimensional flow towards the centre of the collector & the chimney.
3- Temperature of the air at the inlet of the collector is equal to the ambient temperature.
4- Friction losses in the collector and the chimney were neglected.
5- The temperature of the external heat source is constant and equal in all (TECh).

Energy balance for the canopy

(Incident Solar radiation) + (Solar radiation from the ground to the canopy) = (Convection heat transfer from the canopy to the air flow inside collector) + (heat losses from the canopy to the ambient)

$$R_c A_c + q_{r,\,g\text{-}c}\, A_g = q_{con,\,c\text{-}air}\, A_c + q_{loss}\, A_c \quad (1)$$

$$[I\alpha_c A_c]+[h_{r,g\text{-}c}\, A_g(T_g\text{-}T_c)]=[h_{con,\ c\text{-}air}\, A_c(T_c\text{-}T_{air})]+[L_u\, A_c(T_c\text{-}T_{amb})] \quad (2)$$

The solar radiation on the out surface of canopy for the collector is:

$$R_c = I\alpha_c \quad (3)$$

I, is the total solar incidence on the surface of the canopy and L_u is the overall upper heat loss coefficient from the outer surface of the canopy to the ambient which presented according to [10] as:

$$L_u = h_{con,c\text{-}amb} + h_{r,c\text{-}sky} \quad (4)$$

$h_{con,c\text{-}amb}$ is the convection heat transfer coefficient to the wind and can written as follows:

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

Also the radiation heat transfer coefficient $h_{r,c\text{-}sky}$ can write as [10]:

$$h_{r,c-sky} = \frac{\sigma\varepsilon_c (T_c + T_{sky})(T_c^2 - T_{sky}^2)(T_c + T_{sky})}{(T_c - T_{amb})} \quad (6)$$

$$T_{sky} = 0.0552 T_{amb}^{1.5} \quad (7)$$

$h_{r\ ,g\text{-}c}$ is the radiation heat transfer coefficient from the ground to the canopy and can be written as:

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

The convection heat transfer coefficient $h_{con\ ,c\text{-}air}$ from the canopy to the air inside the collector can be written as:

$$h_{con,c-air} = Nu_{c-air} \frac{K_{air}}{L_{c-g}} \quad (9)$$

$Nu_{c\text{-}air}$ is the Nusselt number and K_{air} is the thermal conductivity for the air. $L_{g\text{-}c}$ is the average distance between the ground and the canopy.

The Nusselt number (Nu) calculated by using the following correlation according to Incropera [11] for the flow between the two parallel plates with inclined, where the inclined angle of the plates in this model is zero:

$$Nu_{c-air} = \left(0.825 + \frac{0.387 Ra_{L,g-c}^{1/6}}{[1+(0.492/\Pr)^{9/16}]^{8/27}}\right)^2 \quad (10)$$

Ra = Rayleigh number.

Energy balance for the ground

(Solar radiation) = (Convection heat transfer from the ground to the air flow inside collector) + (Solar radiation from the ground to the canopy) + (conduction heat to the ground)

$$R_g A_g = q_{con,\,g\text{-}air}\, A_g + q_{r,\,g\text{-}c}\, A_g + q_{cond}\, A_g \quad (11)$$

$$[I\,\tau_c\alpha_g A_g]=[h_{con,\,g\text{-}air}\, A_g(T_g\text{-}T_{air})]+[h_{r,g\text{-}c}\, A_g(T_g\text{-}T_c)]+[L_d\, A_g(T_g\text{-}T_{amb})] \quad (12)$$

R_g is the solar energy which absorbed by the ground:

$$R_g = I\tau_c\, \alpha_g \quad (13)$$

$h_{con\ ,g\text{-}air}$ is the convection heat transfer coefficient between the ground and the air inside the collector.

Energy balance of the air inside the collector

(Energy gain by the air inside the collector)= (convection heat transfer from the canopy to the air) + (convection heat transfer from the ground to the air) + (convection heat transfer from the flue channel to the air)

$$[Q_{collector}]=[h_{con,c\text{-}air}\, A_c(T_c\text{-}T_{air})]+[h_{con,g\text{-}air}\, A_g(T_g\text{-}T_{air})] +2n\times[h_{con\ ,channel\text{-}air}\, A_{channel}\,(T_{channel}\text{-}T_{air})] \quad (14)$$

$$[\dot{m}C_p(T_{air,i}\text{-}T_{air,o})]= [h_{con,c\text{-}air}\, A_c(T_c\text{-}T_{air})]+[h_{con,g\text{-}air}\, A_g (T_g\text{-}T_{air})] +2n\times [h_{con,\,channel\text{-}air}\, A_{channel}\,(T_{channel}\text{-}T_{air})] \quad (15)$$

In equation (15), it could be realized that the external heat source which added to the collector, where 2n represent the number of channels in the proposed model which have two faces, also $h_{con,channel\text{-}air}$ is the heat transfer coefficient from the surface of channels to the air inside the collector. This equation is considering the main equation which describes the energy gain by the air (working fluid) inside the collector.

Generally, the above equations (2) to (15) can be written in matrix form:

[A] [T] = [C]

The matrix was solved by using MATLAB code to find the temperature vector T_c, T_g, $T_{air,o}$ as follows:

$[T] = [A]^{-1}\,[C]$

The mass flow rate of the air at the collector could be calculated according to Schlaich [15], as follows:

$$V_{ch} = C_d \sqrt{2gH_{ch}\frac{T_{air} - T_{amb}}{T_{amb}}} \quad (16)$$

C_d = 0.6 according to Ong [12].

Now we can find ($\dot{m}$) by using:

$$\dot{m}=\rho_{air} A_{ch} V_{ch} \quad (17)$$

In all previous equations and according to [12]:

$$T_{air} = \left(\frac{T_{air,o} - T_{amb,i}}{2} \right)$$

T_{air} = Temperature of the air inside the collector
$T_{air,i} = T_{amb}$

Assuming that the temperature of the flue gas channels $T_{channel}$ is constant, we can arrange the equations (2), (12), and (15) in matrix and solve it by iteration method to find the unknowns T_c, T_g, $T_{air,o}$ and $\dot{m}$.
According to Bernardes [13]:

$$p_{tur,\max} = C\dot{m}H_{ch} \left(\frac{T_{air} - T_{amb}}{T_{amb}} \right) \quad (18)$$

$P_{tur,\ max}$ = Maximum power from the air gained by the turbine

Depending on the literatures, the constant (C) have different values between 0 and 1 [15], and this value is ideal:
C= 2/3

$$P_{ele} = \eta_{Turbine}\, \eta_{Generator}\, P_{tur,\ max} \quad (19)$$

P_{ele} = Maximum electric power generated by the turbine
$\eta_{Turbine}$ = Turbine efficiency=0.8
$\eta_{Generator}$ = Generator efficiency = 0.95

RESULTS AND DISCUSSIONS

Validation of the model

The theoretical model is validated by solving the matrix by MATLAB code for Manzanares solar chimney power plant dimensions and the results gave a good agreement with measured data as in table-1[14], where, the temperature raise measured inside the collector (ΔT= 19 ^{o}C) while in theoretical model in this work the value was (19.82 ^{o}C).

Table-1. Measured data from Manzanares and dimensions [14] for Manzanares and presented model.

I (W/m^2)	T_{amb} (K)	H_{col} (m)	H_{ch} (m)	R_{col} (m)	R_{ch} (m)
890	293	2	194.6	122	5

Table-2 shows the comparison between the measured data of Manzanares power plant and the calculated properties for the same plant by using the presented theoretical model in this study without external heat source. The percentage of error between the measured and predicted properties for the plant is calculated. The results of the mathematical model were in good agreement with measured data, and with small percentage error which belongs to the assumptions in mathematical model and the correlation which used to calculate the heat transfer coefficients in the model, these percentage errors is acceptable. Also the power output in Table-2 is calculated by using Equation. (19).

Table-2. Calculated and measured data for SCPP in Manzanares.

Parameters at I=1000 W/m^2	Present mathematical model	Manzanares prototype	Error %
ΔT(K)	19.82	19	4.3
V_{ch} (m/s)	9.64	9.1	5.6
Power output (kW)	50.35	45.6	9.4

Results of the proposed model

The results in the present work predicted the temperatures of the ground, canopy and the air inside the collector without and with External Heat Source (EHS) for the proposed model in Figure-2. As expected, temperatures increase as solar intensity increases. By the results that have been calculated, the temperatures of the ground were always higher than canopy and the air flow, and the temperatures of the canopy were higher than mean air temperatures when the solar intensity exceeded 400 W/m^2 approximately, but before that the value of temperatures were lower. The temperature of the channels assumed constant and equal to 100 ^{o}C. Same behavior of the temperature distribution, when the system integrated with external heat source by using (TECh), and the temperatures were higher for the canopy, ground, and air flow as shown in Figures- 4 and 5.

At 1000 W/m^2 solar intensity, the results of the mathematical model showed that when the temperature of the TECh was 100 ^{o}C for four channels, the temperature of the air inside the collector increased about 2.5-3 ^{o}C, and that gave the working fluid an extra thermal energy (see Figure-5) caused an increase in the power output of the model in this study by 23.1%. In the mathematical model, it was noted that (see Figure-6) the increasing in the electric energy produced by proposed model were few when it begins to solar radiation increase after the value of the solar intensity 1000 W/m^2 compared to the value of solar radiation between (800 and 1000 W/m^2) and this is confirm in this study that the increase in solar radiation have its determinants and cannot be considered the increase are always positive. Also, the influence of EHS

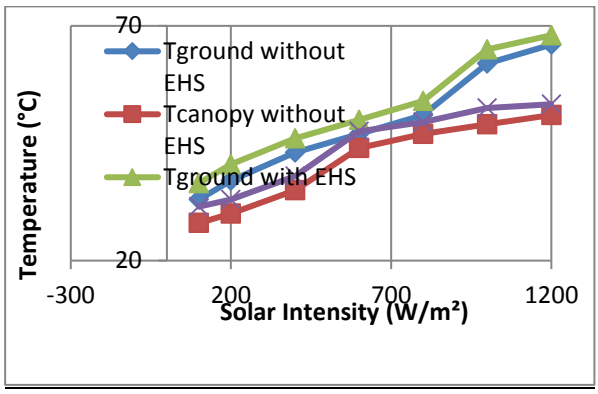

Figure-4. Predicted temperatures of the collector for proposed model with and without (TECh) at various solar intensities. $T_{channels}$ =100 °C.

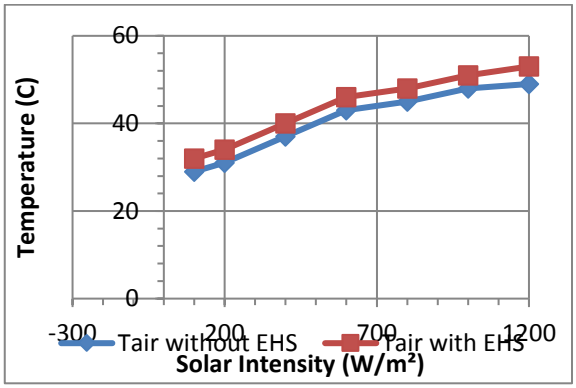

Figure-5. Predicted temperature of the air inside the collector for the proposed model with and without (TECh) at various solar intensities. $T_{channels}$ =100 °C.

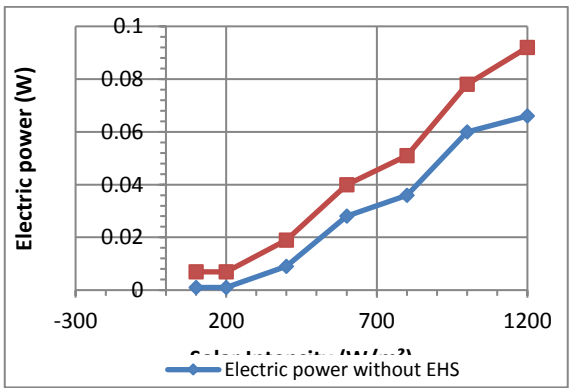

Figure-6. Electrical power generation with and without (TECh) for the proposed model. $T_{channels}$ =100 °C.

make the power output of the plant highly increased when the solar intensity exceed (600 W/m^2) and above.

The results of this model showed that the effect of external heat source highly appears when the intensity was low and then started decreased after the value of solar intensity reached to (400 W/m^2), means clearly that the effect of EHS be significant in the case of the large decline in solar radiation and in the morning hours of the first as well as in the atmosphere cloudy or rainy and in the atmosphere of the night.

It is important to make it clear in this study that the proposed mathematical model focused on enhancing the collector by using (TECh) and didn't consider the effect of the mass flow rate of the EHS inside these channels.

CONCLUSIONS

The temperature distribution for the collector of the SCPP integrated with external heat source has been predicted, and as a result the predicted temperatures, mass flow rate and electrical power generation for the proposed model have been calculated. All results showed improving in performance of the proposed model after add (TECh) especially when the solar intensity low, which means that, this model can give more power if we used the waste heat or any external heat source inside the proposed channels. When the solar intensity was (1000W/m^2), the percentage of the increasing for the temperature of the air inside the collector reached to 5.88 % and for electrical power 23.1% theoretically. The effect of the solar intensity started after 2-3 hours from the sun shine time, where the solar intensity reached to (400 W/m^2) and this study showed that the used of the external heat source enhance the SCPP and make it operate and generate power early after sun shine directly even when the solar intensity low. Furthermore, the proposed model in this study make the plant generates power at night.

NOMENCLATURE	
A_c , A_g , $A_{channel}$	Area of transparent cover, ground and flue channels (m^2).
C_p	Specific heat of air (J/kg K).
g	Gravitational constant (m/s^2).
$h_{con,c-air}$	Coefficient of convective heat transfer from canopy to collector air (W/m^2 K).
$h_{con,g-air}$	Coefficient of convective heat transfer from ground to collector air (W/m^2 K).
$h_{r,g-c}$	Radiated heat transfer coefficient from the ground to the canopy (W/m^2 K).
I	Intensity of solar radiation (W/m^2).
$\dot{m}$	Air mass flow rate (kg/s).
uw	Wind speed (m/s).
T_g, T_{air} , T_c	Mean temperature of the ground, air and canopy, respectively (K).
T_{amb}	Ambient temperature (K).
C_d	Coefficient of discharge.
H_{ch}	Height of chimney (m).
L_{g-c}	Height of collector (m).
V_{ch}	Velocity of air at the inlet of the chimney (m/s).

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