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A Simplified Model for Helical Heat Exchanger for Long-Term Energy Storage in Soil

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Summary

A simplified numerical code for thermal analysis of a helical heat exchanger for use in long-term thermal energy storage in soil was developed. The model was verified for a particular case for which an analytical solution was available from the literature and was validated with experimental data obtained from field experiments. The differences between predicted and measured data were in the range of ±1°C, which is considered satisfactory for engineering design purposes. The model was prepared for use with an personal computer and thus provides a convenient and reliable design tool for such a system. The computer code may be easily modified for the study of the influence of incorporating phase-change material elements in the storage well.

Introduction

The development of alternate energy sources to fossil fuels has acquired high priority in view of the environmental problems associated with the extraction and consumption of these fuels and the instability of oil prices. The utilization of waste heat and renewable energy sources as viable alternatives depends on the development of cost-effective thermal energy storage systems for the short term and, particularly, for the long term. Among the various techniques proposed for long-term thermal energy storage, a method based on soils as the heat storage medium is considered one of the most promising possibilities. Most of the R&D effort on this method has been devoted to cold and moderate zones and usually involves vertical multiple well storage. The different heat exchanger models previously proposed include the U-shaped exchanger of Reuss et al. [1], in which 25-mm diameter polypropylene pipes are inserted into the soil in vertical bores about 0.15 m in diameter and 10 m in depth. For warm and arid regions, a helical heat exchanger for seasonal heat storage has been proposed by Nir et al. [2]. Earlier work on this idea included the investigation of the general concept and the development of theoretical model for the design of such a system for application in unsaturated soils, which constitute the norm in arid zones [2]. Recently, an experimental field system, based on this model, was built and operated at the Institutes for Applied Research of Ben-Gurion University of the Negev. The system was used to obtain experimental data for validation testing of the model and for investigating certain engineering and operational issues [3]. It was shown that for clay soils with a water content above 20% and operational temperatures in the range of 20-80°C the effect

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of soil drying on the heat transfer process may be neglected. Based on this conclusion the objective of the present work was to develop a simplified and reliable code for use in a personal computer for the simulation of thermal energy storage in a helical heat exchanger inserted into the soil. The paper describes the simplified theoretical model, the verification testing, and the results of a comparison with experimental data.

Theoretical Model

A schematic description of the system and the geometric parameters of the helical heat exchanger are given in Fig. 1.

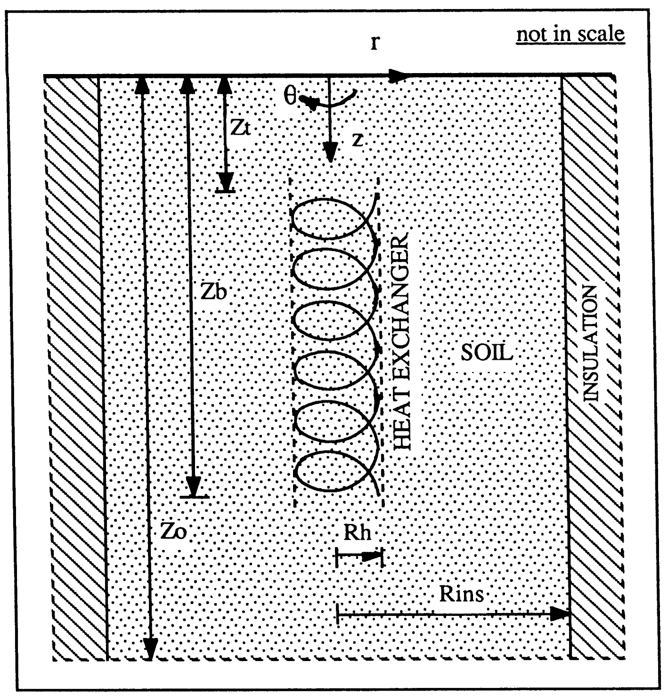

Fig. 1: Schematic description of the system

where r, θ, z are the coordinates of the cylindrical system, Zt and Zb are the distance of the upper and lower sides of the heat exchanger from the surface of the ground, respectively, Zo is the total height of the system, Rh is the radius of the heat exchanger and Rins is the radius of the system.

The theoretical model is based on the following simplified assumptions:

Soil:

1) The effects of moisture and solute transfer are neglected and the heat transfer in the soil is assumed to be carried out only by conduction.

2) The soil is assumed to be isotropic, with average constant thermophysical properties. However, the properties may not be necessarily the same in different defined spaces in the system, for example, properties in the inner region may differ from those in the outer region corresponding to the helical heat exchanger.

3) The energy storage system is one of a field comprising many identical systems.

4) The temperature gradient in the tangential direction is neglected, and the model is considered to be two dimensional (axi-symmetrical) and seminfinite in the vertical direction.

Heat exchanger:

5) The flow regime of the working fluid (i.e. water) in the pipe is laminar.

6) The thermal resistance of the pipe is neglected, and the Biot number which characterizes the heat transfer between the working fluid and the soil is large (Bi>>1), so that the heat transfer is controlled by the heat conduction in the soil. The temperature of the working fluid is constant at each cross section.

7) The helical heat exchanger is modelled as a series of horizontal rings with constant spacing between them.

Based on the above assumptions the governing heat transfer equation for the soil is:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \cdot \frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} + \frac{\dot{q}}{k} = \frac{1}{\alpha} \cdot \frac{\partial T}{\partial t} \qquad (1)$$

where T is the soil temperature, q is an internal heat source, k is the thermal conductivity, α is the thermal diffusivity, and t is the time.

The initial condition is:

$$T(z,r,0) = T_0(z) \qquad (1a)$$

where $T_0(z)$ - is the initial soil temperature.

The boundary conditions are:

$$\frac{\partial T}{\partial r}(z,0,t) = 0 \qquad (1b)$$

$$\frac{\partial T}{\partial r}(z,Rins,t) = 0 \qquad (1c)$$

$$\frac{\partial T}{\partial z}(\infty,r,t) = 0 \qquad (1d)$$

The heat flux on the surface of the system is defined in terms of heat convection from the surface to the air, solar radiation, and thermal radiation from the surface to the sky:

$$k\frac{\partial T}{\partial z}(0,r,t) = q_{surface}(T(o,r,t), T_{air}(t), q_{solar}(t), P_{aw}(t)) \quad (1e)$$

where $q_{surface}$ is the heat flux through the upper surface of the system, T_{air} is the air temperature, q_{solar} is the flux of the solar radiation absorbed by the surface, and P_{aw} is the partial pressure of the water vapor in the air in mm hg.

A further assumption is made that the ground surface is effectively "gray":

$$q_{surface}(r,t) = q_{solar}(t) + h_c(T_{air}(t) - T(0,r,t)) + \varepsilon\sigma(T_{ef}(t)^4 - T(0,r,t)^4) \quad (2)$$

where h_c is the convection heat transfer coefficient, ε is the emissivity of the soil and σ is the Stefan-Boltzman constant.

The convective heat transfer coefficient can be determined from the correlation [4]:

$$h_c = 6.2 + 1.4 \cdot u_{wind} \quad [w/m^2 \cdot K] \quad (3)$$

where u_{wind} is the wind velocity in [m/s].

The effective sky temperature T_{ef} is defined as the temperature of a black body which would radiate to the ground the same flux as actually reaches it from the sky [5]:

$$T_{ef}(t) = \left(0.55 + 0.065\sqrt{P_{aw}(t)}\right)^{0.25} \cdot T_{air}(t) \quad [K] \quad (4)$$

It is interesting to note that eq. 4 suggests that in an arid zone at or near sea level (as is the case at the experimental region) the effective nocturnal sky temperature can be expected to be approximately 15°C below the ambient air temperature.

At temperatures in the range of the ambient temperature it is possible to approximate the fourth power temperature difference of the thermal radiation by a linear difference:

$$q_{surface}(r,t) = q_{solar}(t) + h_c(T_{air}(t) - T(0,r,t)) + h_r(T_{ef}(t) - T(0,r,t)) \quad (5)$$

where the radiative heat transfer coefficient h_r is defined as:

$$h_r = 4\varepsilon\sigma T_{av}^3 \quad (6)$$

and the average temperature, as:

$$T_{av} = (T_{ef} + T(0,r,t))/2 \quad [K] \quad (7)$$

For the working fluid stream:

$$a \cdot \int_0^{2\pi} k \cdot \left.\frac{\partial T}{\partial r^*}\right|_{r^*=a} \cdot d\theta^* = \pi \cdot a^2 \cdot \rho_f \cdot Cp_f \cdot \left(\frac{Cv_f}{Cp_f} \cdot \frac{\partial T_f}{\partial t} + u_{av} \cdot \frac{\partial T_f}{\partial z^*}\right) \quad (8)$$

where θ*, r*, z* are coordinates related to the pipe (Fig. 2), T_f is the temperature of the fluid, a is the radius of the pipe, u_{av} is the average velocity of the fluid, Cp_f is the constant-pressure specific heat of the fluid and Cp_v is the constant-volume specific heat of the fluid.

Considering the fact that for liquid heat transfer the term $\partial T_f/\partial t$ is about three orders of magnitude smaller than $u_{av} \cdot \partial T_f/\partial z^*$ [6], equation (8) of the heat balance becomes:

$$a \cdot \int_0^{2\pi} k \cdot \left.\frac{\partial T}{\partial r^*}\right|_{r^*=a} \cdot d\theta^* = \dot{m} \cdot Cp_f \cdot \frac{\partial T_f}{\partial z^*} \quad (9)$$

where $\dot{m}$ is mass flow rate of the fluid.

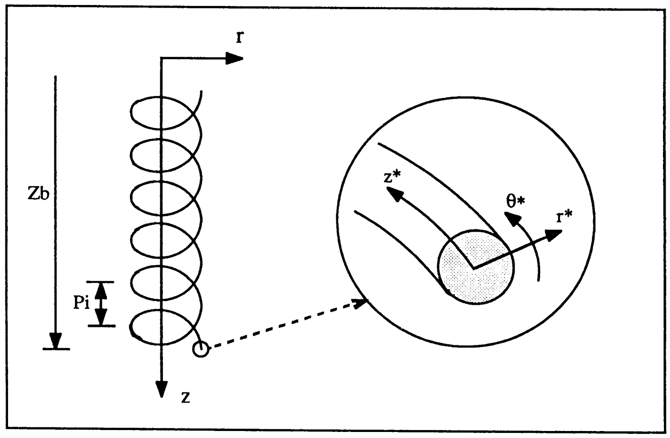

Fig. 2: The system cylindrical coordinates and the pipe cylindrical coordinates.

The boundary conditions are:

$$T_f(z^*,t) = T(z,r,t) \qquad r,z \in \Omega \tag{9a}$$

where Ω is the surface of the heat exchanger pipe.

$$T_f(0,t) = T_{inlet}(t) \tag{9b}$$

where T_{inlet} is the inlet temperature of the working fluid to the heat exchanger, and determined by the heat source of the system.

The initial condition is:

$$T_f(z^*,0) = T_0(z^*) \tag{9c}$$

where the transformation between the pipe coordinates system and the cylindric ground coordinates system is defined as:

$$z = Zb - \frac{z^*}{2\pi} \cdot Pi \tag{10}$$

where Pi is the pitch of the helical heat exchanger (Fig 2).

The theoretical model can be solved by the finite differences method. The system is separated into small ring elements, and in each element a small lump system behavior is assumed. The temperature field is calculated by the following implicit equation:

$$T_{ij}^{p+1} = T_{ij}^{p} + \frac{\Delta t}{C_{ij}}\left[\sum_{n} \frac{T_n^p - T_{ij}^p}{R_{ijn}} + \dot{q}_{ij}^p\right] \tag{11}$$

where the index i,j refers to the discretization with respect to space, and the index p, to time, Δt

is a time interval, C_{ij} is the heat capacity of an element ij defined by (12) and R_{ijn} represents the thermal resistance between element ij and its neighbor n, calculated from eq. 13 for pure conduction and from eq. 14 for convection.

$$C_{ij} = \rho_{ij} \cdot Cp_{ij} \cdot V_{ij} \tag{12}$$

$$R_{ijn} = \begin{cases} \dfrac{\Delta r}{(r_{ij} + 1/2 \cdot \Delta r) \cdot \Delta z \cdot k} & n = i, j+1 \\[2ex] \dfrac{\Delta r}{(r_{ij} - 1/2 \cdot \Delta r) \cdot \Delta z \cdot k} & n = i, j-1 \\[2ex] \dfrac{r_{ij}}{\Delta r \cdot \Delta z \cdot k} & n = i-1, j \\[2ex] \dfrac{r_{ij}}{\Delta r \cdot \Delta z \cdot k} & n = i+1, j \end{cases} \tag{13}$$

$$R_{ijn} = \frac{1}{A_{ij} \cdot h_c} \tag{14}$$

The criterion for the solution stability is:

$$\Delta t \leq \left[\frac{C_{ij}}{\sum_n 1/R_{ijn}} \right]_{min} \tag{15}$$

The numerical scheme (eq. 11) was selected so as to enable the calculation of the temperature field associated with phase change (liquid-solid) processes in the system, which are an important aspect that we intend to study in the next step.

In order to solve eq. 9 numerically the cross section of the pipe is considered as to have a square cross section having the same perimeter as the pipe to simulate an identical heat transfer surface area. The temperature profile in z* direction, in the working fluid, is determined by:

$$T^{p+1}_{in,e+1} = T^{p}_{in,e} + \frac{1}{\dot{m} \cdot Cp_f} \cdot \sum_n \frac{T^p_n - T^p_{b,e}}{R_{ijn}} \tag{16}$$

where $T_{in,e}$ and $T_{out,e}$ are the inlet and outlet temperatures of the fluid in element e, respectively, and T_b is the bulk temperature of the fluid in the element:

$$T^p_{b,e} = \frac{1}{2}\left(T^p_{out,e} + T^p_{in,e}\right) \tag{17}$$

<u>Verification testing of the theoretical model</u>

The first step in testing the numerical computer code was to check the overall energy balance of the system. The overall energy transfer from the working fluid (water) was calculated from:

$$E = \int_0^t \dot{m} \cdot Cp_f \left[T_{inlet}(t) - T_{outlet}(t)\right] \cdot dt \tag{18}$$

where E is the stored energy.

The total energy absorbed by the soil is:

$$E(t) = \iiint_V \rho \cdot Cp \cdot [T(r,z,t) - T(r,z,0)] \cdot dv \tag{19}$$

The energy balance was calculated from the results of a simulation for a six-month energy storage period. The testing was performed with the following numerical values: the initial temperature of the soil was uniform and equal to 20°C, the inlet fluid temperature was 70°C and the mass flow rate was 40 kg/h. In order to assess the effect of neglecting the time dependent term $\partial T_f/\partial t$ in eq. (8), energy conservation was defined as the ratio of the total energy (eq. 19) to the total energy storage (eq. 18). It was observed that 88.0% of the energy conservation was obtained after the first hour, 97.8% after 6 h, 98.9% after 24 h and 99.94% after six months. From these results it can be seen that neglecting the time-dependent $\partial T_f/\partial t$ term in eq. 9 may effect the results only during the very short time after the step change in the fluid input temperature.

The computer code results have also been examined against an exact solution for a simplified problem. In this case a one-dimensional heat conduction problem in an infinite cylinder with constant thermophysical properties was solved. The domain of solution was subjected to a step wall temperature at time t>0. The governing equation for this case is therefore:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \cdot \frac{\partial T}{\partial r} = \frac{1}{\alpha} \cdot \frac{\partial T}{\partial t} \tag{20}$$

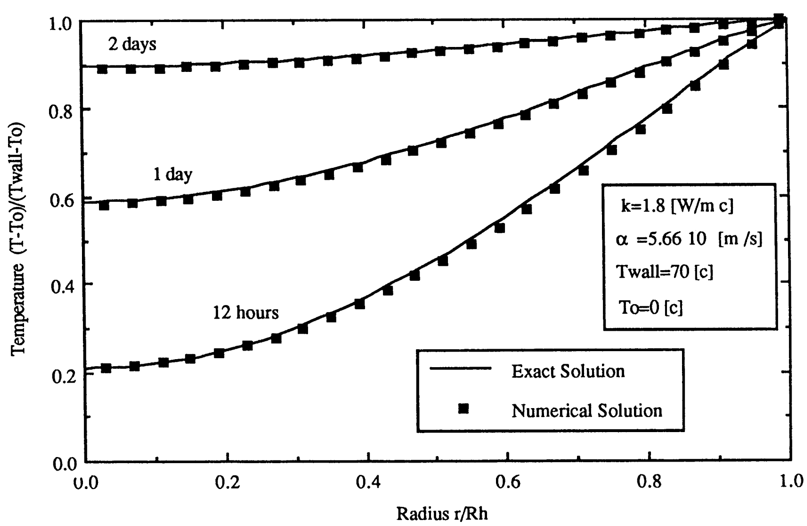

Fig. 3: Comparison between numerical results and the exact solution [7] for transient one-dimentional heat conduction in an infinite cylinder.

The initial condition is:

$$T(r,0) = T_0 \tag{20a}$$

and the boundary conditions are:

$$\frac{\partial T}{\partial r}(0,t) = 0 \tag{20b}$$

and

$$T(a,t) = T_{wall} \tag{20c}$$

The exact solution to this problem as given in [7] is:

$$T(r,t) = T_0 - \frac{2 \cdot T_0}{a} \cdot \sum_{i=1}^{\infty} \exp\left(-\alpha \cdot \gamma_i^2 \cdot t\right) \cdot \frac{J_0(r \cdot \gamma_i)}{\gamma_i \cdot J_1(r \cdot a)} \tag{21}$$

where γ_i is obtained from the solution of the Bessel equation $J_0(a \cdot \gamma_i) = 0$.

The test was carried out with the following numerical parameters: a very low thermal conductivity outside the well, thermal diffusivity of the soil in the well of $\alpha = 5.66 \cdot 10^{-7}$ m^2/s, a high water flow rate of $m = 10^5$ kg/h and a very small pitch were taken in order to simulate a constant wall temperature. The results of the numerical model for this case are superimposed in Fig. 3. Very good agreement between the numerical and exact solution is clearly seen.

Experimental validation testing

The simplified theoretical model was tested vs. experimental data obtained from the experimental field system operating at the Institutes for Applied Research. In this system the helical heat exchanger made out of 0.03 m diameter polypropylene pipe with a 0.1 m pitch was 1 m in diameter and 6 m long. The heat exchanger was inserted into a 10-m deep well. The experimental system is described in detail in ref. 8. The thermophysical properties of the soil were estimated in previous experimental work: thermal conductivity k = 1.3 W/m·K and specific heat Cp $= 2.838 \cdot 10^6$ $J/m^3 \cdot K$. Solar radiation and dry and wet bulb air temperatures were provided by the meteorological station located in the neighborhood. The experiment was run for 30 days (2/2/90-4/3/90). Fig. 4 shows a comparison of the outlet temperature of the water from the heat exchanger as a function of time, as predicted by the simplified theoretical model, with that measured experimentally. Fig. 5 compares the temperature profiles in the soil at radius of 0.3 m from the center of the helical heat exchanger after 10, 20 and 30 days of the experiment. In both Figures it can be seen that the difference between measured and theoretical results is of the order of ±1°C, which is satisfactory agreement for engineering design purposes.

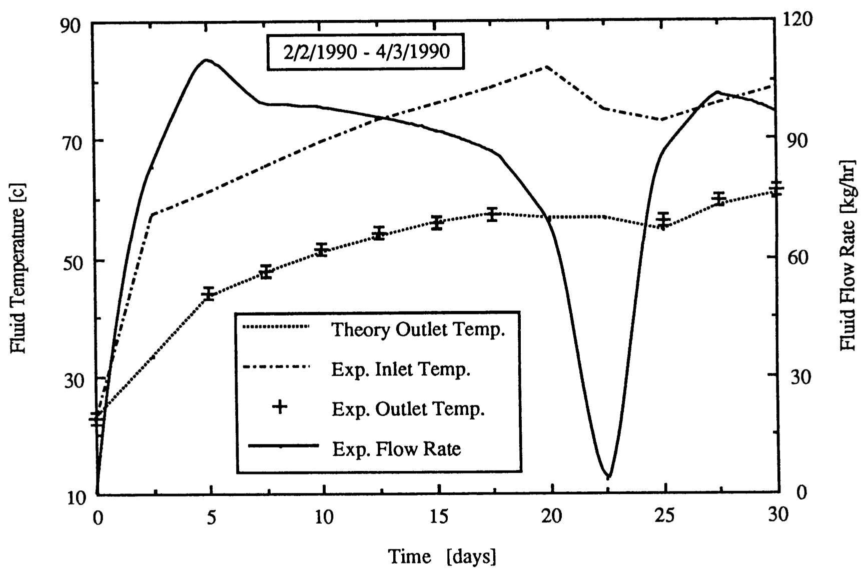

Fig. 4: Comparison between outlet fluid temperatures as predicted by the theoretical model and experimental results (measured from 2/2/90 to 4/3/90).

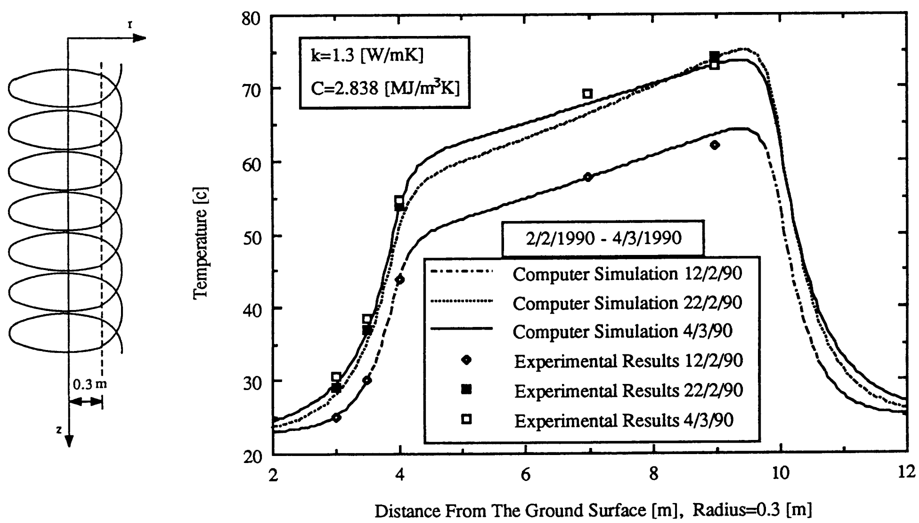

Fig. 5: Vertical temperature profiles in the soil, 0.3 m from the center of the well, as predicted by the theoretical model vs. experimental results.

Conclusions and recommendations for future work

A simplified numerical code for thermal analysis of a helical heat exchanger for thermal energy storage was developed. Theoretical verification testing and experimental validation testing showed that the model gives reliable results for parametric studies and engineering design purposes, for the type of soil with the properties tested in the present work, i.e., clay with a moisture content above 20%.

The numerical scheme was developed in a manner that can also predict the effect of incorporating a phase-change thermal energy storage element in the soil to improve the capacity of the thermal storage in the system. This effect is under study at present time and results will be published in the near future.

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