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RESEARCH ARTICLE

STUDY OF THE PARAMETERS INFLUENCING THE THERMAL PERFORMANCES OF AN EARTH-TO-AIR HEAT EXCHANGER USED IN COOLING MODE

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ABSTRACT

The geothermic low temperature can be used to produce the fresh air with the help of an adequate heat exchanger. This last one is often constructed from a pipe in PVC buried in soil and in which we make circulate the ambient hot air. Certainly, the air outlet temperature depends on the climatic conditions of the site and the geometric and dynamic parameters of the heat exchanger. In this work, we present a study led on an earth-to-air heat exchanger (EAHE) operated under the climatic conditions of Algeria Sahara. A dynamic modeling of the system has been presented in detail and the effect of pipe extremities has also been taken into account in this study. Results indicate that the air outlet temperature decreases with increasing of the length of buried pipe whereas it increases with increasing of the pipe section and the air velocity. However the daily mean efficiency increases when the pipe length increases but it decreases if the pipe section or the air speed increases. In addition, the effect of the pipe thickness is also studied in this work and it was found that the performances of the EAHE are weakly affected by this last parameter.

INTRODUCTION

The Earth-to-Air Heat Exchanger (EAHE) is a subterranean cooling system that consists in a pipe or network of pipes buried at reasonable depth below the ground surface. Ventilation air supply is passed through the pipes and the difference in temperature between the pipe surface and the air drives the heating/cooling of the ventilation air. The magnitude of the heat exchange between air and pipe is dependent on factors such as, soil temperature, air inlet temperature, pipe dimensions, air flow rate, pipe burial depth and soil and pipe thermal properties. The main advantages of the system are its simplicity, high cooling and pre-heating potential, low operational and maintenance costs, saving of fossil fuels and related emissions. Pre-heated fresh air supports a heat recovery system and reduces the space heating demand in winter. In summer, in combination with a good thermal design of the building, the EAHE can eliminate the need for active mechanical and air-conditioning units in buildings, which will result in a major reduction in electricity consumption of a building if the EAHE is designed well. A literature review reveals that various studies had been conducted on the EAHE under different climatic conditions in order to improve its performances. We can cite some of them:

Gauthier *et al*. (1997) presented a numerical study conducted for the thermal behavior of soil heat exchanger-storage systems (SHESSs) aimed to reducing the energy consumption of greenhouses. These systems consist in buried pipes circulating air for storing and removing heat from the soil. Firstly, a transient fully three-dimensional heat transfer model resting on the coupled conservation equations of energy for the soil and the circulating air is presented. The model is validated with experimental data taken from a SHESS installed in a commercial type greenhouse. Secondly, the model is used to investigate the effect of various design and operating parameters on the performance of SHESSs. Results indicate that the total amount of energy stored or recovered daily per volume decreases exponentially with increasing of the pipe center-to-center distance and the pipe length. It increases with the air velocity and this effect is enhanced as the pipe center-to-center distance diminishes. Pfafferot (2003) has studied the performance of three earth-to-air heat exchangers for mid European office buildings in service in order to determinate their efficiency. A general method to compare EAHXs in operation was introduced. First, the temperature behavior is described by plots over time and characteristic lines, and compared by standardized duration curves. Second, the energy gain is illustrated by standardized graphs. Third, a parametric model is used to provide general efficiency criteria.

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Kumar *et al*. (2003), in order to predict energy conservation potential of earth-air heat exchanger system and passive thermal performance of building, have presented a numerical model which improves upon previous studies by incorporating effects of ground temperature gradient, surface conditions, moisture content and various design aspects of earth-air-tunnel (EAT). This model is based on simultaneously coupled heat and mass transfer in the EAT and is developed within the scope of numerical techniques of Finite Difference. The model is validated against experimental data and it is then used to predict the outlet air temperature for various parameters such as humidity variations of circulating air, air flow rate and inlet air temperature. Al-Ajmi *et al*. (2006) have proposed an analytical model of an earth-to-air heat exchanger (EAHE) in order to predict the outlet air temperature and cooling potential of these devices in a hot, arid climate. The model is validated against other published models and shows good agreement. A sub-soil temperature model adapted for the specific conditions in Kuwait is presented and its output compared with measurements in two locations. A building model representative of a typical Kuwaiti dwelling has been implemented and all the models have been encoded within the TRNSYS environment. Results indicate that the EAHE could provide a reduction of 1700 W in the peak cooling load and 30% in cooling energy demand over the peak summer season. Cucumo *et al*. (2008) proposed a one-dimensional transient analytical model to estimate the performance of earth-to-air heat exchangers, installed at different depths, used for building cooling/heating. In this model, two independent space coordinates are considered, one in the longitudinal direction of the buried pipe and the other through the soil, in the vertical direction. An analytical treatment is proposed to predict the temperature fields of fluid in the pipe and of the soil in the proximity of the buried pipe, taking into account the thermal perturbation of the upper free surface and the possible phase change (condensation) in the buried pipes.

Thiers *et al*. (2008) presented a study conducted on a two-dwelling passive building in Formerie (North-West of France), complying the ‘‘Passivhaus” standard. This building has been modeled using the dynamic simulation software COMFIE. In order to account for the implemented ventilation system, including a heat recovery unit and an earth-to-air heat exchanger, a new module has been developed and integrated to COMFIE. The results of simulation are presented for the passive house and a reference house complying with the French thermal regulation for buildings. The heating load and thermal comfort level of both houses are compared, showing for the passive design a tenfold reduction of the heating load and a clear reduction of summer discomfort. Ho Lee *et al*. (2008) developed a new module which is integrated and implemented in the EnergyPlus program for the simulation of earth tubes. Using the new module, a parametric analysis was carried out to investigate the effect of pipe radius, pipe length, air flow rate and pipe depth on the overall performance of the earth tube under various conditions during cooling season. Pipe length and pipe depth turned out to affect the overall cooling rate of the earth tube, while pipe radius and air flow rate mainly affect earth tube inlet temperature. Tittelein *et al*. (2009) developed a new numerical model of earth-to-air heat exchangers. The system is cut up into ‘‘n” sections perpendicular to the exchanger pipe. In each section, the problem of conduction is solved using response factors method in order to reduce computational time. Each response factor is calculated using a finite elements program that solves 2D conduction problems. Heat flow entering the pipe is then expressed as a function of the temperature of air circulating in the pipe and the external solicitations. A heat balance is then applied for each layer to find the resulting outlet air temperature. This model is then compared to another analytical model based on the dynamic finite volume approach.

Chel *et al*. (2009) have developed a thermal model of a vault roof building integrated with an earth-to-air heat exchanger (EAHE). The building under consideration is made of brick vault and adobe (or mud) structures. The energy balance equations were solved simultaneously using fourth order Runge-Kutta numerical technique. The thermal model was validated using experimental observed data. Experimental results showed that the room air temperature during winter was found 5–15.8 °C higher as compared to ambient air temperature while lower during summer months. The results show that annual energy saving potential of the building before and after integration of EAHE were 4946 kWh/ year and 103 21 kWh/ year respectively. Bansal *et al*. (2010) have presented a transient and implicit model based on computational fluid dynamics for predicting the thermal performance and heating capacity of earth-air-pipe heat exchanger systems. The model is developed inside the FLUENT simulation program. The model developed is validated against experimental investigations on an experimental set-up in Ajmer (Western India). Good agreement between simulated results and experimental data is obtained. Effects of the operating parameters (pipe material, air velocity) on the thermal performance of earth-air-pipe heat exchanger systems (EPAHE) are studied. Investigations on steel and PVC pipes have shown that performance of the EPAHE system is not significantly affected by the material of the buried pipe. Velocity of air through the pipe is found to greatly affect the performance of EPAHE system.

Nayak *et al*. (2009) studied during full year the effectiveness of photovoltaic/thermal (PV/T) and earth air heat exchanger (EAHE) integrated with a greenhouse, located at New Delhi, India. A simplified analytical model has being used to study the solar energy application through photovoltaic system and earth air heat exchanger (EAHE) for heating and cooling of a greenhouse. It has being established a comparison between the greenhouse air temperatures when it is operated with photovoltaic/thermal (PV/T) during daytime coupled with earth air heat exchanger (EAHE) at night, and air temperatures when it is operated exclusively with photovoltaic/thermal system (PV/T) and earth-to-air heat exchanger (EAHE), for 24 hrs. The results indicate that air temperature inside the greenhouse can be increased by around 7-8 °C during winter season, when the system is operated with photovoltaic (PV/T), coupled with earth air heat exchanger (EAHE) at night. From the results, it is seen that the hourly use full thermal energy generated, during daytime and night, when the system is operated with photovoltaic (PV/T) coupled with earth-to-air heat exchanger (EAHE), is 9.16 kWh and 6.8 kWh respectively.

Zhang *et al.* (2010) have developed a new method for studying the earth-to-air heat exchangers (ETAHE), which consist in an Artificial Neural Network based on Heat Convection (ANN-HC) algorithm for predicting the local average Nusselt Numbers along the pipe. Furthermore, the ANN-HC algorithm is integrated with a transient three-dimensional heat transfer model based on finite element analysis of heat conduction in the ground domain surrounding the ETAHE to establish a new thermal modeling method for these systems. It is shown that the method can very well simulate the interactions between an ETAHE and its environment. Maerefat *et al.* (2010) presented a study on the use of solar chimney (SC) together with earth to air heat exchanger (EAHE). Theoretical analyses have been conducted in order to investigate the cooling and ventilation in a solar house through combined solar chimney and underground air channel. The results show that the solar chimney can be perfectly used to power the underground cooling system during the daytime, without any need to electricity. Moreover, this system with a proper design may also provide a thermally comfortable indoor environment for a large number of hours in the scorching summer days. Vaz *et al.* (2011) presented an experimental and numerical study of earth-to-air heat exchangers, which are used to reduce consumption of conventional energy for heating and cooling of built environments through the use of thermal energy contained in the soil. The experiment was conducted in southern Brazil in the city of Viamão, and its results were used to validate the computational modeling of heat exchangers. In this study, the variation of air temperature inside the ducts for an annual cycle was investigated. The numerical solution of the conservation equations of the problem is resolved by the help of the FLUENT environment. The results showed the validity and effectiveness of the employed computational model.

Yildiz *et al.* (2011) studied an experimental system in order to investigate the energetic performance of a solar photovoltaic system (PV) assisted earth-to-air heat exchanger that is used for greenhouse cooling at the Ege University, Izmir, Turkey. System was operated about 11 h/ day. As a result, total electricity energy consumption of the system was measured to be 8.10 kWh and 34.55% of this energy demand was provided from photovoltaic cells. Furthermore, 65.45% of the electricity energy demand was provided from grid connection. Su *et al.* (2012) proposed an air-earth-tunnel system used for building energy saving. A computing model of the air-earth (rock) heat exchanger is necessary to predict the thermal performance. A numerical simulating model has been developed for the deeply buried air-earth-tunnel system, in which a one-dimensional implicit transient convection-diffusion sub-model describes the air temperature and humidity, and a one-dimensional transient explicit heat conduction sub-model computes the rock temperature. The model was validated against experimental data. In this paper, we present a numerical modeling of an earth-to-air heat exchanger under the climatic conditions of Algerian Sahara used for cooling of buildings. This study has been made in the summer period and exactly in July where the demand on energy for cooling is at the peak. The effect of the pipe extremities is also taken into account in this analysis. Next, this model will permit us to investigate the influence of the geometric and dynamic parameters of the heat exchanger as well as the air inlet temperature on the air outlet temperature, on the daily mean efficiency of the system and also on the coefficient of performance (COP).

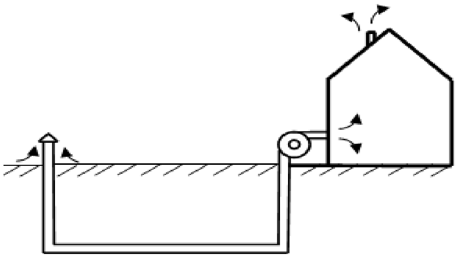

Figure 1. Representative diagram of an earth-to-air heat exchanger

MATERIAL AND METHODS

A. Description & principle of working

The heat exchanger is formed from a pipe in PVC (Polyvinyl chloride) buried at a depth of 2 m in the ground. The buried pipe has a length L, an internal diameter Di and a thickness of 2 mm. The principle of working of this exchanger can be explained as follows. First, the ambient hot air is put in circulation in the buried pipe with a help of an adequate fan. Next, during its passage, air gives up a quantity of its heat to the pipe and it gets cold as it progresses and finally the fresh air is injected in the building (Figure 1).

B. Mathematical modelling (Enumeration of hypothesis)

The modelling of the earth-to-air heat exchanger is established in such a way that the following hypotheses are respected:

- We are interested solely to the evolution of the temperature in the flow direction.
- The air physical and thermal properties are constant and independent of the temperature.
- Soil around the tube is homogeneous and its physical and thermal properties are supposed constants and not influenced by the presence of the pipe.

We consider a pipe in PVC with a length L and diameters inside and outside Di and Do respectively. The ambient hot air circulates in the pipe with a mass flow rate ṁ. We designate by Ds and Tsoil the outside diameter of the disturbed soil and the temperature at the outside surface of the disturbed soil respectively. According to the flow direction, we cut up the system in several identical sections of length Δx. In every section, temperatures of air, the pipe and the disturbed soil are uniform. For a section (j), the energy balance is written by the following manner.

For the air in the buried pipe:

$$\dot{m}Cp_a \frac{dT_a}{dx} = -(\pi D_i)U_{a,t}(T_a - T_t(j)) \qquad (1)$$

Where Ta, Cpa and ṁ are respectively the air temperature, the specific heat capacity of air and the mass flow rate of air. The parameter Ua,t indicate the overall heat transfer coefficient between the air and the pipe.

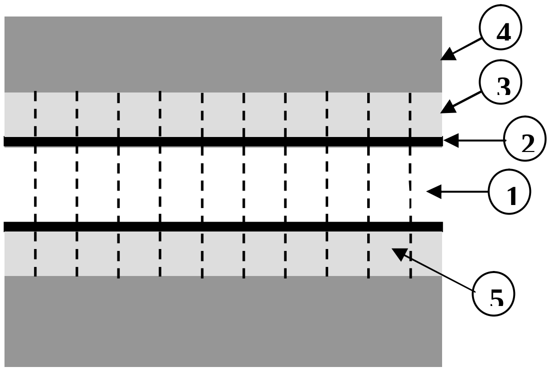

Figure 2. Schematic representation of the studied system: 1) air, 2) pipe, 3) perturbed soil, 4) unperturbed soil and 5) section

For the pipe:

$$M_t Cp_t \frac{dT_t(j)}{dt} = (\pi D_i \Delta x) U_{a,t}(T_a - T_t(j)) - (\pi D_e \Delta x) U_{t,s}(T_t(j) - T_s(j)) \tag{2}$$

The parameters Tt, Cpt, Ut,s and De indicate respectively the pipe temperature, the thermal capacity of the pipe, the overall heat transfer coefficient between the pipe and the disturbed soil and the outside diameter.

For the disturbed soil:

$$M_s Cp_s \frac{dT_s(j)}{dt} = (\pi D_e \Delta x) U_{t,s}(T_t(j) - T_s(j)) - (\pi D_s \Delta x) U_{sol}(T_s(j) - T_{sol}) \tag{3}$$

With:

$$\frac{1}{U_{a,t}} = \frac{1}{hc} + \frac{D_i}{2 \cdot \lambda_t} \ln \frac{(D_i + D_e)/2}{D_i} \tag{4}$$

$$\frac{1}{U_{t,s}} = \frac{D_e}{2 \cdot \lambda_t} \ln \frac{D_e}{(D_i + D_e)/2} + \frac{D_e}{2 \cdot \lambda_s} \ln \frac{(D_e + D_s)/2}{D_e} \tag{5}$$

$$\frac{1}{U_{sol}} = \frac{D_s}{2 \cdot \lambda_s} \ln \frac{D_s}{(D_e + D_s)/2} \tag{6}$$

Where, Tsol, Usol and Ds are respectively the temperature of the undisturbed soil and the overall heat transfer coefficient by conduction between the disturbed and the undisturbed soil and the outer diameter of the disturbed soil. Besides, λt, λs and hc designate respectively the pipe thermal conductivity, the soil thermal conductivity and the coefficient of heat transfer by convection between the air and the pipe internal walls. In the cooling mode, the convective heat transfer coefficient between air and the internal walls of the buried pipe is given by the following relation [16]:

$$hc = 0.023\, Re^{0.8}\, Pr^{0.33}\, \lambda_a / D_h \tag{7}$$

The pressure drop of the system is calculated with the help of the following equation:

$$\Delta p = \left[f \frac{L}{D_h} + 2 \times 1.3 \right] \cdot \rho \frac{v^2}{2} \tag{8}$$

Where:

$$f = (1.82 \log(\mathrm{Re}) - 1.64)^{-2} \tag{9}$$

The mean efficiency of the earth-to-air heat exchanger for a period τ is calculated by the following equation:

$$\eta_{mean} = \frac{\int_0^\tau (T_a^{(e)} - T_a^{(s)}) \cdot dt}{\int_0^\tau (T_a^{(e)} - T_{sol}) \cdot dt} \tag{10}$$

In the same way, the mean coefficient of performance of the system for a period τ is given as follows:

$$COP = \frac{\dot m Cp_a \int_0^\tau (T_a^{(e)} - T_a^{(s)}) \cdot dt}{\Delta p V' \tau / \eta_{Fan}} \tag{11}$$

The η_{Fan} is a parameter that indicates the output of electromechanical conversion of the fan. The temperature of air in exit of a section (j) is gotten by the resolution of the differential equation (1) thus:

$$T_a^{(s)}(j) = T_t(j) + (T_a^{(e)}(j) - T_t(j)) \cdot e^{-\frac{\pi D_i \Delta x U_{a,t}}{\dot m Cp_a}} \tag{12}$$

For every step of time, the temperature of air in exit of the section (j) is considered as the temperature of entrance for the following section (j+1) and so on; so much that (j) is lower to the number of sections (N). The discretization of the differential equations has been done by the implicit finite difference method, afterwards the linear system resulting is solved by Jordan's algorithm. In order to take into account the effect of the pipe extremities, we did corrections on the temperature of air before it enters in the pipe and also after it comes out from the pipe. For that, we have used the following equation that should be solved by the algorithm of Runge-Kutta for every step of time:

$$\dot m Cp_a \frac{dT_a}{dx} = -U_{tot}(T_a - T_{soil}(x)) \tag{13}$$

With:

$$\frac{1}{U_{tot}} = \frac{1}{hc \cdot (\pi \cdot D_i)} + \frac{1}{(2 \cdot \lambda_t \cdot \pi)} \ln \frac{D_e}{D_i} + \frac{1}{(2 \cdot \lambda_s \cdot \pi)} \ln \frac{D_s}{D_e} \quad (14)$$

In the steady-state model, the air outlet temperature is given by the following relationship (Costa 2006):

$$T_a^{(s)} = T_{soil} + (T_{amb} - T_{soil}) \cdot e^{-\frac{U_t L}{\dot{m} Cp_a}} \quad (15)$$

Where:

$$\frac{1}{U_t} = \frac{1}{hc \cdot (\pi \cdot D_i)} + \frac{1}{(2 \cdot \lambda_t \cdot \pi)} \ln \frac{D_e}{D_i} \quad (16)$$

The equation (15) shows explicitly that firstly, when the length of the tube tends to infinity, the air outlet temperature tends to the soil temperature. Secondly, when the air mass flow rate is infinite, the air temperature in the heat exchanger does not change and remains constant.

Table 1. Physical and thermal properties of materials used in this study

Material	Density (kg/m3)	Thermal Capacity (J/kg°C)	Thermal Conductivity (w/m°C)
Air (300k)	1.1774	1005.7	0.02624
Soil	2050	1840	0.52
PVC	1380	900	0.16

RESULTS AND DISCUSSION

This earth-to-air heat exchanger (EAHE) is operated under the meteorological conditions of the town of Adrar. This town is located in the southwest of Algeria. These geographical coordinates are: latitude 27.88° North, longitude 0.28° West and altitude 264m over the sea level. A severe continental climate reigns in this region of the Algerian Sahara. This climate is dry and very hot in summer and very cold in winter (Figure 3). Besides, the summery period is too long and spreads on nearly six months, from the month of May until the month of October. In order to show the feasibility of such system, we chose like period of study the month of July where the demand in air-conditioning is very important because the ambient temperature exceeds 44°C in the shade (Figure 4).

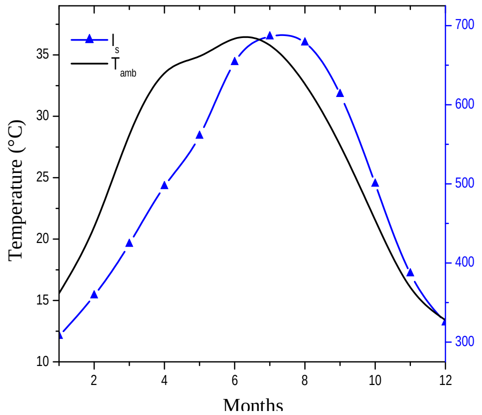

Figure 3. Variation of the average monthly temperature and average monthly global solar radiation for a typical year in Adrar

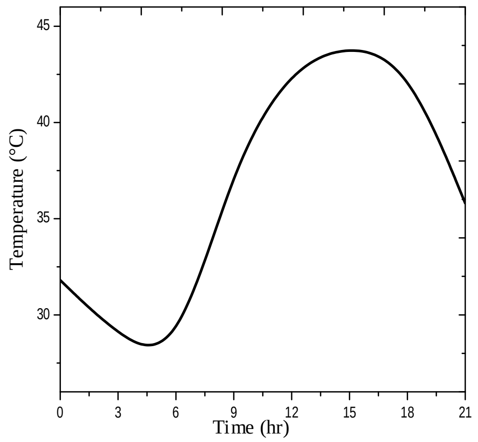

Figure 4. Variation of the mean ambient temperature for a typical day in July

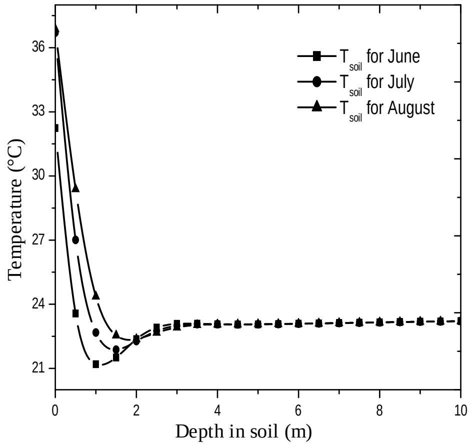

Figure 5. Variation of the soil temperature according to the depth for the summery period

In order to determine the adequate depth of burying of the EAHE, we represented in the Figure (5) the variation of the soil temperature as function of the depth for the summer season. We can see easily that from a depth of 2m, the soil temperature becomes steady during all the season and the soil temperature at this depth reaches 22.27°C. On the other hand, it should be noted that this depth depends on soil kind (sand, clay, rock) (Benhammou *et al*., 2012). In this work, the pipe of the EAHE is in PVC and its thickness is 2mm. It is buried at 2m of depth in soil. The soil temperature corresponding to this depth is 22.27°C. The time (t) is expressed in hours and it spreads on a period of 24 hours. The temperature of air in entrance of the pipe is equal to the ambient air temperature (Tamb). The physical and thermal properties of air, the tube and soil are supposed constant and their values are shown in the table (I) and the parameters of the EAHE used in this simulation are illustrated in the Table (II). If any of these parameters varies during simulation, the others remain constant and take their values of reference.

Table 2. Parameters of the Eahe

Parameter	Value
Length of pipe (L)	50 m
Interior diameter of pipe (D_i)	30 cm
Thickness of pipe	2 mm
Velocity of air (V)	2 m/s
Temperature of the unperturbed soil	22.27 °C

A comparison between the steady state and dynamic models is shown in Figure (6).

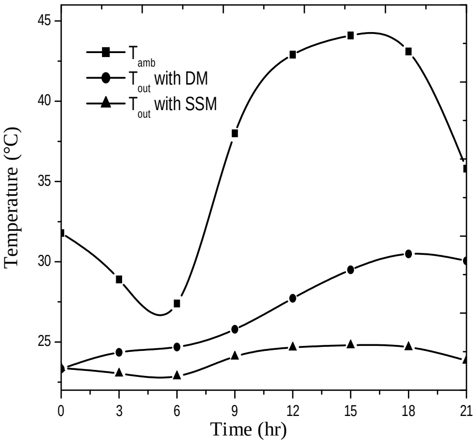

Figure 6. Comparison of the air outlet tempertaure calculated by both models. The black graph is calculated by the dynamic model while the red one is evaluated by the steady state model

We can see that firstly, the air temperature estimated by the steady state model is much lower than that calculated by the dynamic model. At noon (midday), the ambient temperature is near 38 ° C but the corresponding outlet temperature of air is 28 ° C for the dynamic model and 24 ° C for the steady state model. Secondly, the temperature graph obtained by the steady state model varies in the same manner as the ambient temperature. Indeed, there is a phase agreement except that the amplitude decreased. This is due to the fact that the steady state model does not take into account the thermal inertia of the system (pipes, soil). This leads us to conclude that the model is inadequate and gives erroneous results. Thirdly, it is also noted that the air outlet temperature increases with increasing of the air inlet temperature and the air temperature difference between the inlet and outlet of the pipe is particularly important when the air inlet temperature is higher. For example, there has been noted a decrease of 10 ° C for an inlet temperature of 38 ° C but for an inlet temperature of 28 ° C, we found only a decrease of 4 ° C. This is interpreted by the fact that when the air enters with a high temperature in the buried pipe, there occurs a large temperature gradient between the flowing air and the walls of the pipe thus increasing the heat exchange. Hence, the air will lose more heat. For the effect of the pipe length, we can observe in the Figures (7 and 8) that the temperature of air in exit of the heat exchanger decreases with increasing of the pipe length. It is observed that the air maximum outlet temperature decreased from 36.92 ° C to 30.48 ° C when the pipe length was increased from 20 m to 50 m, which corresponds to a decrease of 6.44 ° C (Figure 8). This can be interpreted by the fact that when the pipe length increases, the exchange surface increases also and since the quantity of heat lost by air is proportional to the exchange surface, therefore the air temperature will decrease further. Besides, for a given length, the fall of air temperature is more important when the air inlet temperature is higher. In the figure (9), it is shown the variation of the daily mean efficiency as well as the coefficient of performance according to the pipe length. It is easy to see that the daily mean efficiency increases with the increase of the length while the coefficient of performance decreases. This last result is due to linear losses of pressure which increase with the increase of the pipe length. Concerning the effect of the diameter of the tube, we note in the Figure (8) that the increase of the diameter drags an increase of the air temperature in exit of the heat exchanger. For example, the air outlet temperature has increased from 22.8 ° C to 26.9 ° C when the pipe diameter has increased from 10 to 20 cm, which corresponds to an increase of 4.1 ° C. The reason is that the increase of the pipe diameter drags an increase of the exchange surface and also an increase of the air mass flow rate, which causes an increase in the air thermal inertia and consequently, the air will lose less heat. Besides, the daily mean efficiency as well as the coefficient of performance decreases when the diameter of the tube increases (face 9).

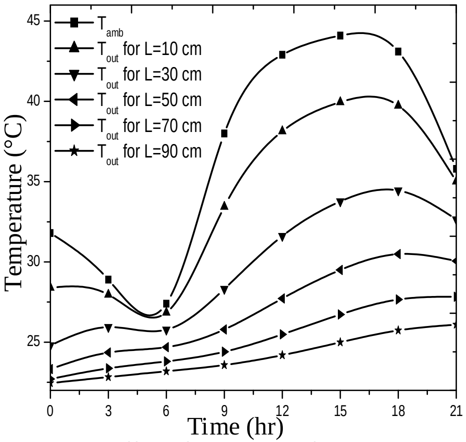

Figure 7. Effect of the length of pipe on the air outlet temperature

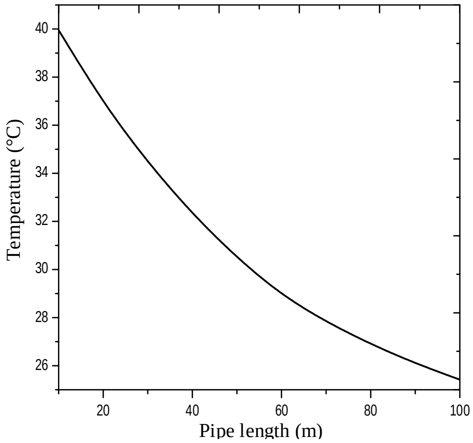

Figure 8. Variation of the maximal outlet temperature of air with the pipe length

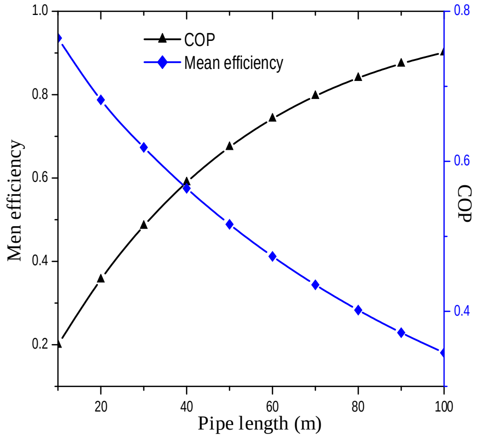

Figure 9. Variation of the mean efficiency and the coefficient of performance with the pipe length

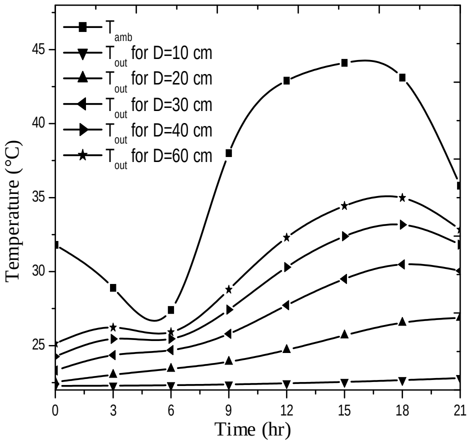

Figure 10. Effect of the pipe inside diameter on the air outlet temperature

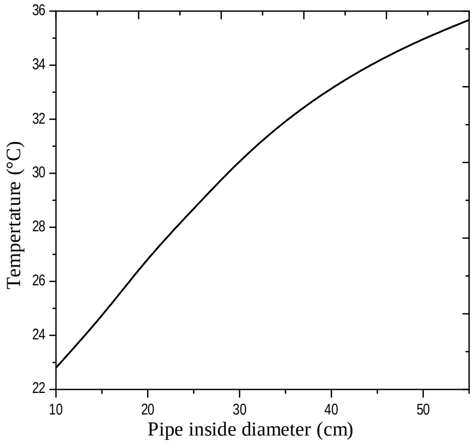

Figure 11. Variation of the air maximal outlet temperature as function of the pipe inside diameter

According to Figures (10 and 11), it appears that the air mean velocity has the same influence that the diameter for the reason that the increase of air mass flow is only the immediate result of the increase of the pipe section and/or the increase of the air velocity. However, the coefficient of performance decreases more quickly with the velocity.

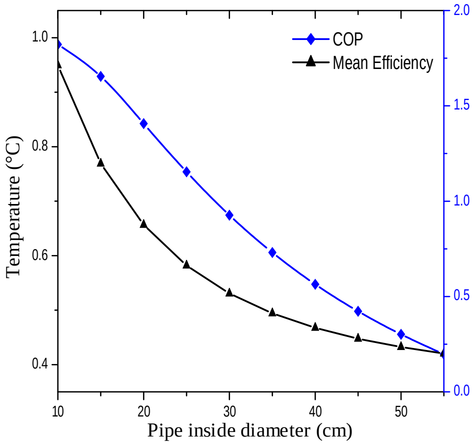

Figure 12. Effect of the pipe inside diameter on the mean efficiency and the coefficient of performance

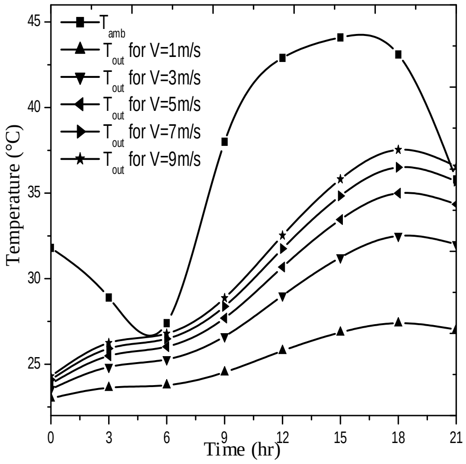

Figure 13. Effect of the air average velocity on the air outlet temperature

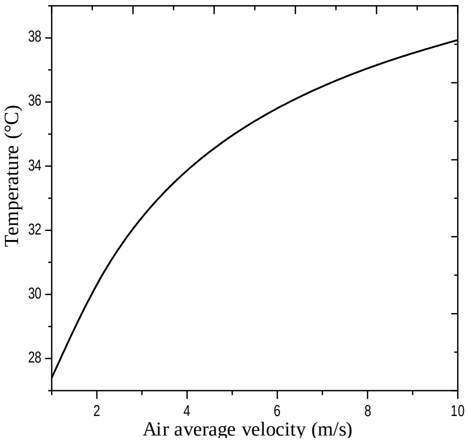

Figure 14. Variation of the air maximal outlet temperature as function of the air average velocity

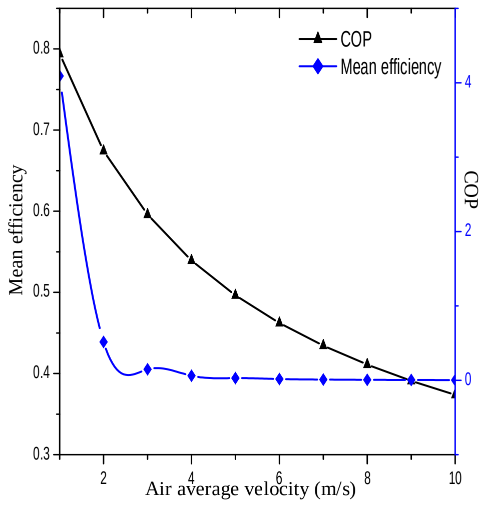

Figure 15. Variation of the mean efficiency and the the coefficient of performance with the air average velocity

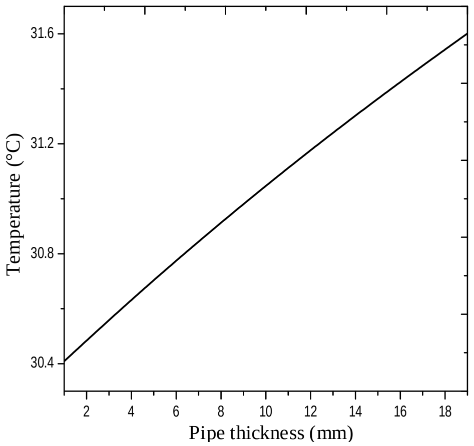

Figure 16. Effect of the pipe thickness on the air maximal outlet temperature

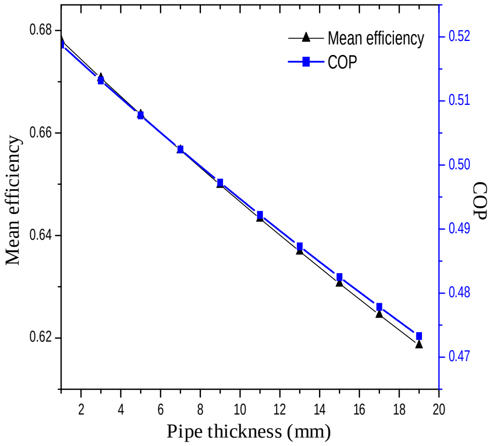

Figure 17. Variation of the mean afficiency and the coefficient of performance with the pipe thickness

To study the influence of the pipe thickness on the performance of the EAHE, we presented in Figure (16) the variation of the air maximal outlet temperature as a function of the pipe thickness. It appears that the effect of the latter on the air outlet temperature is too limited. For example, when the thickness is changed from 2 to 10 mm, the air maximal outlet temperature has increased only by 0.5 ° C. In addition, the Figure (17) shows the variation of the mean efficiency and the coefficient of performance as function of the pipe thickness. It should be noted that the mean efficiency as well as the coefficient of performance decreases almost linearly with increasing thickness. In fact, we have reported a decrease from 66 to 64% for the average efficiency and from 51 to 49% for the coefficient of performance when the pipe thickness increased from 5 to 11mm.

Conclusion

In this preset paper, we studied an earth-to-air heat exchanger intended to the cooling of buildings under the climatic conditions of the Algerian Sahara. This heat exchanger is constituted from a pipe in PVC buried at a depth of 2 ms in soil. We chose like period of study the month of July. After we have simulated the system, we led to the following results:

- The increase of air temperature is important when air enters in the pipe with a high temperature.
- The air outlet temperature is as much weaker than the heat exchanger is longer whereas it increases with the increase of the section of the tube and the speed of air. The effect of the pipe thickness on the air outlet temperature is negligible.
- The daily mean efficiency increases with the increase of the pipe length but it decreases if the pipe section, the air velocity or pipe thickness increases. Therefore, instead to use a pipe with a big section, it is preferable to make a parallel pipe network with a middle section and distanced sufficiently to avoid their coupling.
- The coefficient of performance decreases when the pipe geometric dimensions, pipe thickness or air velocity increases but in this last case, it decreases very quickly.

Nomenclature

Symboles

COPcoefficient of performance
Cpspecific heat (J/kg °C)
Ddiameter (m)
Hcconvective heat transfer coefficient (w/m² K)
Llength of the buried pipe (m)
Mmass (kg)
ṁ mass flow rate of air through the buried pipe (kg/s)
PrPrandtl number
PVCPolyvinyl chloride
ReReynolds number
Ttemperature (K)
Uoverall heat transfer coefficient (w/m² K)
Vvelocity of air (m/s)

Greek letters

ηmean efficiency
Δppressure loss (Pa)

Δxspatial increment in the direction of motion (m)
ρdensity (kg/m3)
λthermal conductivity (w/m °C)

Subscripts

aair
ambambient air
eexterior
(e)at the entrance of the EAHE
iinterior
sdisturbed soil
solundisturbed soil
(s)at the exit of the EAHE
tpipe
tottotal

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