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Earth-Air Heat Exchanger thermal performance in Egyptian conditions: Experimental results, mathematical model, and Computational Fluid Dynamics simulation



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ABSTRACT

In this paper, the thermal performance of an Earth-Air Heat Exchanger (EAHE) used for heating and cooling purposes is investigated under Egyptian weather conditions. The soil temperature profile and the temperature distribution of flowing air through horizontal Earth-Air Heat Exchanger (EAHE) is experimentally studied. Also, a mathematical model based on unsteady, one-dimensional and quasi-state is developed for energy conservation equation. Moreover, an explicit finite difference numerical method is used to solve the developed mathematical model with the help of MATLAB code. Finally, threedimensional, steady and double precision Computational Fluid Dynamics (CFD) ANSYS Fluent simulation model is established to predict the air and soil temperature. Whereas, the standard $\kappa - \in$ model is applied to simulate the turbulence kinetic energy of the flowing fluid. The mathematically developed model and CFD simulation result validated against experimental results. Good agreement is achieved with an average error and correlation coefficient of 2.09, 97% and 3.3 and 95.5% for CFD simulation and mathematical model respectively. The CFD model is used in a parametric investigation. A parametric study carried out to explore the impact of different parameters such as pipe diameter, pipe material, pipe space, pipe length and flowing fluid velocity. The results show that some of these parameters have noticeable results in air temperature. Whereas, the pipe diameter increases the air temperature decreases. The outlet air temperature declines from 20.4 °C to 18.7 °C as the pipe diameter expands from 2 to 3 in. Furthermore, as pipe length increases, outlet air temperature enhances. The temperature changes from 19.7 to 19.9 °C as the pipe length elongates from 5.45 m to 7 m. A bit change occurs in outlet air temperature from 19.7 °C to 19.8 °C when pipe space changes from 0.2 to 0.5 m. Moreover, three different pipe materials such as PVC, steel and copper are implied. The outlet air temperature was 19.7 °C in PVC pipe and 19.8, 19.8 °C for steel and copper respectively. So the conclusion is that the change in outlet air temperature for various pipe material is neglected compared with their prices. Finally, the effect of fluid velocity was investigated. Therefore, the outlet air temperature declines from 20.4 °C to 19.2 °C as air accelerates from 1 to 3 m/s. © 2016 Published by Elsevier Ltd.

1. Introduction

In the last decays, with the shadow of energy crises which strikes all over the world especially the developing countries like Egypt. Moreover, the energy use per capita increased to reach with annually population growth rate of 2.2 in 2014 according to world data bank [1]. The residential energy consumes about 26% of total energy use in Egypt [2]. Air conditioning used basically for cooling; it represents significant energy consumption in the residential building due to relatively high indoor air temperature in summer. It is vital to looking for alternative passive cooling and heating technique. The Earth-Air Heat Exchanger (EAHE) is one of passive technology used for heating and cooling purposes. Whereas, it has both economic and environmental benefits. It utilizes the thermal potential of the underground soil. The soil at a reliable depth has a constant temperature, which used as energy storage/sink in winter and summer seasons. Thermal performance assessment is authentic essential to optimize the design of EAHE. The design includes pipe diameter, pipe length, pipe material, heat exchanger

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Nomenclature						
A1 A7 constants			$T_{\rm evil}$ nine surrounding soil temperature in (K)			
A the cross section of a hollow cylinder of soil around the			$T_{\rm soli}$ the undisturbed soil temperature			
∞	pipe	WAHE	Water Air Heat Exchanger			
A _{inter}	the cross section area of soil from pipe outer radius to	Y_M	the contribution of the fluctuating dilatation incom-			
inter	the soil penetration depth r_{soil}		pressible turbulence to the overall dissipation rate			
A_n	pipe cross section area in (m^2)	TRNSYS	transient system simulation tool			
ĊFD	Computational Fluid Dynamics					
COP	coefficient of performance	Subscript	ts			
Ср	fluid specific heat in (kJ/kg K)	b	buovancv			
D	air thermal diffusivity in (m ² /s)	daily	daily			
е	error	i	inner			
EAHE	Earth-Air Heat Exchanger	inter	intermediate			
E-JUST	Egypt-Japan University of Science and Technology	k	kinetic			
ERE	Energy Resources Engineering Department	0	outer			
$G_{1\varepsilon}, G_{2\varepsilon}$	and $G_{3\varepsilon}$ constants	р	pipe			
G_k and C	G_b generation of turbulence kinetic energy due to mean	Р	pressure or normal stress in (Pa)			
	velocity gradients and buoyancy	pipe	pipe			
GSHP	ground source heat pump	soil	soil			
HP	heat pump	t	time			
K	the turbulence kinetic energy	undistur	bed undisturbed			
N and E	north and east	x	x direction			
N _t	no. of time steps					
N _x	no. of domain elements	Greek sy	mbols			
p DVC	the pressure or normal stress in (Pa)	β_1, β_2 and	ad β_3 constants			
oSoil	polyvinyl chioride	ω	daily cyclic in (s ⁻¹)			
Q _{air}	the effective thermal resistance	δ	daily penetration depth or disturbed layer thickness in			
	revolution por minute		(m)			
r	correlation coefficient	ho	fluid density in (kg/m ³)			
r and r.	outer and inner nine radius	v	fluid velocity in (m/s)			
R.	the nine conduction thermal resistance	μ	the dynamic viscosity in (Pa s)			
$r_{\rm pipe}$	the soil domain radius	Δx	the element size in (m)			
R	the soil conduction thermal resistance	€	the rate of dissipation			
SC	solar chimney	σ_{κ} and σ_{κ}	τ_e the turbulent Prandtl numbers for κ and \in			
S_{ν} and S_{ν}	user-defined source terms	Δt	time step in (s)			
т Т	flowing air temperature in (K)	∞	refer to ground surface			
Tamb	ambient dry bulb temperature in (k)	u, v, w	velocity component in x , y and z direction respectively			
anno						

configuration, buried depth and air flowing speed through buried pipes. Enormous numbers of researches done in this objective with different methodology. In this literature, the author presents some of them. Experimental methods carried out to study the visibility of coupled EAHE with the building. Carvalho et al. [3] showed that ground source heat pumps (GSHP) have high efficiency and high potential for building space conditioning. Moreover, it is suitable for electrical load management because of its load flexibility especially when it is combined with thermal energy storage capacity. Emmi et al. [4] integrated between solar thermal collector and GSHP to balance ground loads over a yearly cycle. They used this system to heat environments in a cold climate. They concluded that such system could assist in maintaining more efficient heat pumps (HP). Also, it reduces the total borehole length and the initial cost of installation. Li et al. [5] proposed a new system consist of the coupling between EAHE, solar collector and solar chimney (SC) used in totally passive air conditioning. Their experimental results show that the SC drove up to 0.28 m³/s (1000 m³/h) outdoor air into space. Furthermore, The EAHE provided a maximum 3308 W total cooling capacity. Finally, the coupled system almost covered the building design cooling load. Flaga-Maryanczyk et al. [6] coupled EAHE with the residential house for heating purposes in the cold climate in Australia. Their results indicate F that such coupling could conceal outside air temperature fluctuation. Also, it carried about 24% of the heating demand in February, and 15% of it during the period from December to April. Vaz et al. [7] investigated soil properties and characteristics, weather conditions in Brazil. They deducted that such system conveyed about 48% of heating and cooling demands. Demir et al. [8] examine the coefficient of performance (COP) of the GSHP in residential heating scope. Moreover, they analysis the relation between soil thermophysical properties and outlet air temperature from GSHP. Their results prove that soil thermal conductivity has a significant effect on fluid outlet temperature. Vaz et al. [9] explore the annual cyclic variation of flowing air temperature. Their results demonstrate that there are +8 and -4 °C temperature difference between outlet and inlet air temperature in heating and cooling respectively. Abbaspour-Fard et al. [10] claimed that COP of EAHE is 5.5 in cooling mode and 3.5 in heating mode at Iran climate conditions. Hatraf et al. [11] introduced a parametric study to evaluate the profile of air temperature inside tubes in the conditions of Algeria. They confirmed that pipe material has no effect on the thermal performance of the heat exchanger. Chel et al. [12] used transient system simulation tool (TRNSYS) to evaluate the dynamic thermal performance of residential building coupled with EAHE and Water Air Heat Exchanger (WAHE). They concluded that EAHE and WAHE had a reduction of the annual heating consumption of 66% and 7% respectively. Gao et al. [13] studied experimentally the benefits of combination between a rain garden and GCHE. They found that the increment in the soil moisture content led to enhancement in the

soil thermal conductivity which improves the thermal performance of GCHE. Also, their results show that water immigration could occur when the sandy soil has low moisture content, for example, 0.1 m³/m³. Kang et al. [14] proposed a novel coupled system between GSHP with heating and power system. Also, they examine the performance characteristics for various load conditions. Their results concluded that their system saves energy compared with tradition system. Investigated the thermal performance of a recently developed vertical spiral-shaped configuration used in geothermal pile-foundation heat exchanger. Also, they compared the performance with other configuration such as 1-U-shaped and 1-W-shaped in cooling mode. They concluded that spiralshape configuration with the series connection was the best performance. Allaerts et al. [15] proposed the coupling between active regeneration system and GCHE to reduce the required area and reduce the initial cost. Their results indicate that the cost reduction reaches to 47%. Awani et al. [16] coupled the flat plate collector and vertical heat exchanger to the heating greenhouse. They concluded that such heating system could save energy and could be competitive with the traditional heating system.

On the other hand, computer program simulation technique used by other researchers. Ahmed et al. [17] studied the impacts of air velocity, air temperature, relative humidity and soil tempera on room cooling performance. They used ANSYS Fluent for assessment the performance of EAHE connected to the room. Their results demonstrate that room temperature decreased with 1.05 °C which leads to saving about 579.91 kW annual energy. Javier et al. [18] developed a new simplified model to simulate single U-tube ground heat exchangers. Their model uses the electrical analogy to model heat transfer within the borehole to estimate heat flow to the surrounding ground.

Ahmed et al. [19] used Computational Fluid Dynamic (CFD) simulation software ANSYS FLUENT to investigate the effect of several parameters. Based on a two-dimensional model. Although, they did not take into account soil domain. They concluded that the values of 1.5 m/s air velocity, 60 m pipe length, 0.062 pipe diameter and 0.003 m pipe thickness and 8 m pipe depth were the best combination give the optimal performance. They success to reduce the indoor temperature with a value of 4.11 °C. Kaushal et al. [20] developed two-dimensional simulation model. They utilize a commercial CFD ANSYS FLUENT to examine the heat transfer in EAHE in the Western Himalayan region. The results reveal that air temperature difference reaches to 14.4 K. Rodrigues et al. [21] examine using multi configurations to snap the maximum thermal performance. Furthermore, they study five different configurations. They concluded that final configuration which consists of two rows and two columns had the best performance. Their configuration promotes thermal performance up to 73% and 115% for cooling and heating respectively. Mathur et al. [22] they consider the transient conditions for three different soil conditions. The results justify the fact that soil thermal conductivity with a value of 1.28 W/m K boost the performance by 5.04%. Moreover, another research used the same simulation tool to emphasize the same results of previous researchers. Yang et al. [23] developed an approach to examine the performance of EAHE working in the harmonic thermal environment. The harmonic environment characterized by temperature phase shifting. Barakat et al. [24] present a mathematical model to predict the performance of gas turbine coupled with EAHE. The results show increment in both efficiency and output power with values of 9% and 4.8% respectively. Bezyan et al. [25] investigated the thermal performance of a recently developed vertical spiral-shaped configuration used in geothermal pile-foundation heat exchanger. Also, they compared the performance with other configuration such as 1-U-shaped and 1-W-shaped in cooling mode. They concluded that spiral-shape configuration with the series connection was the best performance. Erbay and Hepbasli [26] investigated the exergy of GSHP drying system used in food drying. Their results indicate that the most critical system component is the condenser due to the design standpoint.

From this literature, it is evident that a great analysis has done in this aspect. Some of them in experimental approach and the other by program aided simulation. However, by the deep insight of these methods, it is clear that there is fewer experimental research done to examine serpentine heat exchanger configuration. Moreover, none of the previous papers measure soil temperature changes with depth and time. However, a few researchers taking into account the effect of inlet and outlet pipes as well as the soil temperature distribution around EAHE pipes. Finally, few investigations carried out to study the performance of EAHE operation in North Africa, especially in Egypt. In this paper, the author tried to fill this gap. The air temperature distribution through 5.5 m long, 2 in, diameter, PVC pipe designed in serpentine heat exchanger buried at 2 m depth investigated experimentally. Whereas, the thermal performance of EAHE evaluated in summer and winter seasons under Egyptian conditions. Soil temperature variation with depth and time simultaneously measured and recorded. Also twodimensional, unsteady heat transfer model developed to predict air temperature variation with pipe length. Likewise, a threedimensional finite volume difference CFD simulation developed to solve momentum and energy equations for fluid domain simultaneously with energy equation for the soil domain. Then, a comparison between experimental and mathematical and CFD simulation model results accomplished to examine their validation. Finally, the validated CFD model applied in a parametric investigation to explore the effect of both design and operating parameter on soil and flowing fluid temperature. Design parameters just as pipe diameter, pipe length and pipe space distance while, operating parameters such as fluid velocity and pipe material. The parametric study conducted for the heating mode with specific conditions.

2. Experimental setup

Experimental measurements were carried out in Energy Resources Engineering Department (ERE) laboratory back yard. Which belongs to the campus of the Egypt-Japan University of Science and Technology (E-JUST) campus (Latitude/Longitude: N3 0°55'/E29°42'). Whereas, a trough with dimensions of $2 \times 2 \times 2$ m³ is dug and refilled with Loamy sand soil. A serpentine horizontal heat exchanger was buried at a depth of 2 m as shown in Fig. 1a. The experimental set-up schematic diagram is shown in Fig. 2. The heat exchanger composed of 5.5 m long horizontal PVC pipe with inner diameter 0.0508 m (2 in.), buried in flat land. In the time of digging the trough, a sample of soil took at a depth 2 m. The thermal properties of the soil samples measured by Hot Disk (TPS2500S, Made in Sweden) at room temperature conditions. The thermal conductivity of the soil is 2.806 W/m K. The soil moisture content was 6% content by mass content. It measured by Armfield tray dryer model number UOP8. The inlet end of EAHE pipe connected to a floor standing tray dryer with a tunnel. The tray dryer has an axial fan (maximum flow rate of 0.08712 m³/s and a top speed of 2800 RPM). Which adjusted manually to manipulate air velocity. In these experiments, the velocity varies from 1 to 3.9 m/s. Fifteen temperature sensors (T-type thermocouples) distributed along the length of the pipe. These thermocouples (T1 to T15) fixed inside pipe near its longitudinal center line. The distance between each thermocouple arranged as shown in Fig. 2. Five temperature sensors viz. T₁₆ to T₂₁ mounted at a depth of 0 m, 0.5 m, 1 m, 1.5 m, 2 m respectively from the ground surface. It fixed by the aid of vertical black iron pipe to measure soil temperature variation with depth and time as shown in Fig. 1b. The thermocouples



Fig. 1. Pictures of experimental setup (a) PVC heat exchanger and (b) soil temp profile measurements.



Fig. 2. Schematic diagram of EAHE where (1) arm-field tray dryer, (2) flow rate controller, (3) temperature controller, (4) on/off switch, (5) axial fan and (6) electric heater.

calibrated against a standard calibrated thermocouple type T (Beta calibrator TC-100 made in the USA). The error was in the normal range where the deviation between the thermocouples readings and that of the standard one was from +0.1 to +0.5 °C. A multipoint digital data logger (NEC DC 6100 remote scanning made in Japan) used to measure and record the temperature every minute. The ambient dry bulb temperature (T_{amb}) measured and recorded simultaneously. Measurements of heating and cooling modes carried out during the period from 16 December to 1st of January and from 2nd of August to 7 of August respectively.

3. Mathematical model

3.1. Energy conservation equation

A mathematical model is developed to predict air temperature variation with pipe length and time. Explicit finite difference method used to solve heat transfer equation applied with the following assumptions:

- (1) Unsteady state condition.
- (2) One dimension heat transfers by conduction.
- (3) Air is radiative non-participating media.
- (4) Air flows with constant and uniform velocity.
- (5) Fluid properties are constant and evaluated at inlet temperature.
- (6) Soil thermos-physical properties are constant and didn't influence by pipe presence.
- (7) The soil is homogeneous anywhere in the domain.
- (8) Fluid assumed to be incompressible with constant density and specific heat.
- (9) For daily variations, the heat conduction from the region near pipe is assumed to follow a quasi-steady state behavior.
- (10) Neglect heat transfers in the radial direction.

The change in heat exchange fluid (air) temperature predicted by applying energy conservation equation on control volume presented in Fig. 3.



Fig. 3. Cross section of identified control volume.

The general form of energy conservation equation with diffusion and convection term presented by Eq. (1) [27]

$$\frac{\partial \rho_I}{\partial t} + \underbrace{\operatorname{div}(\rho T u)}_{\text{Convection term}} = \underbrace{\operatorname{div}(k \text{ grad } T)}_{\text{Diffussion term}} + \underbrace{\operatorname{Q}_{\operatorname{air}}^{\operatorname{Soil}}}_{\text{Source Term}}$$
(1)

After applying the previous assumption Eq. (1) becomes as follows in Eq. (2).

$$\frac{\partial T}{\partial t} = D \frac{\partial^2 T}{\partial x^2} - \nu \frac{\partial T}{\partial x} + \frac{1}{\rho \, Cp \, A_p} Q_{\text{air}}^{\text{Soil}} \tag{2}$$

where *D* is air thermal diffusivity in (m^2/s) . ν is fluid velocity in (m/s). ρ is fluid density in (kg/m^3) . *Cp* is fluid specific heat in (kJ/kg K). *A_p* is pipe cross section area in (m^2) .

 $Q_{\rm air}^{\rm Soil}$ is the heat flux from/to the subsurface in (J/s m), it negative during the summer season where the soil acts as heat sink and positive during the winter season as soil works as heat source calculated from Eq. (3).

$$Q_{\rm air}^{\rm Soil} = \frac{T_{\rm soil} - T}{R} \tag{3}$$

where T_{soil} is pipe surrounding soil temperature in (K). *T* is flowing air temperature in (K).

 $R = R_{pipe} + R_{soil}$ is total thermal conduction resistance between flowing air and soil called effective thermal resistance and R_{soil} is soil conduction thermal resistance calculated by Eq. (4)

$$R_{\rm soil} = \frac{1}{2\pi \, k_{\rm soil}} \ln \left(\frac{r_{\rm soil}}{r_o} \right) \tag{4}$$

Finally Eq. (2) rearranged as shows in Eq. (5)

$$\frac{\partial T}{\partial t} = D \frac{\partial^2 T}{\partial x^2} - \nu \frac{\partial T}{\partial x} + \beta_1 (T_{\text{soil}} - T)$$
(5)

where

$$\beta_1 = \frac{1}{R\rho C_p A_p} \tag{6}$$

In the same manner, the change in soil temperature (T_{soil}) developed by applying energy balance equation on soil control volume as indicated in Eq. (7)

$$\frac{\partial I_{\text{soil}}}{\partial t} = \beta_2 (T - T_{\text{soil}}) - \beta_3 (T_{\text{soil}} - T_\infty)$$
(7)

where

$$\beta_2 = \frac{1}{(R \ \rho_{\text{soil}} C_{P\text{soil}} A_{\text{inter}})} \tag{8}$$

$$\beta_3 = \frac{1}{(R_{\infty} \rho_{\text{soil}} C_{\text{Psoil}} A_{\infty})} \tag{9}$$



Fig. 4. Discretization of fluid and soil domains.

3.2. Finite difference approximation

The explicit finite difference approximation used to solve Eqs. (5) and (7). A grid of N_x nodes used to represent the physical domain. EAHE consists of 5.45 m length PVC pipe with 2 in. diameter and 3 mm thickness. The heat exchanger configuration and the discrete domain show in Fig. 4.

Explicit discretization method used to discretize both time and space domains. Space domain discretizes to some elements $N_x = 37$ with an element size Δx . On the other hand, time domain discretizes to a number of time steps N_t with a time step of Δt selected to achieve the condition indicated in Eq. (10) [28].

$$\Delta t \leqslant \frac{\left(\Delta x\right)^2}{4D} \tag{10}$$

After applying explicit finite difference method to both sides of Eq. (5) moreover, Eq. (7), these equations could be arranged and rewrote as indicated by the following equations.

$$T_{i}^{j+1} = A_{1} T_{i}^{j} + A_{2} T_{i+1}^{j} + A_{3} T_{\text{Soil},i}^{j} + A_{4} T_{i-1}^{j}$$
for $2 < i < N_{x} - 2$ and $1 < j < N_{t}$
(11)



$$T_{i}^{j+1} = A_1 T_i^j + A_3 T_{\text{soil},i}^j + A_4 T_{i-1}^j$$
 for $i = N_x - 1$ and $1 < j < N_t$ (12)

$$T_i^{j+1} = T_{i-1}^{j+1}$$
 for $i = Nx$ and $1 < j < Nt$ (13)

$$T_{\text{soil},i}^{j+1} = A_5 T_{\text{soil},i}^j + A_6 T_i^j + A_7 T \bigg\} \quad \text{for } 1 < i < Nx \quad \text{and} \quad 1 < j < Nt$$
(14)

With initial condition indicated in Eqs. (15) and (16)

$$T_i^j = T_{\text{amb}}$$
 for $1 < i < Nx$ and $j = 0$ (15)

$$T_{\text{soil},i}^{j} = T_{\text{undisturbed}}$$
 for $1 < i < Nx$ and $j = 0$ (16)

where constants A1, ..., A7 calculated as, follow

 $\begin{array}{ll} A_1 = \Delta t \times \left(\frac{1}{\Delta t} - \frac{2D}{\Delta x^2} - \frac{v}{\Delta x} - \beta\right), & A_2 = \Delta t \times \left(\frac{D}{\Delta x^2}\right), & A_3 = \Delta t \times (\beta), \\ A_4 = \Delta t \times \left(\frac{D}{\Delta x^2} + \frac{v}{\Delta x}\right), & A_5 = \Delta t \times \left(\frac{1}{\Delta t} - \beta_2 - \beta_3\right), & A_7 = \Delta t \times (\beta_3) \quad \text{and} \\ T_{\text{undisturbed}} \text{ is undisturbed soil temperature.} \end{array}$







Fig. 6. Solid and fluid domains dimensions used in CFD simulation.

The code was written and solved with the help of MATLAB to solve these equations through steps indicated by the flow chart in Fig. 5.

4. CFD simulation set-up

To simulate both air flow and heat transfer processes in an EAHE system, a Computational Fluid Dynamics (CFD) software package ANSYS FLUENT v. 15.0.7 used in this study. ANSYS Fluent software contains the broad physical modeling capabilities needed to model flow, turbulence, heat transfer, and reactions. It has turbulence models which cover a variety of applications. Moreover, the effects of other physical phenomena, such as buoyancy and compressibility can be modeled. Finally, natural, forced, and mixed convection heat transfer model with or without conjugate heat transfer and porous media can also be modeled [29]. The threedimensional geometry is drawn for both solid and fluid domains with dimensions indicated in Fig. 6. CFD based analysis has been employed to resolve the temperature field around the horizontal buried pipe of EAHE, using an unstructured grid for soil and fluid domain. Mesh cell number optimized to decrease computational time and get a most accurate solution. To do this, mesh independent solution technique is applied to have the best grid density. Since the temperature changes more sharply near the pipe wall, the grid must be fine in that area while it is more coarse farther away from the pipe wall. A steady with implicit interpolation scheme used for simulation. The SIMPLE algorithm applied for the pressure-velocity coupling in the segregated solver. In the case of steady state iteratively solved a problem like our case, it is not necessary to fully resolve the linear pressure-velocity coupling, as the changes between consecutive solutions are no longer small. So, for relatively uncomplicated problems in which the pressurevelocity coupling limits the convergence, SIMPLE scheme preferred to obtain a converged solution more quickly [30]. The second order upwind scheme adopted for the discretization of the governing

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Thermo-physical properties of material used in CFD simulation.

Material	Density (kg/m ³)	Specific heat capacity (J/kg K)	Thermal conductivity (W/m K)
Air	1.225	1006	0.024
Soil	2050	1840	2.806
PVC	1380	900	0.16

equations. The convergence criteria for all variables were set to be 10^{-6} . The standard turbulent $\kappa - \in$ model: the turbulence kinetic energy, κ and its rate of dissipation, \in is applied to model the transport of turbulent kinetic energy. The $\kappa - \in$ model is one of the most common turbulence models. It includes two extra transport equations to represent the turbulent properties of the flow. Also, it gives good results for wall bounded and internal flows with small mean pressure gradients. The turbulence model selected for the thermal modeling of the flow passing through the buried pipes was turbulent (Reynolds Number, Re > 4000), where Reynolds number ranged from 4900 to 13,000. The model satisfies definite mathematical constraints on the Revnolds stresses and is consistent with turbulent flow physics [17]. Moreover, this allows a two equation model to account for history effects like convection and diffusion of turbulent energy. Whereas The K-epsilon model has been shown to be useful for free-shear layer flows with relatively small pressure gradients. Similarly, for wallbounded and internal flows [31].

In the present study the following assumptions are used:

- The air is incompressible.
- The soil is homogeneous, and its physical properties are constant.
- The temperature of soil surrounding the pipe remains constant.
- The property of the pipes and ground materials do not change with temperature.
- Engineering materials used in the CFD model are isotropic and homogeneous.

The thermal parameters of different engineering materials used in the simulation listed in Table 1.

The boundary condition applied in CFD simulation took as follows. Uniform inlet velocity values entered from experimental results varies from 1 to 3.9 m/s. Also, its direction is normal to inlet with 5% turbulence intensity and 0.0508 m characteristic inlet length. The upper and bottom soil domain walls assumed to be isothermal walls at the temperature measured by experiment. Furthermore, soil domain side walls assumed to be adiabatic walls. Moreover, soil pipe interface coupled heat transfer condition is taken. Also, non-slip surface for momentum condition is assigned. Finally, zero gauge pressure applied to pressure outlet condition. The governing transport equation which FLUENT based on can be found in Table 2 in supplementary material.

5. Results

5.1. Experimental results

Experiments conducted during the period from 16 December to 1st of January and from 2nd of August to 7th August. The first period selected to investigate the potential and capability of using the system to absorb heat from the surrounding soil to heat air flowing through buried pipes. On the other during the second period, the heat dissipated to the adjacent soil to cool the air passed through the heat exchanger. From weather data recorded simultaneously with experiments, the ambient air temperature varied from



Fig. 7. Soil temperature profiles for (a) winter condition and (b) summer condition.



Fig. 8. Inlet, outlet, ambient air temperature and ground temperature at 2 m depth and inlet air velocity for (a) winter condition and (b) summer condition.

16.3 °C at 16 December to 10.0 °C at 1st January with an average value of 14.7 °C during heating condition. Also, it ranged from 37.3 °C at 6th August to 32.1 °C at 7th August with a mean value of 32.9 °C during cooling condition.

5.1.1. Soil temperature distribution profile

Separate experiments are conducted to understand the ground thermal behavior variation with depth at the different time. Whereas, this variation indicated as shown in Fig. 7a and b.



Fig. 9. Temperature variation with pipe length for (a) winter condition and (b) summer condition.



Fig. 10. Temperature distribution along pipe length at different air velocity (a) at $V_{air} = 1$ m/s.

Table 2Error and correlation coefficient of CFD simulation and mathematical model.

	CFD simulation			Mathematical model				
	Heating		Cooling		Heating		Cooling	
Velocity	e, %	r, %	e, %	r, %	e, %	r, %	e, %	r, %
1	1.63	97	1.54	93	2.97	90	1.76	98
1.5	2.22	98	0.84	99	3.27	93	2.52	97
2	3.08	98	0.77	98	4.58	94	2.24	96
2.5	3.05	98	0.46	99	3.51	95	3.13	96
3	2.46	97	1.11	98	2.35	94	3.17	96
3.9	4.94	99	3.06	97	3.56	98	6.62	96
Average	2.89	97	1.29	97	3.37	94	3.24	97

Fig. 7a indicates that temperature varies from 14.4 °C at earth surface to 22.2 °C at 2 m soil depth over the duration from 16 December 2014 to 1th January 2015. While, Fig. 7b shows a divergence between 34.1 °C and 26.8 °C. From these graphs, it is concluded that there is -7.7 °C and +7.3 °C differences between the soil surface and the soil at 2 m depth. The negative sign indicates that

surface of the ground temperature is less than the ground temperature at 2 m depth; this character can be utilized as heating potential. However, positive sign shows the adverse condition.

5.1.2. Inlet and outlet air temperature of an EAHE

Fig. 8a and b indicates the variation of inlet air temperature, outlet air temperature, ambient air temperature, soil temperature at 2 m depth and the exit air velocity with time. Fig. 8a illustrates that the experiments start at 16 of December with 1 m/s inlet air velocity to reach to 3.9 m/s at 1st January. The average inlet air temperature is 2.7 °C more than ambient air temperature. However, it is 3.7 °C less than outlet air temperature at 2 m depth. Therefore, it can be concluded that inlet air temperature depends on ambient air temperature at the depth of buried despite air velocity. Consequently, inlet air velocity has less effect on exit air temperature. So we can say that convective heat transfer between flowing air and pipe inner surface has less influence compared to



Fig. 11. Temperature contours at fluid flow velocity of 1.5 m/s at (a) inlet plane, (b) outlet plane, (c) horizontal plane at depth 2 m and (d) lateral plane normal to z-axis.

conductive heat transfer between outside pipe surface and surrounding soil. Consequently, the soil temperature increases, the outlet air temperature increases. The same explanation can be said for Fig. 8b. In vice versa manner.

5.1.3. Thermal performance of EAHE

Flowing air temperature variation with pipe length for both heating mode and cooling mode respectively shown in Fig. 9a and b. Fig. 9a indicates that the inlet and outlet pipes did not affect so much the heat transfer process between flowing air and soil. The most dominant heat transfer process occurs via horizontal pipes. However, the length of exit pipe affects the performance of the heat exchanger negatively. Where it is affected by soil surface temperature, from Fig. 9a, it can be seen that air temperature increases as tube length increase until reaches to the beginning of outlet pipe at pipe length of 8.7 m the temperature decreases with pipe length due to heat losses to surrounding cold soil. The reverse happened in Fig. 9b. From these results, it can be recommended to insulate the exit pipe length to keep the good thermal performance of heat exchanger. Also from these figures, it is evident that outlet air temperature varied with 2 °C between different volume flow rates except for flow rate of 28 m³/s. At which the ambient air temperature was at its lowest value of 9.9 °C; inlet air temperature was 15.0 °C and soil temperature at 2 m depth was 21.0 °C.

5.2. Validation

Mathematical and CFD simulation models are validated against experimental results. The validation carried out for both heating and cooling operation mode. Also, validation presented at a different fluid flow velocity. Fig. 10 indicates that CFD simulation model has best fit with the experimental result than a mathematical model with an error (e) and correlation coefficient (r) as shown in Table 2.

These values indicate that CFD simulation fit experimental results than a mathematical model. This result returns to the assumptions taken into account in the mathematical model such as one-dimensional and neglect heat transfer in radial direction assumptions which are relaxed in CFD simulation. Whereas, the heat transferred from one pass to the other through the annulus soil part is overlooked in the mathematical model. This amount of heat plays a significant role in the heat transferred by conduction. Consequently, as soil thermal conductivity increased this value will increase.

5.3. CFD results

The validated CFD simulation model used to visualize and analyze the temperature contours for both solid and fluid domains. Temperature contours of both fluid and soil domain are shown in Fig. 11. The right side displays the cooling mode while the left side indicates the heating mode. It can be noticed from these contours that there is a significant difference between heating and cooling modes. The first difference is that soil temperature variation from the surface of the ground to depth 2 m contradicts in both modes as shown in Fig. 11a and b. The temperature increased with depth in heating mode and decreased in cooling mode. Moreover, the fluid temperature increased and decreased with increasing pipe length in heating and cooling mode respectively. Temperature contour in the plane cross the inlet pipe indicates that air temperature gradient did not exceed 1.5 °C in both modes at different flow velocities. On the other hand, the gradient in fluid temperature at plane cross outlet pipe did not exceed 0.5 °C because of glass wool insulation around outlet pipe presented in Fig. 11b. Temperature contour at horizontal plane crosses horizontal heat exchanger indicates that the temperature gradient may exceed 3.5 °C shown in Fig. 11c. So the significant heat exchange occurs at the horizontal part of the EAHE. So the length of this pipe part plays a important role in heat transfer process. Contour (C) shows that the thermal energy propagation from/to soil more obvious around and near inlet pipe and that around outlet pipe this because the temperature difference between fluid and soil decreased in the direction of fluid flow. On the other hand, these propagation become more evident with increasing fluid flow velocity because as fluid velocity increased as there is no enough time for heat to exchange between fluid and soil still large than what happens at low air velocity.

5.4. Parametric study

The validated CFD model used to explore the effect of design parameters and operating parameters on flowing fluid temperature distribution along pipe length. Design parameters such as pipe diameter, pipe length, and space between pipes passes. While, operating parameters such as pipe material, and flowing fluid velocity. The following figures indicated the effect of each parameter in heating mode operation. The parametric study conducted for heating mode only. Firstly, Fig. 12 indicates air temperature variation with pipe length at different pipe diameter of 2, 2.5 and 3 in. It can be concluded that as pipe diameter increased, the air temperature decreased. This because the decrement in the space between pipes passes. Which leads to increment in heat transfer surface area. Consequently, the convection heat transfer coefficient decreases. Also, this leads to increase the ability of soil to transfer heat between heated pipes passes from hot to the adjacent cold one. The outlet air temperature decreased from 20.4 °C to 18.7 °C as pipe diameter increased from 2 to 3 in.

The effect of pipe length on outlet air temperature is shown in Fig. 13. It is evident from the figure that as pipe length increases, outlet air temperature increases. As pipe length increases, the air takes enough time to exchange thermal energy with surrounding soil. The figure shows that temperature increases from 19.7 to 19.9 °C as pipe length increases from 5.45 m to 7 m. It can be concluded from this result that the increment in outlet air temperature of 0.2 °C did not proportionate with the increment in pipe length of 1.55 m.

Figs. 14 and 15 show a change of air temperature with pipe length at three different passes spacing distance of 0.2, 0.3 and 0.5 m. It seems from the figure that there is a bit change in outlet air temperature from 19.7 °C to 19.8 °C corresponding to 0.2 and 0.5 m pipe space respectively. This result because as the distance between pipes decreases the temperature of the soil adjacent to



Fig. 12. Air temperature variation with pipe length for diameters of 2, 2.5 and 3 in.



Fig. 13. Outlet air temperature variation with pipe length.



Fig. 14. Air temperature variation with pipe length for pipe space of 0.2, 0.3 and 0.5 m.

the pipe decrease as well which decrease heat transfer rate; it can be seen from temperature contour shown in from Fig. 15.

The effect of three different pipe material of steel, copper and PVC on flowing air temperature shown in Fig. 16. It concludes that air flows in PVC pipes have less temperature compared with copper and steel pipes. This because of high thermal resistance of PVC pipes compared with less thermal resistance for both copper and steel pipes. High thermal resistance is a result of low thermal conductivity which reaches to 0.16 W/mK for PVC while high



Fig. 16. Flowing air temperature variation with pipe length for three different pipe material of PVC, steel and copper.



Fig. 17. Flowing air temperature variation with pipe length for different fluid velocity of 1, 2 and 3 m/s.

thermal conductivity reaches to 16.27 and 387.6 W/m K for steel and copper respectively. Outlet air temperature was 19.7 °C in PVC pipe and 19.8, 19.8 °C for steel and copper respectively. So the conclusion is that the outlet air temperature variation between different pipe material is neglected compared with their prices. The effect of flowing air velocity on flowing air temperature variation with pipe length indicated in Figs. 17 and 18. Three different fluid velocities of 1, 2 and 3 m/s applied. It is seen from these figures that fluid velocity has a significant impact on air temperature.



Fig. 15. Temperature contour of soil and flowing air domain in the horizontal plane of 2 m depth at different pipe space (a) 0.2 m, (b) 0.3 m and (c) 0.5 m.



Fig. 18. Temperature contour of soil and flowing air domain at horizontal plane of 2 m depth at different fluid velocity (a) 1 m/s, (b) 2 m/s and (c) 3 m/s.

Therefore, the increment in fluid velocity decreases the rate of heat transfer from soil to air as the residence time needed for thermal energy exchange proportionally decrease. The outlet air temperature decreased from 20.4 °C to 19.2 °C as fluid velocity increases from 1 to 3 m/s.

6. Conclusion and future work

In this paper, the temperature distribution of flowing air through horizontal Earth-Air Heat Exchanger (EAHE) experimentally studied. A mathematical model based on unsteady, onedimensional and quasi-state energy conservation equation developed for flowing fluid. Also, the explicit finite difference numerical method used to solve the developed heat transfer equation with the help of MATLAB code. Finally, 3D, steady and double precision CFD Fluent simulation model applied to predict the air temperature. The transport equation for the standard $\kappa - \in$ model applied to simulate the turbulence kinetic energy of the flowing fluid. Validation of developed mathematical model and CFD simulation are investigated. A good agreement found between models and experimental results with an average error and correlation coefficient of 2.09, 97% and 3.3 and 95.5% for CFD simulation and mathematical model respectively. The validated model used in the parametric study to investigate the effect of both design parameters and operating parameters. The effect of pipe diameter, pipe length, pipe space, pipe material and flowing fluid velocity are investigated. The following conclusions could be extracted

- As pipe diameter increased the air temperature decreased. This because of the increment in the heat transfer surface area. So the convection heat transfer coefficient decreases. The outlet air temperature decreased from 20.4 °C to 18.7 °C as pipe diameter increased from 2 to 3 in.
- As pipe length increases, outlet air temperature increases. This is returned to more time the flowing air takes in thermal energy exchange with surrounding soil. It can be seen from the figure that temperature increases from 19.7 to 19.9 °C as pipe length increases from 5.45 m to 7 m. It could be concluded from this result that the increment in outlet air temperature of 0.2 °C did not proportionate with the increment in pipe length of 1.55 m.

- It seems from the figure that there is a bit difference between outlet air temperature of 19.7–19.8 °C corresponding to 0.2 and 0.5 m pipe space respectively.
- It concludes that air which flows in PVC pipes has less temperature compared with that passes through copper and steel pipes. This because of high thermal resistance of PVC pipes compared with less thermal resistance for both copper and steel pipes. High thermal resistance is a result of low thermal conductivity which reaches to 0.16 W/m K for PVC while less value for high thermal conductivity reaches to 16.3 and 387.6 W/m K for steel and copper respectively. Flowing air outlet temperature was 19.7 °C in PVC pipe and 19.8, 19.8 °C for steel and copper respectively. So the conclusion is that the outlet air temperature difference from different pipe material is neglected compared with their prices.
- Therefore, the increment in fluid velocity decreases the rate of heat transfer from soil to flowing air as the residence time needed for thermal energy exchange proportionally decrease. The outlet air temperature decreased from 20.4 °C to 19.2 °C as fluid velocity increases from 1 to 3 m/s.

In future work; the author plans to investigate the thermal performance of passive heating, cooling and ventilation system of the residential building. This system utilizes the capability of EAHE to provide the building required heating and cooling loads. Moreover, residential building coupled with Trombe wall and Solar Chimney. These two techniques used for passively ventilation and heating purposes. These two methods rely on a variation of indoor air density, which generates the buoyancy force. The buoyancy effect is the driving force of natural draft initiated in the building which in turns suck the outside air to passing through EAHE. So the overall system will operate in a passive manner. However, in future work, the results of complete continuous one year will be one of the recommended objectives.

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Appendix A. Supplementary material

Supplementary data associated with this article can be found, in the online version, at http://dx.doi.org/10.1016/j.enconman.2016. 05.053.

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