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Numerical simulation of a vertical "U" type terrestrial heat exchanger using coastal zone boundary conditions

Simulación numérica de un intercambiador de calor terrestre vertical tipo "U" usando condiciones de frontera de zona costera

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Abstract

Among the various factors that affect the energy produced a detailed numerical simulation of a ground heat exchanger assuming the mathematical formulacion of the governing ecuations (continuity, momentum and energy) and the energy balance in the wall has been carried out. This formulation requires the use of termo-physical properties, material properties and the ground; in this case, the experimental temperature profile in the ground were measured at Universidad Veracruzana, Coatzacoalcos campus (latitude 18°08'39" N, longitude west 94°28'36" and altitude 10 msnm).

Diffusivity, Control volumen, Depht

Resumen

El presente trabajo aborda la simulación numérica de un intercambiador de calor terrestre, basada en la formulación matemática de ecuaciones gobernantes de flujo (ecuación de continuidad, ecuación de movimiento, ecuación de energía) y la ecuación de transferencia de calor en elementos sólidos. Estas ecuaciones requieren información de propiedades termo-físicas del fluido, del material, y de la tierra; para este caso se utilizó como condición de frontera un perfil de temperatura de la tierra con datos que fueron medidos en la Universidad Veracruzana Campus Coatzacoalcos (latitud 18°08'39" N, longitud oeste 94°28'36" y altitud 10 msnm).

Difusividad, Volume de control, Profundidad

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Introduction

The present work shows the theoretical modeling and numerical simulation of a terrestrial heat exchanger using a temperature profile measured in a coastal area. In this case, the terrestrial heat exchanger has the ability to take advantage of the heat capacity and the relatively stable temperature of the subsoil for the purpose of cooling water, transferring heat from the fluid to the subsoil. The terrestrial heat exchanger is made up of two parallel vertical tubes, joined at the lower ends by a “U” -shaped return. The fluid enters through a tube, while through the union in a U it is redirected to the other tube until the fluid exits to the adjoining end.

About the works reported in the literature of terrestrial heat exchangers we can emphasize the following. According to Florides and Kalogirou (2008) it is important to have knowledge about the distribution of the subsoil temperature around the pipes of the terrestrial heat exchanger. The study was conducted under conditions in Cyprus, where the temperature of the earth is always cooler in the summer and warmer in the winter. Ally et al. (2015) presents the study of a heat exchanger coupled to a conventional water heating system, obtaining an increase in fluid temperature from 37.8 ° C to 49 ° C. Pu et al. (2014) investigated the effect of the Reynolds number, the diameter of the pipe and different configurations in the installation of terrestrial heat exchangers. The presented study is validated by experimental soil thermal response tests.

The Authors present two simulations, one physics of the installation model with the GAMBIT software and another in the ANSYS FLUENT 14.0 software to calculate the flow field, temperature, pressure and heat transfer between the fluid and the ground.

In the terrestrial heat exchanger there are physical phenomena such as: conduction in the pipe, convection and loss of fluid pressure. The present work aims to develop a computational model based on the governing equations of continuity, momentum and energy that describe the phenomenology of a terrestrial heat exchanger. The foregoing in order to help engineers and researchers in the future design, construction, optimization and control of this class of systems.

Determination of the temperature profile and thermo-physical properties of the earth

To determine the temperature profile in the subsoil, temperature sensors type T (copper-constantan) were installed at different depths of the subsoil for a temperature range of 0 to 350 ° C, with an accuracy of ± 0.5 ° C calibrated with a AMETEK equipment model CTC-140 at a range of -30 ° C to 140 ° C. The calibration error was calculated in an order of ± 0.02 ° C; data capture was performed using an Agilent model 34972A acquirer.

The measurements were made at the Universidad Veracruzana Campus Coatzacoalcos, latitude 18 ° 08'39 ”N, west longitude 94 ° 28'36” and altitude 10 meters above sea level. Figure 1 shows the temperature profile of the subsoil, down to a depth of 1m.

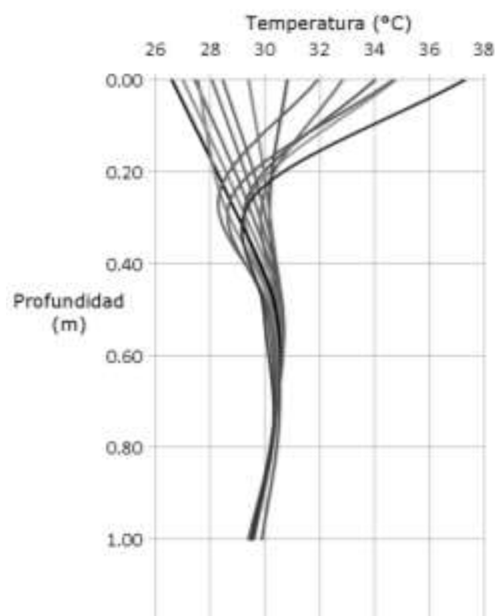

Figure 1 Temperature profile in the subsoil

Figure 1 shows the geothermal profile, where the temperature remains relatively constant at depths greater than 0.8m. The thermal diffusivity of the soil was determined "in situ" by the harmonic method of thermal wave phase shift, using linear regression in determining the depth at which the surface temperature and the temperature in the subsoil are lagged from the cycle by one period. full. Equation 1 was used to calculate the thermal diffusivity of the soil.

$$\alpha = \frac{Z^2}{4\pi\varphi} \quad (1)$$

Where α is the diffusivity in (m2 / s), Z is the depth at which the wave is out of phase one complete cycle (m) and φ is the period (s).

The density ρ and specific heat C_p of the subsoil were determined in the laboratory and the thermal conductivity λ was calculated from equation 2 of thermal diffusivity α.

$$\alpha = \frac{\lambda}{\rho C_p} \tag{2}$$

Mathematical model

The governing equations that explain the heat transfer and fluid-dynamic of the working fluid in the terrestrial heat exchanger and the assumed mathematical formulation for the solid element (wall) are described below.

Heat transfer and fluid-dynamic flow

According to García-Valladares (2004), a control volume (VC), Figure 2, is a finite volume that delimits a physical space corresponding to partial or global zones of the thermal unit. For the formulation of equations that govern this system in flux, the equations of continuity, momentum and energy of each VC are solved. The solution of the governing equations for a single phase is described in Colorado et al. (2011).

The hypotheses that are assumed for the formulation of this model are:

- One-dimensional flow.
- Pure fluid (water).
- Radiation heat transfer is neglected.
- Newtonian fluids.
- Constant diameters and roughness in the pipe

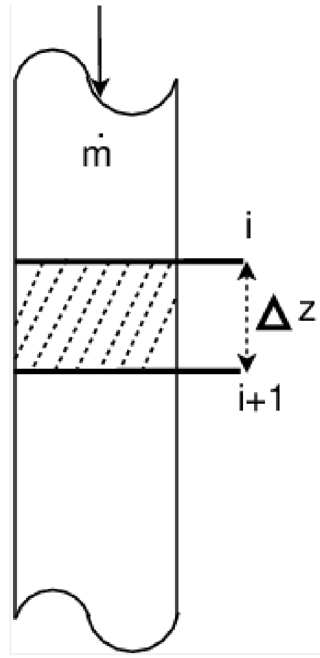

Figure 2 Control volume in a mass flow down pipe.

The ordinary differential equations that describe the fluid-dynamic and thermal behavior of the working fluid within the terrestrial heat exchanger are described below. The mass flow M at the outlet of the control volume is obtained by the discretization of the Continuity equation:

$$M_{i+1} = M_i - \frac{A\Delta z}{\Delta t}\rho_l \tag{3}$$

Where Δt is the fixed time step, ρ_l is the density of the working fluid and A is the area of the passage section. Once the mass flow at the outlet has been calculated, the liquid velocity v_l is calculated as:

$$v_l = \frac{M}{\rho_l A} \tag{4}$$

The discretization of the equation of Motion allows to find the outlet pressure p:

$$p_{i+1} = p_i - \frac{\Delta z}{A}\left[\frac{\bar{f}_l}{4}\frac{\bar{M}^2}{2\rho_l A^2}P + \bar{\rho}_l A g sen\theta + \frac{M(v_l)_i^{i+1}}{\Delta z} + \frac{\bar{M}-\bar{M}^0}{\Delta t}\right] \tag{5}$$

Where f̅l is the Darcy-type friction factor, g is gravity and θ is the angle of inclination of the tube. The outlet temperature T is obtained by subtracting the continuity equation multiplied by the specific energy in the center of the control volume from the Energy equation:

$$T_{i+1} = \frac{2qP\Delta z - M_{i+1}a + M_i b + \frac{A\Delta z}{\Delta t}c}{Cp_{i+1}\left[M_{i+1} + M_i + \frac{\bar{\rho}_l^0 A\Delta z}{\Delta t}\right]} \tag{6}$$

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Where:

$a = (v_l)^2_{i+1} + gsen\theta\Delta z - Cp_i T_i$

$b = (v_l)^2_i - gsen\theta\Delta z + Cp_i T_i$

$c = (p_i + p_{i+1}) - (p_i^0 + p_{i+1}^0) + (\tilde{\rho}_l^0)\left((Cp_i^0 T_i^0 + Cp_{i+1}^0 T_{i+1}^0) - Cp_i T_i\right) - \left(\tilde{\rho}_l \left(\frac{v_i + v_{i+1}}{2}\right)^2 - \tilde{\rho}_l^0 \left(\frac{v_i^0 + v_{i+1}^0}{2}\right)^2\right)$

Inner wall energy equation

The tube or wall of the terrestrial heat exchanger is modeled according to the following hypotheses in order to develop the heat conduction equation.

– A one-dimensional temperature distribution is assumed
– Heat exchanged by radiation is neglected.

A characteristic control volume is shown in figure 3 where P is the central node, E and W the neighboring nodes, where "e", "w", "n", "s" are the faces of the control volume.

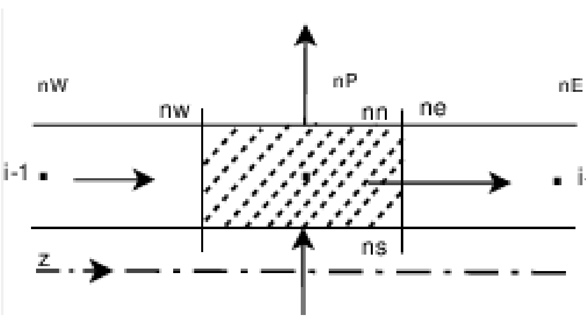

Figure 3 Heat flow in solid elements.

Integrating the equation of energy over the control volume shown in figure 3, the following equation is obtained:

$$(\tilde{q}_s P_s - \tilde{q}_n P_n)\Delta z + (\tilde{q}_w - \tilde{q}_e)A = m\frac{\partial \tilde{h}}{\partial t} \quad (7)$$

Where the heat flux $\tilde{q}_s$ has been evaluated from its respective surface coefficient of heat transfer in free or forced convection and the heat fluxes by conduction are evaluated from the Fourier law, that is:

$\tilde{q}_e = -\lambda_e \left(\frac{\partial T}{\partial z}\right)$; $\tilde{q}_w = -\lambda_w \left(\frac{\partial T}{\partial z}\right)$;

$\tilde{q}_n = -\lambda_{tr} \left(\frac{\partial T}{\partial z}\right)$

For the temporal integration of the governing equations, an implicit numerical scheme has been used. The terms of the governing equations are discretized by the following approximation: $\frac{\partial \phi}{\partial t} \cong (\phi - \phi^0)/\Delta t$, where ϕ represents a dependently generic variable (ϕ = T, h, λ, ρ,…). The mean values over a control volume have been estimated as the arithmetic mean between the inlet and outlet sections, that is: $\tilde{\phi} \cong \bar{\phi} = (\phi_i + \phi_{i+1})/2$. The average physical properties are evaluated at their corresponding average variables. Applying the aforementioned numerical approximations, an equation can be obtained for each node:

$$aT_i = bT_{i+1} + cT_{i-1} + d \quad (8)$$

Where the coefficients are:

$a = \frac{\lambda_w A}{\Delta z} + \frac{\lambda_e A}{\Delta z} + \left(\alpha_s P_s + \frac{\lambda_{tr} P_{tr}}{\Delta z}\right)\Delta z + \frac{A\Delta z}{\Delta t}\rho c_p$;

$b = \frac{\lambda_e A}{\Delta z}$

$c = \frac{\lambda_w A}{\Delta z}$;

$d = \left(\alpha_s P_s T_{tubo} + \frac{\lambda_{tr} P_{tr} T_{tr}}{\Delta z}\right)\Delta z + \frac{A\Delta z}{\Delta t}\rho c_p T_i^0$

Where α_s is the heat transfer coefficient by convection in the working fluid and P is the perimeter.

Empirical coefficients for the governing equations

The mathematical model requires information about the friction factor f and the convective heat transfer coefficient to solve the momentum equation and the energy equation, respectively. This information is generally obtained from empirical correlations.

The correlation for the calculation of the friction factor for flow in a single phase is evaluated using the expression of friction factor in laminar regime and for turbulent flow the one proposed by Ito (1959), shown in equations 9 and 10, respectively.

$$Re < 2300 \quad f = \frac{64}{Re} \quad (9)$$

$$Re \geq 2300 \quad f = 1.216Re^{-0.25} \quad (10)$$

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Where Re is the Reynolds number.

The correlation for the calculation of the heat transfer coefficient for the flow in a single phase is calculated using the equations of Nusselt and Dittus-Boelter (1930), for laminar flow calculated from equation (11) and for turbulent flow with the equation (12).

$$Re < 2300 \quad Nu = 3.66 \tag{11}$$

$$Re \geq 2300 \quad Nu = 0.23\, Re^{0.8} \tag{12}$$

Where:

$coef = 0.4$, si la $T_{pared} > T_{fluido}$
$coef = 0.3$, si la $T_{pared} < T_{fluido}$ band Nu is the Nusselt number.

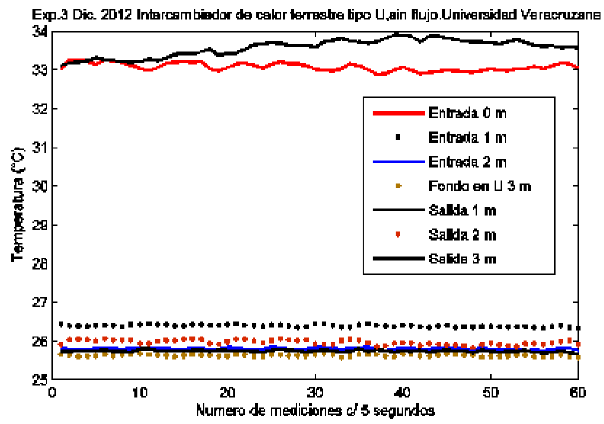
Solution of governing equations

The solution of the governing equations has been coupled to a method called step by step. The domain of the terrestrial heat exchanger is divided into control volumes. The solution process is carried in the aforementioned way in the direction of flow. With known values at the entrance of the section and having defined the boundary conditions, the values of said variables at the output of the control volume are obtained from the discretization of the governing equations (continuity equation, motion equation and equation power).

Once the solution is obtained at the output of the control volume, it becomes the input values for the next control volume. This procedure is followed until the end of the domain. The thermo-physical properties of the working fluid (density, heat capacity, etc.) for the solution of governing equations were calculated by Steam 1997 (IAPWS-IF97) presented by Holmgrem (2007). For each control volume a set of algebraic equations is obtained from the governing equations (3, 5, 6 and 8).

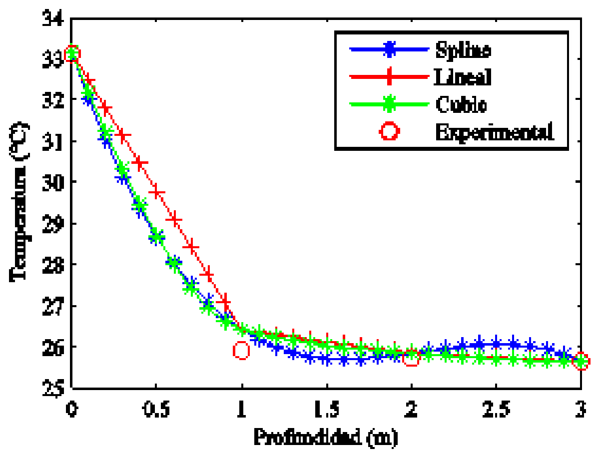
Results and Discussion

According to the geothermal profile previously shown (see Figure 1), the temperature does not present significant variations at depths greater than 0.8 m. Temperatures during the day and night converge, for this reason, in the present work the short period of temperature measurements at depths of 1, 2 and 3 m is considered sufficient.



Graphic 1 Distribution of soil temperatures measured up to a depth of 3 meters.

Graph 1 shows the distribution of temperatures on earth, 60 measurements every 5 seconds were recorded. For each measured depth, 7 temperature sensors were installed located 2 to 0 meters from the ICT inlet, 2 to 1 meter deep, 2 to 2 meters and 1 to 3 meters, attached to the inlet and outlet geothermal exchanger tube. The thermal conductivity calculated for the sandy and humid soil of the Universidad Veracruzana Campus Coatzacoalcos, was 3,065 W / m ° C. In order to have values for each node required by the resolution algorithm, an analysis was carried out with different types of interpolations. 3 types of interpolation were analyzed: linear, spline and cubic. The results of the interpolations are shown in graph 2. From which the spline was chosen (it is a piecewise interpolation of defined degree and with certain derivability properties), since it has the sinusoidal shape commented on by the authors (Florides & Kalogirou 2007). The interpolation was estimated for both tubes. The effects of tube curvature are neglected in the present work.



Graphic 2 Earth temperature interpolations and experimental temperatures from the surface to depths of 3m

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Table 1 shows the design parameters selected for the proposed scenario of the terrestrial heat exchanger.

Parameter	
Evaluated materials	PVC, 316 steel, aluminum, copper,
External diameter	33.4 mm
Inside diameter	30.4 mm
Tube length	6 m
Temptation	40 °C
Pentrada	atmosférica
Volumetric flow	0.68 m^3/hr

Table 1 Base Design Parameters of Terrestrial Heat Exchanger

The data in table 2 show the power values delivered to the subsoil, these are measured for the 1-inch nominal diameter (DN) pipe. For the pipe with a nominal diameter (DN) 1 inch the PVC an average power of 1.3927 kW was calculated, for the 316L Steel an average power of 1.4029 kW, considering aluminum as a construction material the average power calculated was 1.5117 kW and finally assuming copper, a power of 1.6011 kW was estimated. That is, the simulated heat flux increase, assuming the DN 1 inch PVC as the base case, with the simulated materials with DN 1 inch, is as follows: using PVC as the tube material, an increase of 0.73% was calculated considering steel. 316L. An increase of 8.54% is calculated considering aluminum and an increase of 14.96% is estimated using copper as a construction material.

PVC			
Time	5s	120 s	300 s
T outlet (° C)	38.2432	38.2482	38.2417
Heat flow (kW)	1.3932	1.3893	1.3944
Steel 216L			
Time	5s	120 s	300 s
T outlet (° C)	38.2311	38.2361	38.2297
Heat flow (kW)	1.4027	1.3988	1.4039
Aluminum			
Time	5s	120 s	300 s
T outlet (° C)	38.0919	38.0968	38.0904
Heat flow (kW)	1.5122	1.5083	1.5134
Copper			
Time	5s	120 s	300 s
T outlet (° C)	37.9782	37.9830	37.9767
Heat flow (kW)	1.6016	1.5978	1.6028

Table 2 Simulated heat flow using experimental ground temperature as a boundary condition.

The results show that the highest thermal gain with the same ground conditions and mass flow were those obtained with copper.

Conclusions

A computational tool was developed assuming a one-dimensional analysis of the governing equations (continuity, momentum and energy) in a terrestrial heat exchanger, in order to help in the design and optimization of this class of equipment. Experimental ground temperature data were used as boundary conditions for the algorithm solution. PVC, steel, aluminum and copper were considered as construction materials, their performance in the U-tube vertical ICT was estimated.

Within which the following is concluded:

- Of the four materials simulated with the same conditions as the base case, with a nominal diameter (DN) of 1 inch, copper is the one that obtains the highest heat exchange with the subsoil. Obtaining an average power of 1.6011 kW.
- The increase in heat flow in DN 1 inch pipe vs PVC pipe (base case) of DN 1 inch is given as follows: steel = 0.73%, Aluminum = 8.54% and copper = 14.96%.

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