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# Solar Chimney coupled to an Earth-to-Air Heat Exchanger for Passive Cooling and Power Generation: a Payback Analysis

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**Abstract.** *The International Renewable Energy Agency, IRENA, reports that 49% of energy consumption is related to heating and cooling processes, which are included the air conditioning needs for the buildings. Earth-Air Heat Exchanger (EAHE) is a passive cooling technique used to heat or cool the air temperature, and Solar Chimney (SC) is a passive ventilation technique. This work aims to analyze the combination of a 2 m depth EAHE and a 1.6 m height SC, as a sustainable way to reduce the energy costs associated with the cooling system. The simulation results were compared to a commercial 18,000 BTU/h inverter type air-conditioner. The economic analysis considered the estimated installation costs of the proposed system and compared them to the operating costs of the commercial air conditioner, showing promising results. In a specific scenario that aims to maintain predefined environmental conditions, a dwelling using an air conditioner and an EAHE simultaneously would be able to reduce energy consumption to about 40%, which could still be partially supplied by the PV panel. The investment payback to implement this system is estimated to be 6 years, making the proposed system an interesting alternative to reduce energy consumption related to heating and cooling in buildings.*

**Keywords:** *sustainability, solar energy, earth-air heat exchangers, passive cooling system, thermal comfort.*

## 1. INTRODUCTION

Air conditioning and ventilation are essential processes to ensure the Air Changes per Hour (ACH), removing pollutants and provide thermal comfort to a dwelling. According to the International Energy Agency, IEA (2018), these activities account for 20% of residential power consumption. The IRENA (2020) report reveals that heating and cooling industrial processes account for 49% of fossil fuel consumption worldwide. An approach to reduce costs and make it more sustainable is to use solar chimneys coupled with earth-air heat exchangers. This is a system designed to induce airflow and cool it by passing inside the ground on hot days.

As discussed in Oliveira and Zotin (2020), Solar Chimneys (SC) are devices designed to provide ACH in a dwelling. They work due to the density difference between the inlet and outlet air, which is a consequence of the heated outlet air produced by solar energy. SC performance is considerably affected by the tilt angle and the received overall solar energy, although others parameters can play a minor role. Wang *et al.* (2020), for instance, have identified that the external wind changes the vortexes and the air distribution in the indoor environment. New contributions to the topic were proposed by Guo *et al.* (2020), using solar chimneys to ventilate a green data center. Recently a new design was proposed by Moosavi *et al.* (2020), suggesting the coupling of a windcatcher with the solar chimney.

Earth-Air Heat Exchanger (EAHE) is a passive technique that uses the soil temperature to reduce air temperature. Once the air temperature is higher than the soil temperature, the buried pipe will exchange thermal energy with the ground through conduction and convection heat transfer processes. Some applications for EAHE systems have been noted by Agrawal *et al.* (2019) and include cooling/heating of residential and agricultural buildings, in solar panels, and in industry as a pre-heater or pre-cooling system. Once the EAHE requires a source of ventilation to force the air to flow through the pipes, one of the possibilities of application is the hybrid system, as used by Misra *et al.* (2012), who found an 18% reduction in energy consumption when the EAHE was coupled to a 18000BTU/h window air conditioner.

To enhance a thermal analysis of an EAHE system requires information about the thermal properties of the soil and temperature measurements. Hermes *et al.* (2020) managed to map these properties considering temperature profiles

according to the depths and the year seasons for soil at the State of Rio Grande do North, Brazil. Alves and Schmid (2015) identified that the best cooling condition occurs when the pipe is buried at about 4m or the undisturbed ground temperature level and that grassed soils performed better than concrete. Although recent advances in scientific publications, it remains challenging to obtain a solid database to perform realistic simulations in other countries.

This work aims to simulate a solar chimney coupled to an EAHE and a Photovoltaic (PV) panel using meteorological and soil data to provide an energy and financial return analysis of the project, focusing on the reduction of electric energy consumption. For this purpose, data were collected from the NDAWN (2015) database at 3 different times throughout the day.

## 2. Simulation procedure

### 2.1 Weather data and system geometry

For a better comparison of the reduction in power consumption, the adopted system was simulated in 3 different cases, presented on Tab. 1. These tables contains soil and climate data collected from NDAWN (2015). The 3 cases in this paper relate to measurements on August 28th, 2017. The difference between the cases is the collected measurement hour on the weather station being 4 p.m. for case 1, 2 p.m. case 2, and 9 a.m. case 3. The collected information can be seen in Tab. 1. The selected weather station is located in the Bowman district, 46.20°N, 103.47°W and the solar radiation was measured on a horizontal plane. To simulate the received solar radiation on an inclined solar chimney, calculations were made as in Oliveira and Zotin (2020) adjusting some equations to North Latitude.

Table 1. Collected data from the weather station located in the Bowman district on August 28th, 2017, at 4 p.m. (case 1), 2 p.m. (case 2), and 9 a.m. (case 3).

	Air temp., K	Rel. Humidity, %	Wind speed, m/s	Solar rad., W/m <sup>2</sup>	Baro. Press., Pa	Soil temp. At 2m depth, K
Case 1	305.74	16.33	2.20	637.20	91090	286.71
Case 2	304.52	18.57	1.77	792.22	91180	286.80
Case 3	293.42	47.15	0.756	345.55	91330	287.14

The SC has an 0.5m<sup>2</sup> inlet and outlet area and the air channel is limited by a glass panel and a photovoltaic panel spacing 0.5m each other. Table 2 shows the characteristics of the PV panel. The PV Panel limits the height and width of the SC to 1.7m and 1m, respectively, and has a maximum capacity of 280 W and a maximum efficiency of 17.83%.  $\alpha_w$  represents the absorptance value and it was taken from Bassiouny and Koura (2008).  $\varepsilon_w$  represents the emissivity value and it was taken from Mathur *et al.* (2006).

Table 2. Photovoltaic panel properties

$\alpha_w$	$\varepsilon_w$	Height, m	Width, m	Power, W	Max Efficiency, %
0.95	0.95	1.7	1	280	17.83

Only in case 1, the PV panel has an insulation on its back face with 0.020 m thick and thermal resistance of 0.0275 W/m<sup>2</sup>K. The glass cover has the same height and width as the PV panel. Its absorptance value 0.06, transmissivity value 0.84 and emissivity 0.90, as suggested by Bassiouny and Koura (2008) and Incropera *et al.* (2000).

Table 3 shows the EAHE properties. The system has 0.1 m internal diameter, and it is 20 m length tube buried at 2 m on the soil.

Table 3. EAHE properties

Internal diameter	External diameter	Length	Depth	Material	Thermal conductivity
0.1m	0.103m	20m	2m	PVC	0.15 W/mK

Figure 1 shows a schematic drawing of the solar chimney and EAHE assembly. The system is composed of a buried duct responsible for the air intake and on the other side, the solar chimney, responsible for the air exhaust.

### 2.2 Energy balance for SC and PV panel

The SC energy balance is determined on three different control volumes: the glass, the PV panel, and the air channel. Through the glass plate, heat gain is obtained from incident solar radiation and part of this energy is lost by the external wind and reflection to the sky. Another fraction of energy is used to heat the inside air by convection and the PV panel by

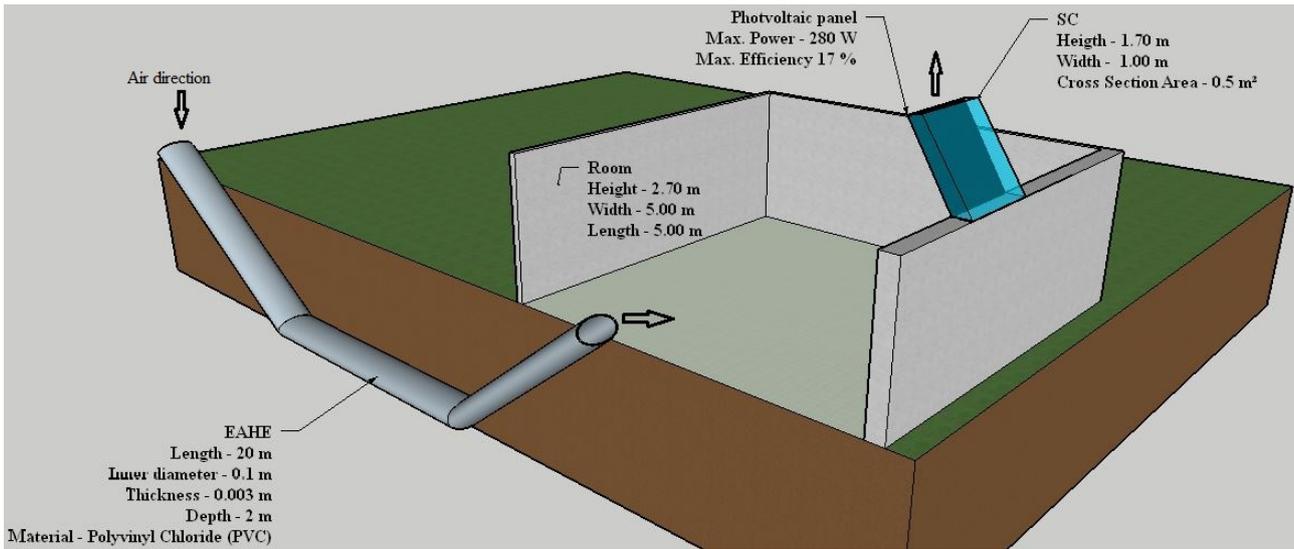


Figure 1. Schematic of a solar chimney and earth-to-air heat exchanger.

radiation. This energy balance is described as

$$S_1 A_g + h_{rwg} A_w (T_w - T_g) = h_g A_g (T_g - T_f) + A_g (h_{wind} + h_{rs}) (T_g - T_a) \quad (1)$$

where  $S_1$  is the solar radiation,  $h_{rwg}$  and  $h_{rs}$  are the radiative transfer coefficient due to the glass to the PV panel and the sky, respectively,  $h_g$  and  $h_{wind}$  are the convection transfer coefficient due to the glass to the inside air and the external wind, respectively. The variable  $T$  represents the temperature values and the variable  $A$ , the area. Its subscripts  $g$  refers to the glass,  $w$  the photovoltaic panel,  $f$  the mean temperature of air inside the channel and  $a$  the external air. The  $h_{wind}$  coefficient is given by Jürges Apud Duffie and Beckman (2013) as

$$h_{wind} = \begin{cases} 5.6 + 3.9U, & U < 5 \text{ m/s} \\ 7.2U^{0.78}, & U \geq 5 \text{ m/s} \end{cases} \quad (2)$$

where  $U$  is the wind speed. Following Kalogirou (2013) works, the radiative heat transfer coefficient between the glass and the PV panel is given by

$$h_{rwg} = \frac{\sigma(T_w + T_g)(T_w^2 + T_g^2)}{\frac{1}{\varepsilon_w} + \frac{1}{\varepsilon_g} - 1} \quad (3)$$

where  $\varepsilon_w$  is 0.95,  $\varepsilon_g$  0.90 and  $\sigma$  is the Stefan Boltzmann constant, and the radiative heat transfer coefficient between the glass and the external ambient is given by

$$h_{rs} = \varepsilon_g \sigma (T_g + T_a)(T_g^2 + T_a^2) \quad (4)$$

The air inside the SC is heated by convection to the glass and the PV panel and the heat flux  $q$  can be determined as a function of the difference between the inlet and outlet temperatures, the specific heat and the mass flow rate, as

$$q'' = h_g A_g (T_g - T_f) + h_w A_w (T_w - T_f) = \frac{\dot{m} c_p (T_s - T_i)}{WL} \quad (5)$$

where  $q''$  represents the heat flux and  $h_w$  is the convection coefficient between the absorber plate and the air inside the duct,  $\dot{m}$  is the mass flow rate,  $c_p$  is the specific heat at constant pressure,  $T_i$  is the air temperature at the inlet nozzle and  $T_s$  the air temperature at the outlet nozzle,  $W$  is the length of the chimney and  $L$  is the height of the chimney.

Since the outlet temperature  $T_s$  is not a given parameter, an alternative is to use a linear correlation between de inlet and outlet temperature. This linearity was suggested by Hirunlabh *et al.* (1999) as

$$T_f = 0.75T_s + 0.25T_i. \quad (6)$$

Consequently, Eq. (5) can be rewritten using Eq. (6), and the output

$$q'' = \frac{\dot{m} c_p (T_f - T_a)}{\gamma WL} = M(T_f - T_a) \quad (7)$$

where  $M = \frac{\dot{m}c_p}{\gamma WL}$ . It should be noted that the temperature  $T_i$  is equivalent to the temperature of the room to be ventilated,  $T_r$ . The photovoltaic panel converts solar energy into electricity and it also heats the air inside the SC. The energy balance of the photovoltaic panel

$$A_w(S_2\alpha_w\tau_g - h_{rwg}(T_w - T_g)) = A_w(S_2\eta_w + 298 \times 10^3\eta_w\beta_wT_w + h_w(T_w - T_f) + U_b(T_w - T_r)) \quad (8)$$

where  $S_2$  is the solar radiation on the PV panel,  $10^3$  is the power during the Standard Tests Conditions (STC), in Watts, and 298 the STC temperature in Kelvins.  $\alpha_w$  is the absorptance coefficient of the PV panel, estimated to be 0.95.  $\tau_g$  is transmittance coefficient of the glass, equals to 0.84, as suggested by Mathur *et al.* (2006).  $U_b$  is the losses associated with the thermal insulation at the back of the panel.  $\eta_w$  is equal to 0.171, which is the maximum PV panel efficiency, and  $\beta_w$  is equal to 0.39, which is the temperature coefficient, both provided by the manufacturer.

To calculate the power provided by the PV panel it is necessary to calculate the PV panel efficiency. It is suggested by Mattei *et al.* (2006) the following equation for PV panel efficiency:

$$\eta = \eta_w(1 - \beta_w(T_a - T_w)) \quad (9)$$

where  $\eta$  is the PV panel efficiency. The generated power  $P_w$  can be expressed as

$$P_w = \eta A_w S_2 \quad (10)$$

### 2.3 Energy balance for earth-air heat exchanger

Consider a pipe made of polyvinyl chloride, also known as PVC, buried 2 meters deep. The system energy balance was calculated considering the heat exchange by conduction between the ground and the pipe wall and convection between the pipe wall and the internal air, as

$$\frac{T_e - T_r}{T_e - T_a} = e^{-\frac{U_e A_e}{\dot{m}c_p}} \quad (11)$$

where  $T_e$  is the soil temperature,  $T_r$  is the outlet air temperature,  $T_a$  is the inlet air temperature,  $\dot{m}$  the air mass flow rate,  $c_p$  is the specific heat of the air at constant pressure,  $A_e$  is the pipe section area. The term  $U_e$  refers to the conduction and convection heat exchanges associated with the tube and can be represented by

$$(U_e A_e)^{-1} = \frac{1}{L} \left[ \frac{1}{h_w \pi D_i} + \frac{\ln \frac{D_o}{D_i}}{2\pi k} \right] \quad (12)$$

where  $L$  is the length of the pipe,  $h_w$  is the convection coefficient between the PVC tube and the air,  $D_o$  is the outer diameter of the tube and  $D_i$  the inner diameter, and  $k$  is the heat exchange coefficient.

Equations 13 to 17 are used to calculate the convection heat transfer coefficient, suggested by Incropera *et al.* (2000) and represent Nusselt's number, Reynold's number, air viscosity and Prandtl's number, respectively.

$$h_w = \frac{Nuk}{L} \quad (13)$$

$$Nu = 0.023 Re^{\frac{4}{5}} Pr^{0.3} \quad (14)$$

$$Re = \frac{4\dot{m}}{\pi D_i \mu} \quad (15)$$

$$\mu = 1.729 \times 10^{-5} \left( \frac{T_a + 273}{273} \right)^{0.7} \quad (16)$$

$$Pr = \frac{c_p}{k} \mu. \quad (17)$$

The air heat reduction can be calculated as

$$Q = \dot{m}c_p(T_r - T_a) \quad (18)$$

### 2.4 Mass flow rate

Although Wang *et al.* (2004) discussed that one of the important system losses is air infiltration, this parameter will not be considered. The air mass flow rate in the heat exchanger is equal to the chimney air mass flow rate. Eq. 19 presents the equation formula for the SC.

$$\dot{m} = c_d \rho A_o \sqrt{\frac{2gL \sin \theta (T_f - T_r)}{\left(1 + \frac{T_r}{T_f} \left(\frac{A_o}{A_i}\right)^2\right) T_r}} \quad (19)$$

where  $c_d$  is the discharge coefficient for the outlet section,  $\rho$  is the air density,  $A_o$  is the outlet area,  $A_i$  is the inlet area, and  $g$  is the gravity. The discharge coefficient  $c_d$  is an empirical value and corresponds to the ratio between theoretical and actual mass flow on discharge and is also described as

$$c_d = \frac{1}{\sqrt{\sum h_l}} \quad (20)$$

where  $\sum h_l$  is the head loss' system. The head loss equation is

$$\sum h_l = \sum l_{in} + \sum l_{out} + f \frac{L}{D_h} \quad (21)$$

where  $\sum l_{in}$  is the inlet losses,  $\sum l_{out}$  is the outlet losses and  $f \frac{L}{D_h}$  the friction loss. According to Ortega (2011) the inlet loss coefficient is assumed equals to 0.5, the outlet loss coefficient equals to 1 and the roughness value is equal to 0.5. By Moody's diagram, the friction factor is a value between 0.025 and 0.05. So, the head loss ranges from  $2,13 < \sum h_l < 2,77$  and the discharge coefficient will be varying between 0.60 and 0.685.

Others authors studied others factors that could be changing the  $c_d$  value. For instance, the wind direction can significantly change discharge coefficient, as shown by de Oliveira Neves and da Silva (2018), which found a variation between 0.56 and 0.8 according to the wind direction. For a vertical solar chimney, Zha *et al.* (2017) led an experiment where the results suggested 0.51 for  $c_d$  value. Although the database used in this work provides the wind speed and direction, the wind does not behave as a constant parameter throughout a city and the solar chimney can be easily affected by topography, surrounding buildings, vegetation and its own geometry and materials. So, in this study, it was decided to adopt a fixed value for the discharge coefficient as suggested by Flourentzou *et al.* (1998), with  $c_d = 0.6$ . All this steps described above are summarized on Fig. 2

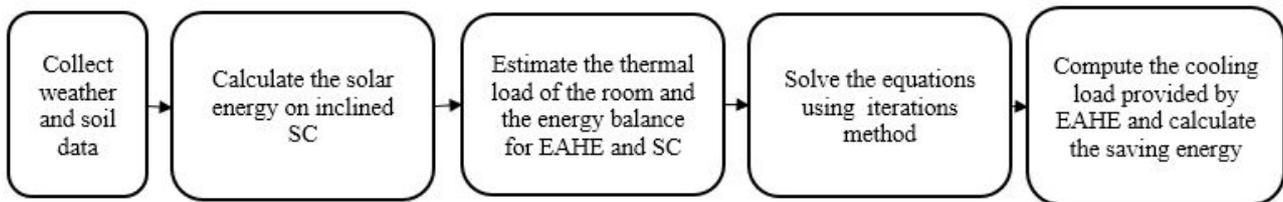


Figure 2. Flowchart of the calculations

## 2.5 Payback investment analysis

To evaluate the feasibility analysis, it was considered a 25m<sup>2</sup> dwelling with four windows sizing 1.2m x 1m windows that needs to be maintained at an internal temperature of 295K. For cases 1, 2, and 3, the dwelling heat load was estimated as 4515.2 W, 5418.25 W, and 3386.4 W, respectively. These values were estimated based on the temperature and solar radiation and may differ from reality due to the infiltration and sensible heat loss. The energy-saving was calculated based on the energy demand for the air conditioner when EAHE is installed and not. The installation costs were estimated to calculate the payback time to recover the investment. The selected Heating, Ventilation, and Air Conditioning (HVAC) is an inverter type 18000 BTU/h, that consumes 1720 W of electrical energy. The payback time is calculated by the division between the total cost due to the system development and the energy saving.

## 3. Results

### 3.1 Overall Analysis

The study was conducted with 3 different cases of weather and soil measurements queried from the NDAWN (2015) database for August 28th, 2017, at 9 am, 14 pm, 16 pm. From these data, it is possible to understand the temperature and solar radiation distribution throughout the day. At first, cases 1, 2 and 3 weather data were simulated with a range of soil temperatures, varying them from 273 to 303 K. The air mass flow values were compared to the solar chimney heating process, measured with the difference between the room temperature and the external air as shown on Fig. 3. The x axis shows the difference between the Mean Air Temperature in the Channel ( $\Delta T$ ) and the y axis shows the air mass flow rate.

Case 3 has a minor level of incident solar radiation. So, the higher mass flow is achieved when soil temperatures are equal to 273 K. For all the cases, as the air mass flow is higher as the soil temperature is lower, indeed. Case 1 has an intermediary level of incident solar radiation and 0.02 m insulation on the PV panel. So, the air mass flow for this case is always higher than in Case 3. Except for solar radiation value, Case 1 and 2 has similar weather data, but case 1 has insulation and case 2 doesn't have it. The insulation improves the SC heating process, achieving higher  $\Delta T$  and affects

the air mass flow value. If  $\Delta T$  ranges between 2.7 and 3.1, Case 2 achieves higher mass flow rates because it had a higher solar radiation incidence. But the air mass flow rates in Case 1 are higher than Case 2 if  $\Delta T$  is higher than 3.1.

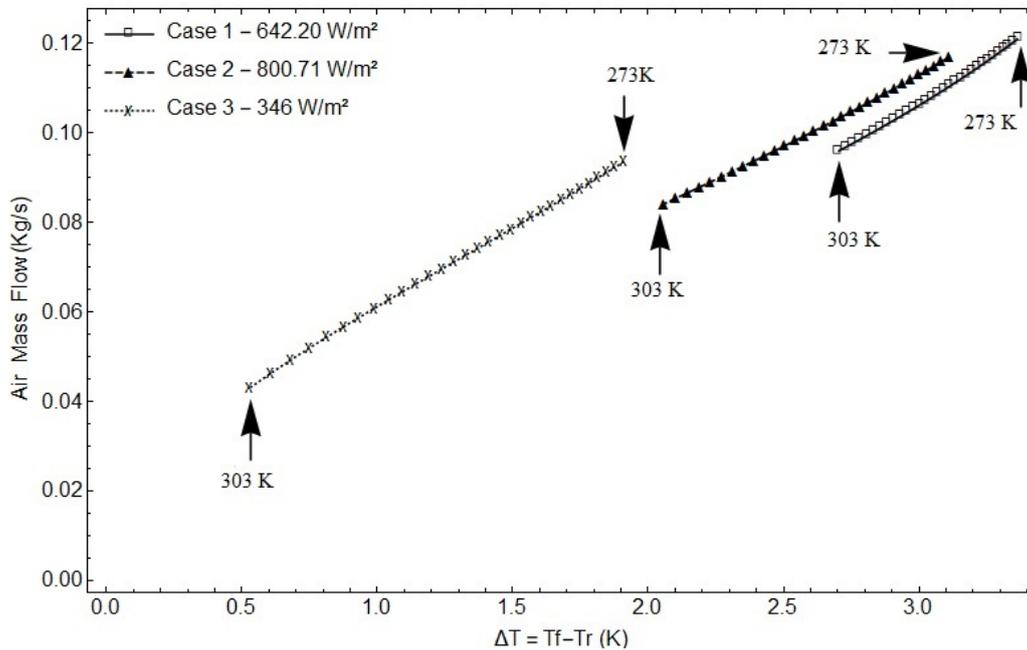


Figure 3. Difference between the chimney air temperature and the inlet air temperature vs air mass flow rate.

Thermal insulation has a high influence on the temperature value of the PV array, as shown in Fig. 4. As the simulated soil temperature varied between 273 and 303 K, the PV panel temperature increased proportionally. However, when comparing the value of  $T_w$ , it decreased as the flow rate was increased, for both cases. The effect of thermal insulation directly impacted the PV panel temperature, where Case 1 has highest values for  $T_w$ . The flow rate in Case 2 showed a smaller variation concerning the  $T_w$  temperature, making the PV panel temperature in case 2 lower.

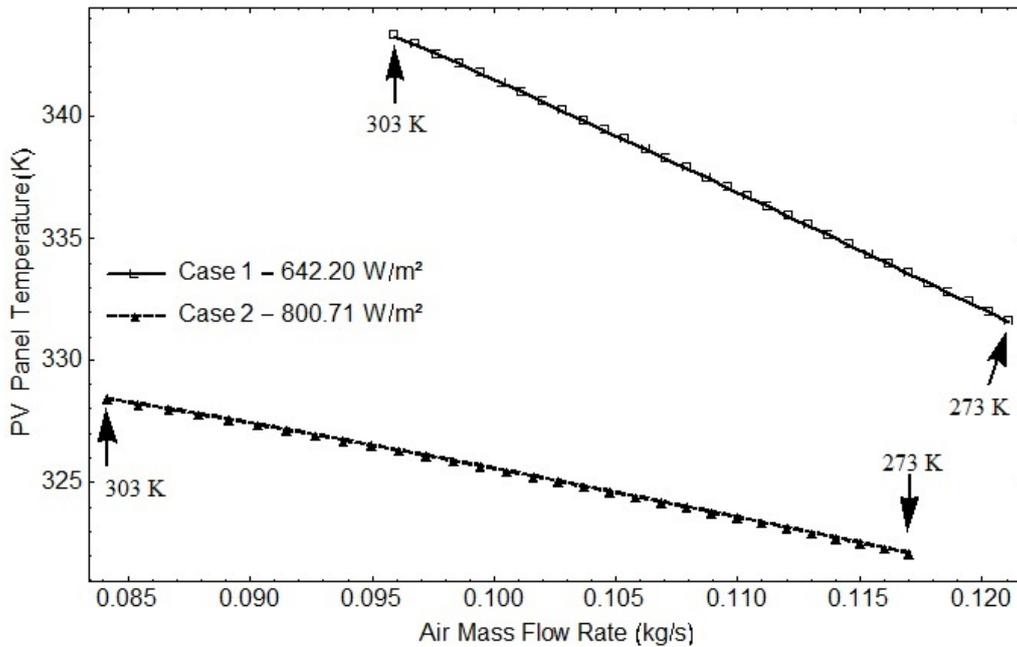


Figure 4. PV panel temperatures x air mass flow

The dwelling temperature is shown in Fig. 5 for the variation of soil temperatures. Cases 1 and 2 had similar dwelling temperatures, but slightly higher in case 1. Case 3 has the lowest dwelling temperature, but this does not indicate a higher efficiency of the heat exchanger. Occurs that in this case, the external air temperature is lower than the other cases. It is also observed that while the external air temperature is 293.42 K in case 3, and the simulated soil temperature has higher temperature values, the system starts to heat the dwelling, instead of cooling it.

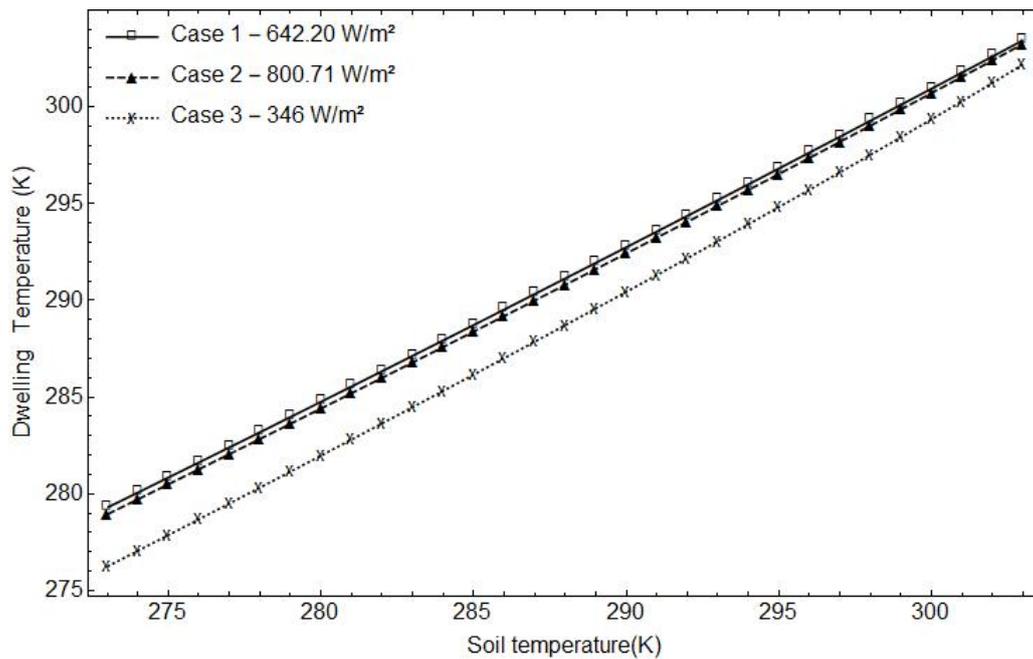


Figure 5. Dwelling temperature x Soil temperature

After the analysis for a range of soil temperatures was completed, we proceeded with the specific evaluation of the three collected climate measurements. The results for the 3 cases are shown in Tab 4.

Cases 1 and 2 show similar results because the weather parameters are relatively close between each other. While case 1 has 305.74 K for external temperature, case 2 had 304.72 K. At the EAHE outlet, the temperature decreases to 290.10 and 289.83, respectively. The received solar energy on case 2 is higher than case 1, but the insulation led to a higher PV panel temperature on case 1. So, using Eq. 10 the generated power by the PV panel is about to 157 W for case 1 and 207 W for case 2.

Comparing cases 1 and 3, it is possible to see that as observed for a simple solar chimney, as in Oliveira and Zotin (2020), the inlet air temperature in the SC and the solar energy received has a relevant influence on the airflow.

Table 4. Simulation results

	EAHE			SC			
	Inlet temp., K	Outlet temp., K	Air mass flow rate, kg/s	Mean air temp., K	PV panel Temp., K	Solar radiation over SC, W/M <sup>2</sup>	PV panel efficiency, %
Case 1	305.74	290.1	0.11	293.19	336.92	642.91	14.41
Case 2	304.72	289.83	0.10	292.50	325.02	800.71	15.20
Case 3	293.42	287.98	0.075	289.38	305.03	346.2	16.53

Due to the lowest solar radiation, the relatively stable soil temperature at 2 m depth, and the lowest outside air temperature, case 3 presents lowest air mass flow values. However, the air flowing through the chimney is cooler when compared to the other cases studied, providing the lowest temperature in the PV panel, which increases its efficiency. However, even with a higher efficiency, power generation is still dependent on the received solar radiation. For case 3, it is estimated to generate 97 W.

### 3.2 Payback analysis

To calculate the investment payback for the building, the thermal load for case 1 was estimated at 4515.2 W, case 2 at 5418.25 W, and case 3 at 3386.4 W. Using data from Tab. 4, Tab. 5 presents the calculation performed to estimate the reduction in energy consumption. For case 1, where the outside temperature was 305.74 K, an energy consumption reduction of 38.28%, approximately, was estimated to keep the dwelling at 295 K.

The similar flow and temperature values for cases 1 and 2 were discussed earlier. Thus, it can be seen in Tab. 5 the energy-saving is smaller than the previous case due to the higher thermal load. In case 2, the energy-saving was estimated to be 26.45% for the given conditions.

Finally, case 3 presents a situation that would be plausible in real cases, where there is no need to keep the air

Table 5. Step-by-step energy saving calculation

Calculations	Case 1	Case 2
Dwelling thermal load, W	4515.20	5418
Power cooling from EAHE, W	1728.43	1539.74
Power demand for HVAC, W	2786.77	3878.26
Power consumption for HVAC, W	909.19	1265.28
Energy saving, W	563.9	454.71
Saving's percentual, %	38.28	26.45

conditioner running. In this case, the goal was to keep the room at 295 K, but the external temperature was lower than that. At this point, it was not possible to benefit from the EAHE heating process, as a minimum soil temperature of 296K would be required. The estimated costs for SC and EAHE installation are shown in Tab. 6.

Table 6. Estimated costs to build the system

Item	Specification	Quantity	Rate cost	Total cost
Solar Panel	280 W and 17% efficiency	1	\$111.62 per panel	\$111.62
Glass plate	Tempered Glass 8mm	3.4	\$51.17 per $m^2$	\$173.98
Tube	Aluminium tube 4" X 1/8" (10,17cm X 3,17mm)	24	\$26.63 per $m^2$	\$636.75
Tube accessories	Aluminum	2	\$18.95 per unity	\$37.90
Soil perforation	Machine rental	20m straight x 2m depth	\$94.75 per day	\$94.75
			Total	\$1,055.00

Tab. 7 shows the payback time for the investment. The installation cost is added by 2% for fee maintenance and operation. The unit kWh consumption is estimated to be \$0.16. Based on the mean energy reduction for cases 1 and 2, the system would have a payback of 13,205 hours of operation. If this condition is satisfied for 6h/day, the payback of the investment, in years, would occur in 6 years, which is a very interesting period if compared to the useful life of a building.

Table 7. Payback time calculation

Parameters	Value
Mean energy saving, W	509.30
Energy cost, \$	0.16
Installation costs,\$	1,055.00
Maintenance costs, \$	21.10
Total cost,\$	1,076.10
Payback time, Hours	13,205.49
Payback time, Years	6.03

#### 4. Conclusions

In order to investigate a sustainable solution for saving energy in HVAC systems, it was proposed to use an EAHE to cool the air and a solar chimney as an air blower to balance the pressure drop. To improve energy efficiency, the solar chimney was designed with a photovoltaic panel to provide energy to other demands, but it is noted that the power generation is dependent on the received solar energy radiation and the system can also be used to increase efficiency by reducing the operating temperature. The proposed system presented a good reduction in air temperature and proved to be a feasible option for reducing energy consumption, reaching energy-saving between 26% and 38%. Under these conditions, an investment would pay off in 6.03 years of operation. Some aspects of this work that were not discussed remain as suggestions for future work, such as, the best geometry for the system, optimizations on power generation varying the SC inclination and the impact of a mechanical air blower to increase the airflow.

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