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Thermal analysis of a helical heat exchanger for ground thermal energy storage in arid zones

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Abstract—A mathematical model for thermal analysis of a helical heat exchanger for long-term thermal energy storage in soil for use in arid zones was developed. The helical heat exchanger was modeled as a series of horizontal rings with a constant pitch distance between them. The model was solved by a finite difference method, using a microcomputer, and validated with experimental data obtained from field experiments. Based on the model, theoretical results of the following parametric studies are presented: thermal properties of the soil, cycle period, and height and pitch distance of the helical heat exchanger.

INTRODUCTION

Increasing world-wide awareness of the serious environmental problems caused by the consumption of fossil fuels has highlighted the need for a reduction in the use of this energy source, both by improvement of the efficiency of existing systems and by expanding the utilization of nonpolluting renewable energy sources, such as solar energy. An energy storage unit is an essential component of any system that uses a time-variable energy source, such as solar energy, industrial waste heat or geothermal water. Such a unit can also play an important role in the load management of constant power sources, for example, a generating plant that has to meet time-variable demands. These factors have motivated intensive research and development of various energy storage methods that will widen the practical applications of thermal energy storage technologies in the future.

Thermal energy storage in the ground is considered to be an attractive method for long-term energy storage. The system is based on soil, which is relatively the most widespread and the least expensive energy storage medium. A variety of concepts have been considered, and a number of full-scale systems have been set up to demonstrate the technology. Most of the work has been performed for space-heating applications in cold climate zones for the seasonal storage of solar energy, i.e. from summer to winter, and the systems are usually coupled with a heat pump for space heating applications [1]. In contrast, the technology developed by us is designed for application in warm climate zones.

The general concept and a theoretical model for seasonal thermal energy storage for use in arid and semi-arid zones has been previously described [2]. The method is based on burying a large diameter (1–1.3 m) vertical helical heat exchanger in unsaturated soil, which is the commonly available ground in such areas.

Compared with the commonly used vertical tube heat exchanger, the helically shaped heat exchanger offers unique flexibility in the design of the system:

(1) The required heat exchanger surface per unit well may be easily obtained by adjusting the pitch distance of the helix.

(2) The large diameter of the well facilitates the exploitation of some of the space for incorporating special devices to improve the thermal performance of the system, such as irrigation equipment with moisture sensors to maintain the moisture content and thermal properties of the soil, or incorporating phase-change energy storage elements to increase the thermal capacity of the system.

An experimental field system based on this concept has been designed, built and operated at the Institutes for Applied Research, Ben-Gurion University of the Negev, Beer-Sheva, Israel. The experimental data used for validation of the model and the experience obtained from experimental work has been used for addressing engineering problems and for evaluating the economic feasibility of the implementation of this system in arid zones [3–7]. A preliminary cost estimation for the construction and installation of a modular heat storage system with a heat exchanger having a diameter of 1.3 m, a pitch of 0.1 m and a length of 18 m shows that the cost of the polybutylene pipe accounts for more than 70% of the total cost of the system [3]. Further theoretical and experimental studies of this concept are required for better design of the heat exchanger, not only for a seasonal operation mode but also for a shorter cycle period.

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NOMENCLATURE

a	radius of the heat exchanger pipe [m]
C	volumetric specific heat [J m^{-3} °C^{-1}]
C_p	specific heat [J kg^{-1} °C^{-1}]
E	energy [J]
k	thermal conductivity [W m^{-1} °C^{-1}]
m	mass flow rate of the working fluid [kg s^{-1}]
M	mass of the working fluid per unit length of the heat exchanger pipe [kg m^{-1}]
P_{aw}	water vapor pressure in air [Pa]
Pi	pitch of the helical heat exchanger [m]
q	heat flux [W m^{-2}]
R	radius [m]
t	time [s]
T	temperature [°C]
u	velocity [m s^{-1}]
V	volume [m^3]
Z	depth [m]
r, θ, z	cylindrical coordinates of the storage system
r^*, θ^*, z^*	local coordinates of heat exchanger pipe.

Greek symbols

α	thermal diffusivity [m^2 s^{-1}]
ρ	density [kg m^{-3}].

Subscripts

0	initial
abs	absorbed
av	average
b	bottom of the helical heat exchanger (Fig. 1)
f	working fluid
h	helical heat exchanger
inlet	fluid entrance to the helical heat exchanger
ins	insulation between storage unit and storage field (Fig. 1)
outlet	fluid exit from heat exchanger
s	soil
t	top of the helical heat exchanger (Fig. 1)
trans	transfer.

In previous theoretical studies [2–7], the theoretical model was solved by a computer code (designated PT) that was developed at the Lawrence Berkeley Laboratory as a general code for simulation of hot-water geothermal reservoirs. In order to enable the solution of the theoretical model by this computer code, the helical heat exchanger had to be modeled as an annular cylindrical conduit of equivalent volume and surface.

The agreement between the theoretical model and the practical system is reduced as the pitch distance increases. Since the pitch distance of the helical exchanger was found to play an important role in cost of the system, it was imperative to develop an alternative computer code for which the helical heat exchanger could be modeled with a finite pitch distance; such a model would be more suitable for representing the thermal performance of the system. Furthermore, previous studies [3–6] indicated that for the soil type and working conditions of our system, the heat transfer in the soil could be described as a purely conductive process with effective thermal conductivity. This conclusion further motivated the development of an simplified code that would provide a convenient and reliable design tool for such a system.

Thus, the objectives of the present work were: (1) to develop (by means of a microcomputer) a reliable computer code for thermal analysis of a helical heat exchanger with a finite pitch distance and (2) to perform parametric studies on the effect of various parameters on the performance of the heat exchanger.

A general description of the simplified theoretical model has been previously presented [8]. In this paper the assumptions of the theoretical model, are discussed in greater detail, and a special analysis of the ground surface boundary conditions and experimental validation of the model are presented. Theoretical results on parametric studies related to thermal properties of the soil, cycle period, height and pitch distance of the helical heat exchanger are also given.

THEORETICAL MODEL AND VERIFICATION TESTING

A schematic description of the system and the geometric parameters of the helical heat exchanger are given in Fig. 1. The theoretical model is based on the following simplified assumptions with regard to the soil:

(1) In general, a model of heat transfer in soil has to take into account the fact that soil is a multicomponent medium consisting of three phases (solid, liquid and gas). It also has to take into account the effects of the coupling process of heat and mass transfer in soils. In the case of partially saturated soil, the situation becomes more complicated because determination of soil parameters such as water diffusion requires extensive theoretical and experimental efforts. In our particular case the operating temperature is limited to <80°C and the water diffusivity in unsaturated soil is relatively low (in the range of 10^{-8} to 10^{-10} m^2 s^{-1}) compared to the thermal diffusivity (order of 10^{-6} m^2 s^{-1}). Furthermore, it has been

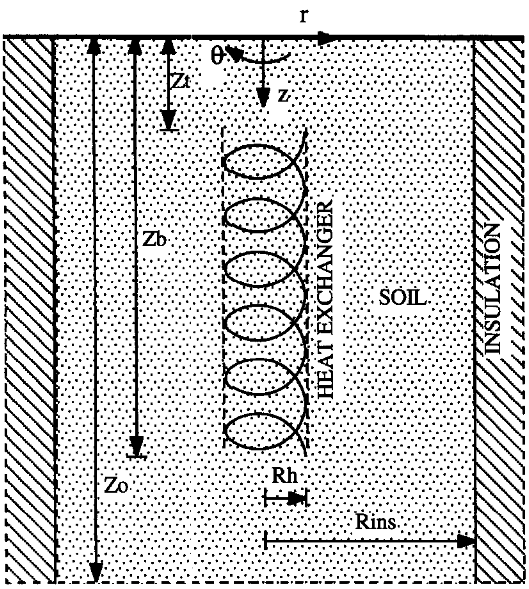


Fig. 1. Schematic description of the energy storage system.

previously shown [4–6] that for clay soils with a water content above 20% and operating temperatures in the range of 20–80°C, the effect of soil drying on the heat transfer process may be neglected. Therefore, in this work the effects of moisture and solute transfer are neglected, and the heat transfer in the soil is assumed to be carried out solely by conduction, using effective thermal conductivity.

(2) The soil is assumed to be isotropic, with average constant thermophysical properties.

(3) The model is taken as a single unit in a multi-unit field system; thus, the model assumes perfect thermal insulation at a fixed distance from the center of the center coordinate axes of the heat exchanger.

(4) The temperature gradient in the tangential direction is neglected, and the model is considered to be transient, two-dimensional (2D) and axisymmetric.

Based on the assumptions mentioned above, the equation governing heat transfer in the soil is given by:

$$\frac{\partial^2 T_s}{\partial r^2}+\frac{1}{r}\frac{\partial T_s}{\partial r}+\frac{\partial^2 T_s}{\partial z^2}=\frac{1}{\alpha_s}\frac{\partial T_s}{\partial t}. \quad (1)$$

The initial condition of the system is:

$$T_s(z,r,0) = T_{s0}(z). \quad (2)$$

Three of the four boundary conditions of the system are:

$$\frac{\partial T_s}{\partial r}(z,0,t) = 0 \quad (3)$$

$$\frac{\partial T_s}{\partial z}(z_0,r,t) = 0 \quad (4)$$

$$\frac{\partial T_s}{\partial r}(z,R_{ins},t) = 0 \quad (5)$$

where the boundary condition (5) considers the storage system as a single unit in a multiunit field and $2R_{ins}$ is the distance between the center axis of two neighboring energy storage units.

With regard to the fourth boundary condition, i.e. that relating to the upper surface of the system, three different approaches may be taken into consideration, as follows:

(a) The first approach is based on a scenario in which the heat exchanger is placed 4–6 m below the ground surface, where the climatic conditions have a negligible effect on the thermal performance of the energy storage system, i.e. it is assumed that there is thermal insulation at a certain depth located between the ground surface and the helical heat exchanger.

(b) The second approach is based on the assumption that the presence of the energy storage system has no effect on the thermal performance of the surroundings and therefore the temperature profile near the surface (at a depth of about 0.5 m) will be the same as that in undisturbed soil in an open field (without the energy storage system). This method demands that preliminary work be performed to acquire data on the average temperature profile in the upper layer of the soil at the location of the selected site over a whole year [6].

(c) The third approach is based on the determination of the heat flux on the ground surface; this parameter combines heat convection from the surface to the air, solar radiation, and thermal radiation from the surface to the sky. This boundary condition may thus be defined by the ambient conditions:

$$k_s\frac{\partial T_s}{\partial z}(0,r,t) = q_{\text{surface}}^{(T(0,r,t),T_{air}(t),q_{solar}(t),P_{aw}(t))} \quad (6)$$

where $q_{surface}$ is the heat flux on the ground surface of the system, T_{air} is the ambient temperature, q_{solar} is the solar radiation and P_{aw} is the water vapor pressure in air. Details of the determination of $q_{surface}$ are given in refs. [8, 9].

A schematic description of the helical heat exchanger and the coordinate systems are given in Fig. 2. The following basic assumptions are made with regard to the helical heat exchanger:

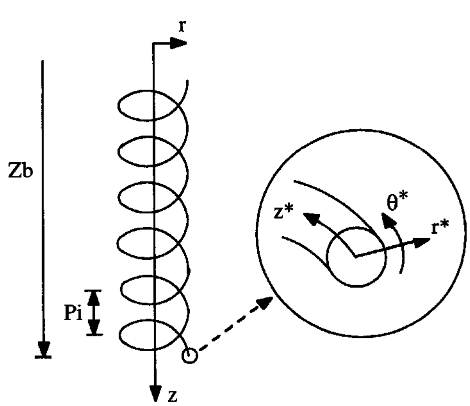


Fig. 2. Schematic description of the helical heat exchanger and its coordinate system.

(1) The thermal resistance to heat transfer by convection in the fluid and the thermal resistance to heat transfer by conduction in the pipe can both be neglected; thus, heat transfer is assumed to be controlled solely by the heat conduction in the soil. Furthermore, at each cross-sectional area of the pipe the temperature is assumed to be uniform (under the conditions of the system the Biot number is greater than 20).

(2) The helical heat exchanger is modeled (Fig. 3b) as a series of horizontal rings with a constant pitch distance between them. The vertical pipe sections between the horizontal rings, from the ground surface to the helical heat exchanger, and from the helical heat exchanger back to the ground surface, are considered to be mathematically insulated and thus represent only the flow direction.

Under these assumptions, the equation for heat transfer in the fluid is given by:

$$a\int_0^{2\pi} k_s \left.\frac{\partial T}{\partial r^*}\right|_{r^*=a} d\theta^* = M_f C_{pf}\left(\frac{\partial T_f}{\partial t} + u_{av}\frac{\partial T_f}{\partial z^*}\right) \quad (7)$$

where θ^*, r^*, z^* are the local coordinates of the pipe. Since the term $\partial T_f/\partial t$ is about three orders of magnitude smaller than $u_{av} \cdot \partial T_f/\partial z^*$ [10], equation (7) may be expressed as:

$$a\int_0^{2\pi} k_s \left.\frac{\partial T}{\partial r^*}\right|_{r^*=a} d\theta^* = \dot{m} C_{pf}\frac{\partial T_f}{\partial z^*}. \quad (8)$$

Assuming that the initial temperatures in the heat exchange pipe and in the soil domain are identical, the boundary and initial conditions of the working fluid are:

$$T_f(z^*, t) = T_s(z^*, a, t) \quad (9)$$

$$T_f(0, t) = T_{f,inlet}(t). \quad (10)$$

The transformation between the local pipe coordinate system (Fig. 2) and the master cylindrical coordinate system (Fig. 1) is given by:

$$z = Z_b - \frac{z^*}{2\pi R_h} Pi. \quad (11)$$

The 2D axisymmetric transient heat conduction problem in the soil can then be solved numerically by a finite difference method as presented in refs. [8, 9].

In order to check the numerical scheme, the overall energy transfer from the working fluid (water) was calculated by:

$$E_{trans}(t) = \int_0^t \dot{m} C_{pf}[T_{f,inlet}(t) - T_{f,outlet}(t)]\, dt. \quad (12)$$

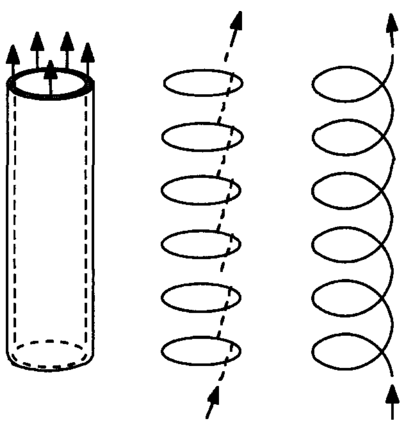
The value thus obtained was compared with the energy absorbed by the soil:

$$E_{abs}(t) = \iint_v\int \rho_s C_{ps}[T_s(r, z, t) - T_s(r, z, 0)]\, dV. \quad (13)$$

Equation (13) represents a perfectly insulated system, in which the only heat source is that of the heat exchanger.

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 working parameters: a uniform initial temperature of the soil of 20°C, fluid inlet temperature of 70°C, and mass flow rate of 40 kg h^{-1}. The energy conservation factor, calculated from the ratio of the energy transferred to the system [equation (12)] to the energy absorbed by the system [equation (13)] is 0.88 after 1 h, 0.978 after 6 h, 0.989 after 24 h and 0.999 after six months. From these results it can be seen that neglecting the time-dependent term in equation (8) has an effect only in the very short term after the step change in the fluid inlet temperature.

The next step was to compare the results of the numerical model with simplified cases of the 1D and 2D transient analytical solution from Jager and Carslaw [10]. The verification test presented elsewhere [8, 9] showed very good agreement between the numerical and exact solutions.



A B C

Fig. 3. Modeling of the helical heat exchanger tube. (a) annular cylindrical conduit; (b) horizontal rings and (c) actual helix shape.

EXPERIMENTAL VALIDATION TESTING OF THE MODEL

The theoretical model was tested against experimental data obtained from the experimental field system operating at The Institutes for Applied Research. In this system the helical heat exchanger was made of polybutylene pipe with the following geometric parameters: 0.03 m diameter pipe, 0.1 m pitch, 1 m diameter ($R_h = 0.5$ m) and 6 m in length. The heat exchanger was inserted into a 10 m deep well. The experimental system is described in detail in [3, 9]. The thermophysical properties of the soil at the experimental site are: average thermal conductivity $k_s = 1.3$

W m^{-1} °C^{-1} and average volumetric specific heat $C_{ps} = 2.84$ MJ m^{-3} °C^{-1} [9]. Solar radiation and dry and wet bubble air temperatures were supplied by the meteorological station located near the site of the field experiment. The experiment was run for 30 days (2 February 1990–4 March 1990). Figure 4(a) presents 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. Figure 4(b) compares temperature profiles vs depth in the soil at a radius of 0.3 m from the center of the helical heat exchanger for 10, 20 and 30 days of the experiment. In both sets of graphs it can be seen that the difference between measured and theoretical results is of the order of ± 1°C, which is a satisfactory agreement for engineering design purposes.

PARAMETRIC STUDIES

In general, the simulations for parametric studies were carried out for the same geometric parameters

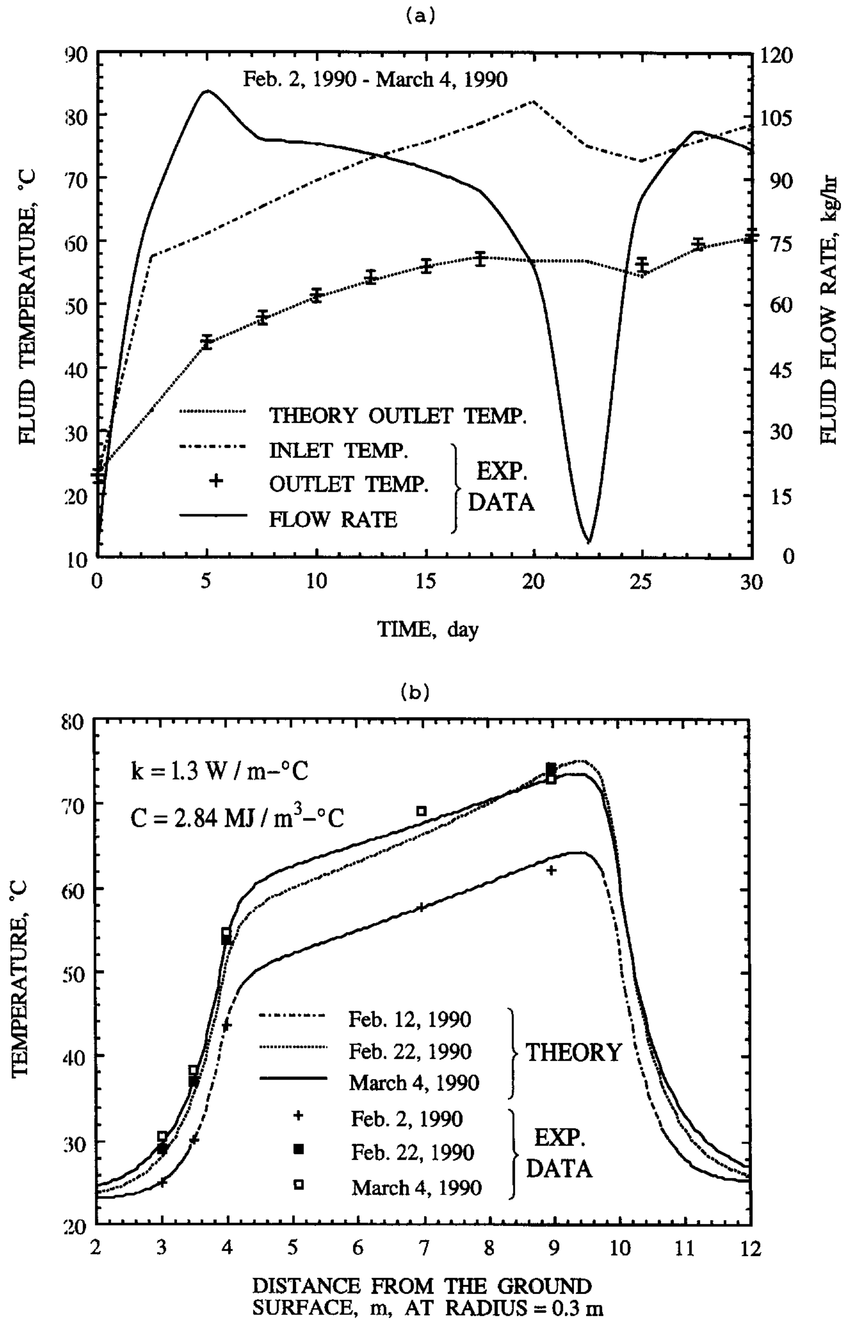


Fig. 4. (a) Outlet fluid temperatures and (b) vertical temperature profiles in the soil, 0.3 m from the center of the well, as predicted by the theoretical model vs experimental results [8].

and soil thermophysical properties as in the field experiment (described in Section 4). The operational conditions were as follows: mass flow rate of the working fluid during both the charging and discharging periods 40 kg h^{-1}, and inlet fluid temperatures of 70 and 20°C for the charging and discharging periods, respectively. A constant temperature boundary condition of 20°C at a depth of 0.5 m was assumed. In the cases in which numerical values different from those stated above were used, this is stated in the text.

Upper boundary condition

In order to examine the significance of the differences among the three different ways of dealing with the boundary condition that corresponds to upper part of the ground energy storage medium, simulations were performed for each one in turn. The simulations were carried out for a cycle period comprising eight months of energy charging followed by four months of discharging, lasting from the beginning of December to the end of March.

For case (a) the insulated surface was placed at a depth of 0.5 m. For case (b) the temperature profile for undisturbed soil was taken from measurement data recorded in the field experiment area during 1988–1989 [11]. For case (c) the solar radiation, wind velocity, and ambient temperature were taken as daily average values. This case is considered to be the one that best represents the real system. However, in addition to the drawback of requiring meteorological data from the site area, case (c) needed extensive computation efforts.

Comparisons among the simulation results for the three cases are presented in Figs. 5(a) and (b). It can be seen that the thermal behavior of the insulated system (case a) differs considerably from that of free boundary case (c) that is considered to be the most accurate solution; therefore, case (a) is not suitable for further use in our analysis. The biggest difference between the results for the case of fixed temperature profile near the surface boundary (case b) and case (c) is 2.5°C [Fig. 5(a)]. The differences in the total energy storage are within the range of 15% [Fig. 5(b)]. It can be expected that the difference will become larger with the years or for shorter energy storage cycles.

Although the second type of the upper boundary condition (case b) can be used only as a first approximation, it is very useful for further parametric studies in which a yearly average ground temperature is known. For arid zones, such as those in which the experimental system was placed, an annual average temperature of 20°C at a depth of 0.5 m can be used.

Thermal properties of the soil

The most important soil properties that have to be considered in the design of a ground thermal energy storage system are thermal conductivity and volumetric specific heat. These properties depend mainly on the type, bulk density, water content and temperature of the soil. We have restricted our study to the silty clay soil type that is present in our field experimental system and have examined a range values from dry soil conditions $k_s = 0.8$ W m^{-1} °C^{-1} and $\alpha_s = 3.2\ 10^{-7}$ m^2 s^{-1} to fully saturated conditions $k_s = 1.8$ W m^{-1} °C^{-1} and $\alpha_s = 5.66\ 10^{-7}$ m^2 s^{-1}.

The field temperature distributions in these two cases at different times are given in Figs. 6 and 7. After a charging period of five months the energy stored in saturated soil was about 50% higher than that in unsaturated soil. However, a comparison of the temperature field in the soils after five months of charging [Figs. 6(a) and (b)] shows that the energy stored in the unsaturated soil was located nearer to the heat exchanger surface and was therefore more readily available for the discharging process. Despite the fact that the thermal conductivity of the saturated soil was higher than that of the unsaturated soil, the outlet water temperature and heat discharging rate were higher in the unsaturated soil than in the saturated soil during the first two weeks of the discharging period [Fig. 8(a)]. In the first discharging month, the outlet water temperature in the unsaturated soil dropped to about 28°C vs about 10°C in the saturated soil [Fig. 8(a)].

The results for a short cycle of one month of charging and one month of discharging are depicted in Figs. 9(a) and (b). The total energy stored in the system is higher for saturated soil than for the unsaturated soil, 4.8 GJ vs 3.4 GJ, respectively. However, during the first 10 days of the discharging process, the water outlet temperature in the saturated soil was higher, and thus the heat flow rate was lower; thereafter, the outlet water temperature was about the same in the two systems. This interesting finding indicates that for this particular case the thermal performance of the system in unsaturated soil was better than in that in saturated soil.

Effect of charging period

The outlet water temperature as a function of time for different charging periods, i.e. 2, 4, 6 and 8 months, and a four-month discharging period are given in Fig. 10. During the first month of the charging period there was an almost linear increase in the water outlet temperature vs time (0.5°C day^{-1}), and thereafter it decreased by about 10-fold. This phenomenon was also evident in the discharging process: the outlet water temperature decreased in the first month by about 0.4°C day^{-1}, and thereafter the change in slope depended on the length of the charging period, Fig. 10. This finding could be explained by the fact that the transit time constant of the system related to the thermal mass in the interior well was about 30 days.

Height of the heat exchanger

To study the influence of the height of the heat exchanger on the thermal performance of the system two simulations were carried out for heat exchanger

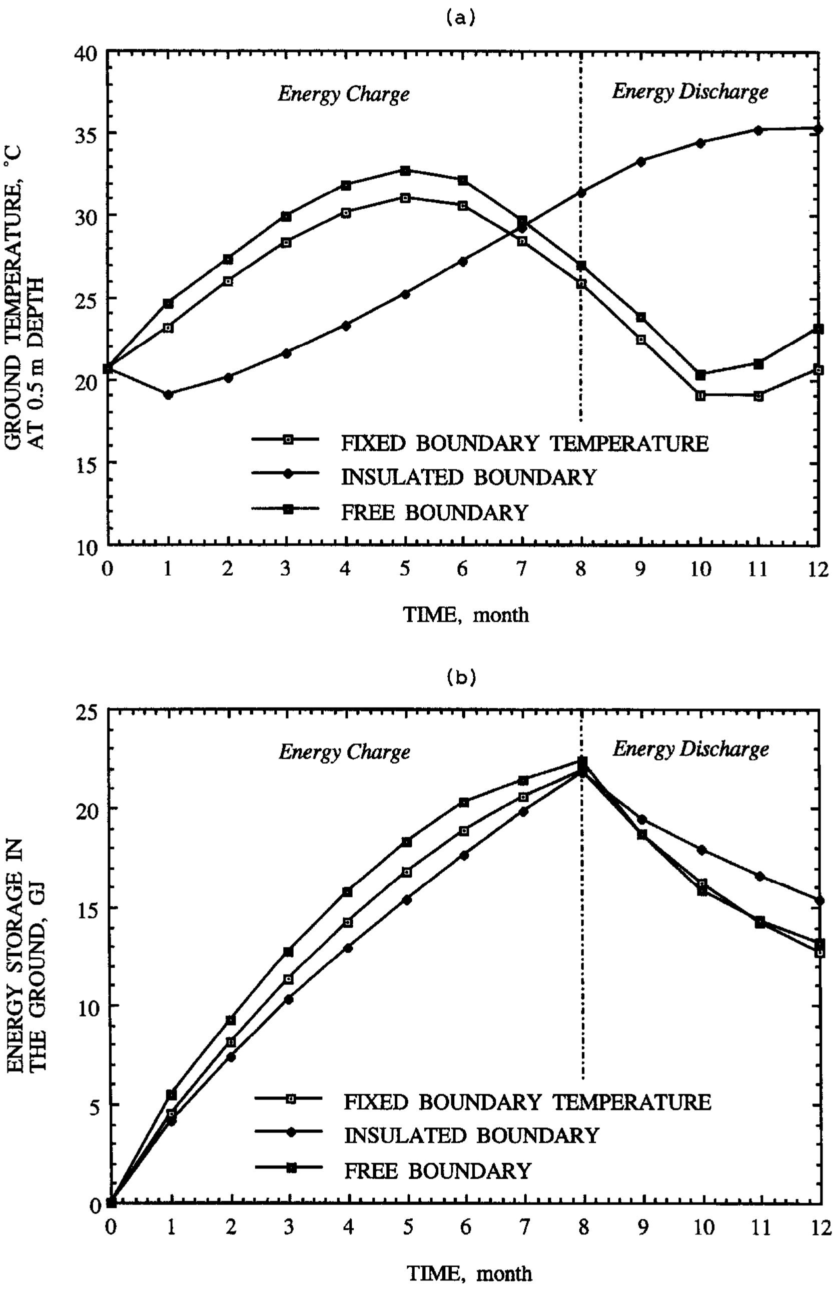


Fig. 5. (a) Temperature distribution at a depth of 0.5 m and (b) energy storage in the ground for the three different upper boundary conditions.

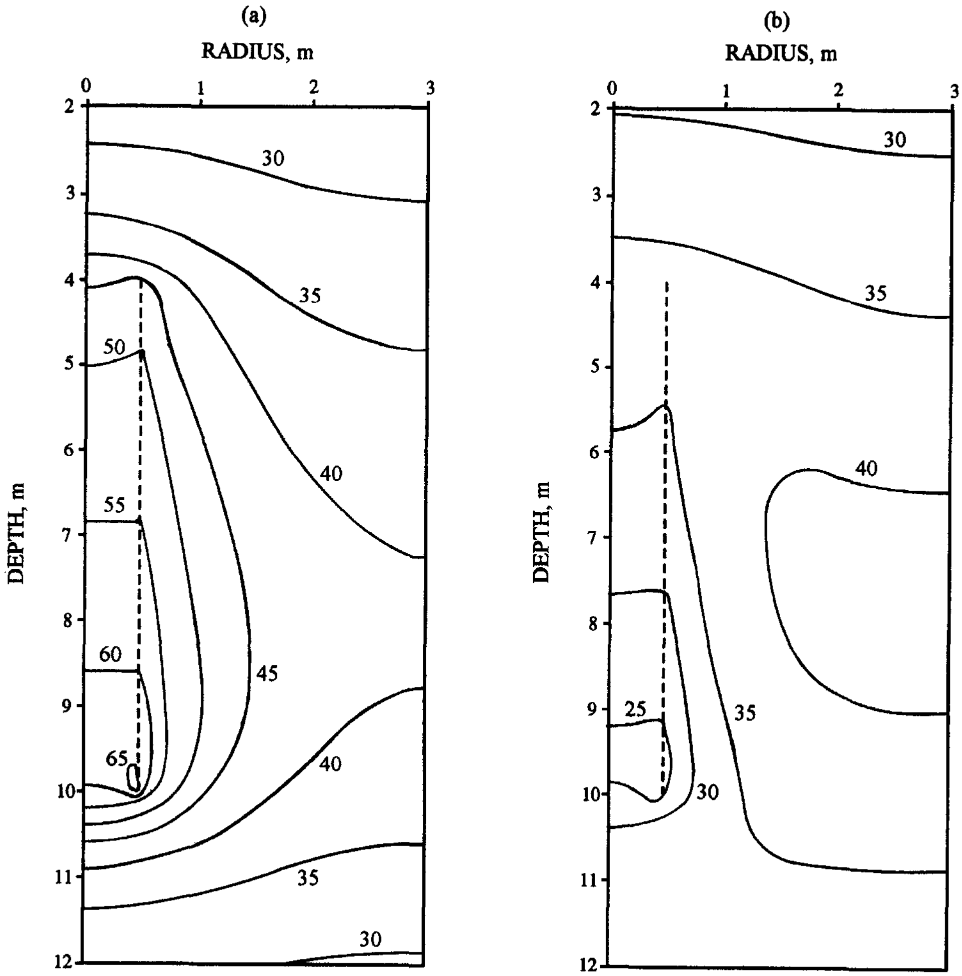


Fig. 6. Temperature distribution in saturated soil $\alpha = 5.66 \times 10^{-7}$ m^2 s^{-1} after (a) 150 days of charging and (b) 150 days of charging and 15 days of discharging.

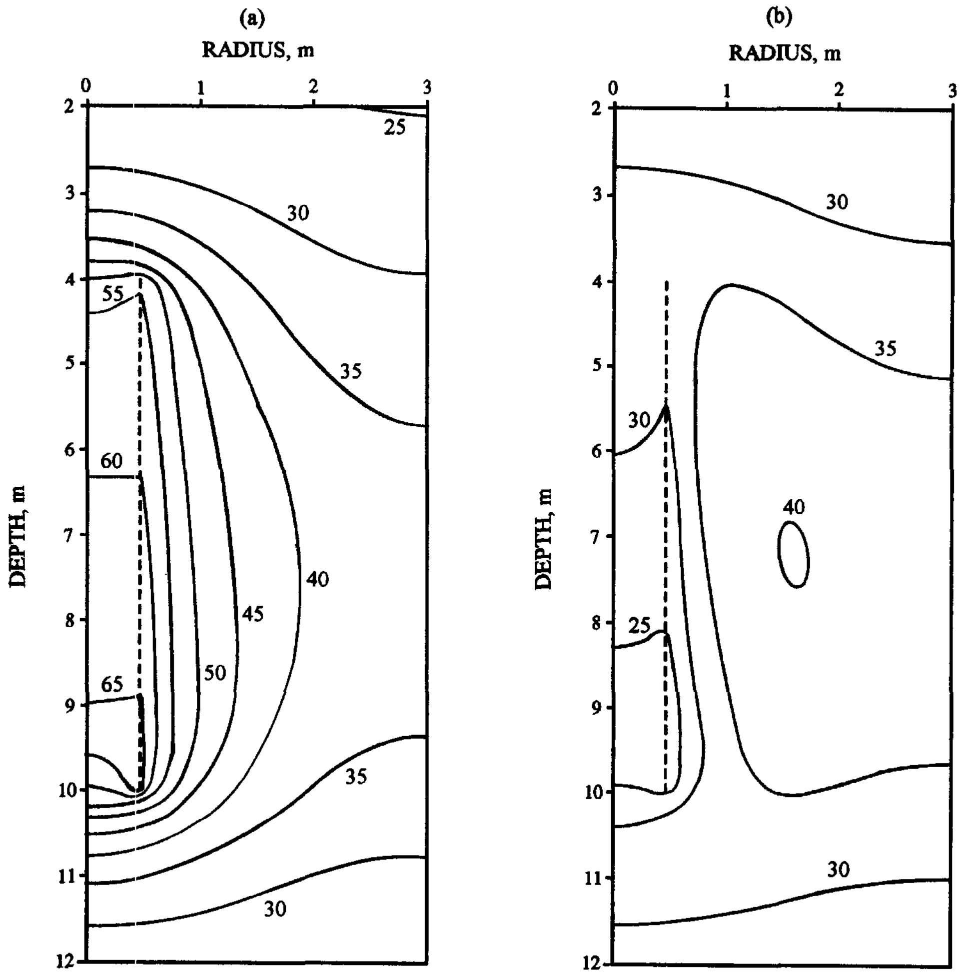


Fig. 7. Temperature distribution in unsaturated soil $\alpha = 3.2 \times 10^{-7}\ m^2\ s^{-1}$ after (a) 150 days of charging and (b) 150 days of charging and 15 days of discharging.

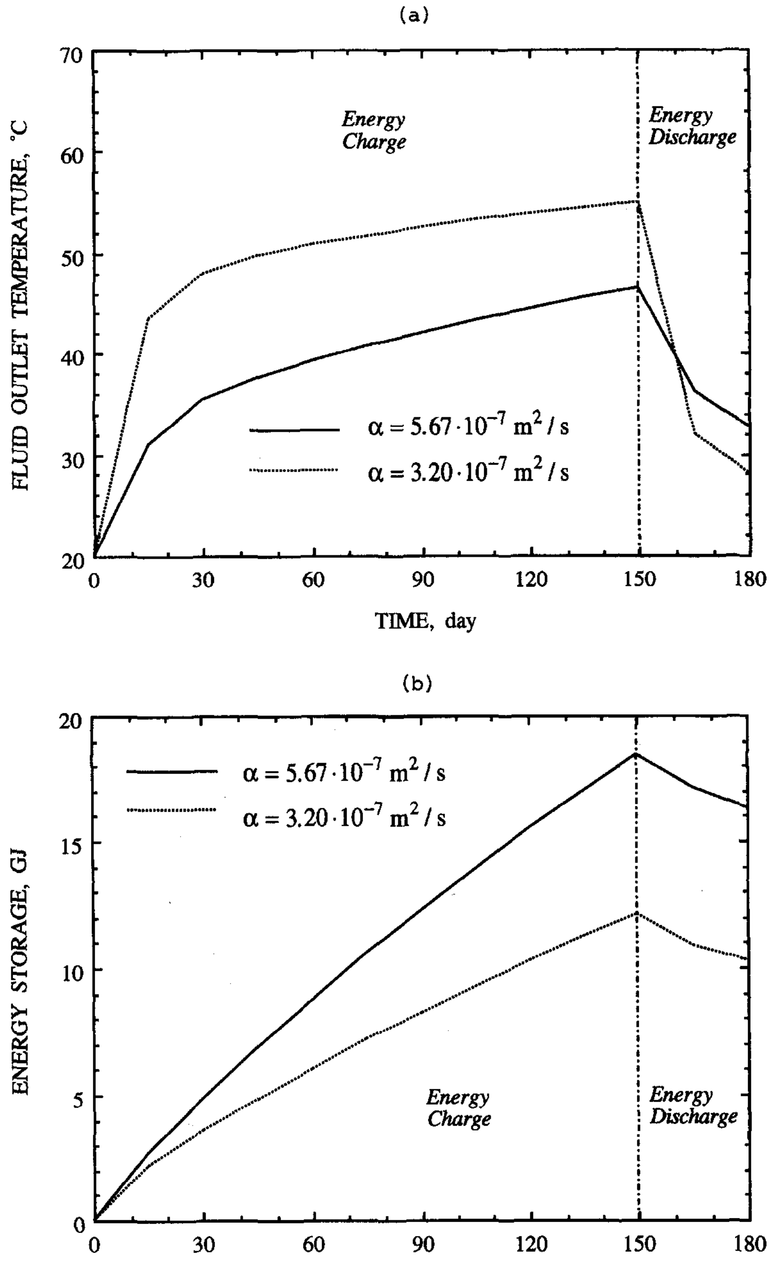


Fig. 8. (a) Outlet fluid temperature vs time and (b) energy storage vs time for saturated soil $\alpha = 5.66 \times 10^{-7}$ $m^2\ s^{-1}$ compared with unsaturated soil $\alpha = 3.2 \times 10^{-7}$ $m^2\ s^{-1}$ (five months of charging, one month of discharging).

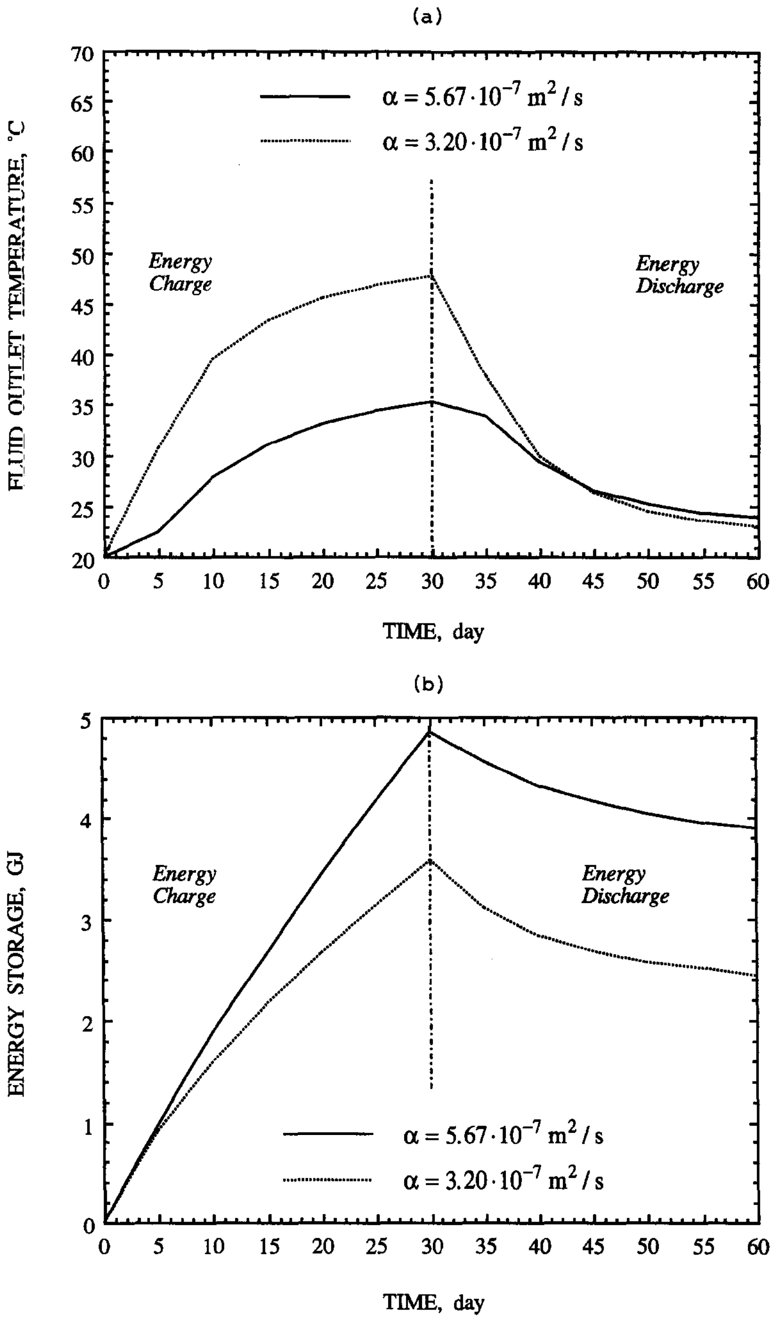


Fig. 9. (a) Outlet fluid temperature vs time and (b) energy storage vs time for saturated soil $\alpha = 5.66 \times 10^{-7}$ $m^2\ s^{-1}$ compared with unsaturated soil $\alpha = 3.2 \times 10^{-7}$ $m^2\ s^{-1}$ (one month of charging, one month of discharging).

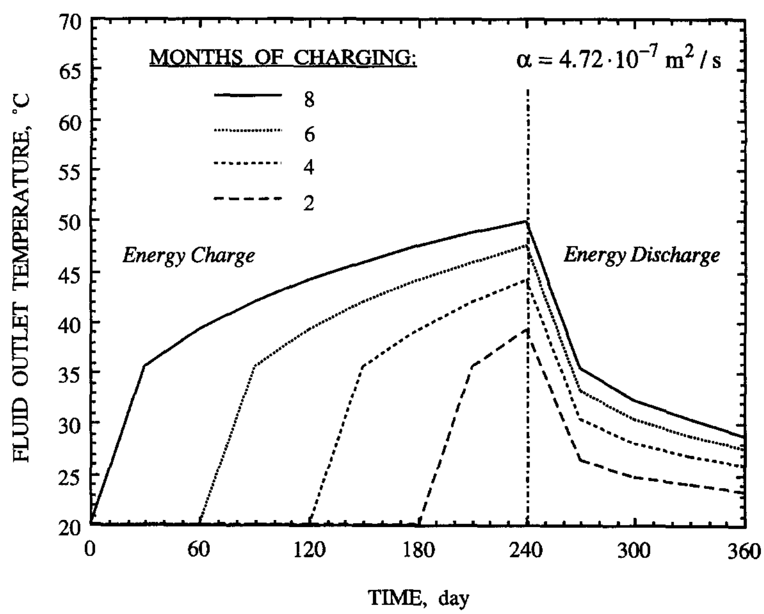


Fig. 10. Outlet fluid temperature vs time for different charging periods of 2, 4, 6 and 8 months and a 2-month discharging period.

heights of 6 m and 7.5 m. The results of a cycle of three-months of charging and one month of discharging are presented in Figs. 11(a) and (b). After five energy cycle periods, the system almost reached a quasi-state, in which the fluid outlet temperatures were in the range of 45 ± 10°C and 45 ± 5°C for heat exchanger heights of 6 m and 7.5 m, respectively. Thus, it is evident that systems working in a narrow band of outlet temperatures have a better thermal performance. A system having a longer heat exchanger reaches a quasi state after a higher number of cycles, since the effective thermal mass of the system is higher.

Pitch distance

The pitch distance is one of the geometric parameters that is directly related to the length of the piping of which the heat exchanger is made. Since the piping constitutes a significant part of the cost of the system [3], it is of great importance to study the effect of this parameter. The theoretical study was carried out for three months of charging followed by one month of discharging. From Fig. 12 it can be seen that the thermal efficiency increased with the number of cycles and became almost constant after four cycles. The effect of the pitch distance was reduced as the number of cycles increased. Under the conditions of the test, a pitch distance in the a range of 0.1–0.3 m did not have a significant effect on the thermal performance of the system. This finding indicates that for a longer cycle period an even larger pitch distance could be used without considerable reduction in the thermal efficiency on the system.

SUMMARY AND CONCLUSIONS

A new simplified mathematical model for the thermal analysis of a helical heat exchanger for ground thermal energy storage was developed. The model was solved by a straight-forward numerical scheme, with the aid of a microcomputer. The mathematical model and the numerical scheme were verified and validated with experimental data from field experiments. From these tests the model was found to be satisfactory for engineering calculations and hence for the thermal analysis and design of such system. The following conclusions can be drawn from the parametric studies:

(1) The commonly accepted idea that saturated soil is always a better thermal energy storage medium than unsaturated soil (since the former has higher thermal heat capacity and thermal conductivity) is not always true. In our case of a helical heat exchanger inserted in silty clay-type soil, energy stored in the soil is located near the heat exchanger surface and it is therefore more readily available for use during the initial discharge period. It was found that for a relatively short cycle period, one month of charging followed by one month of discharging, unsaturated soil is the better energy storage medium.

(2) The effect of pitch distance is reduced as the cycle period and cycle number are increased. It was found that increasing the pitch distance up to 0.3 m had no significant effect on the thermal performance of the system for a seasonal thermal energy storage cycle. This indicates that the pipe length could be reduced threefold compared with the existing field experiment system [3].

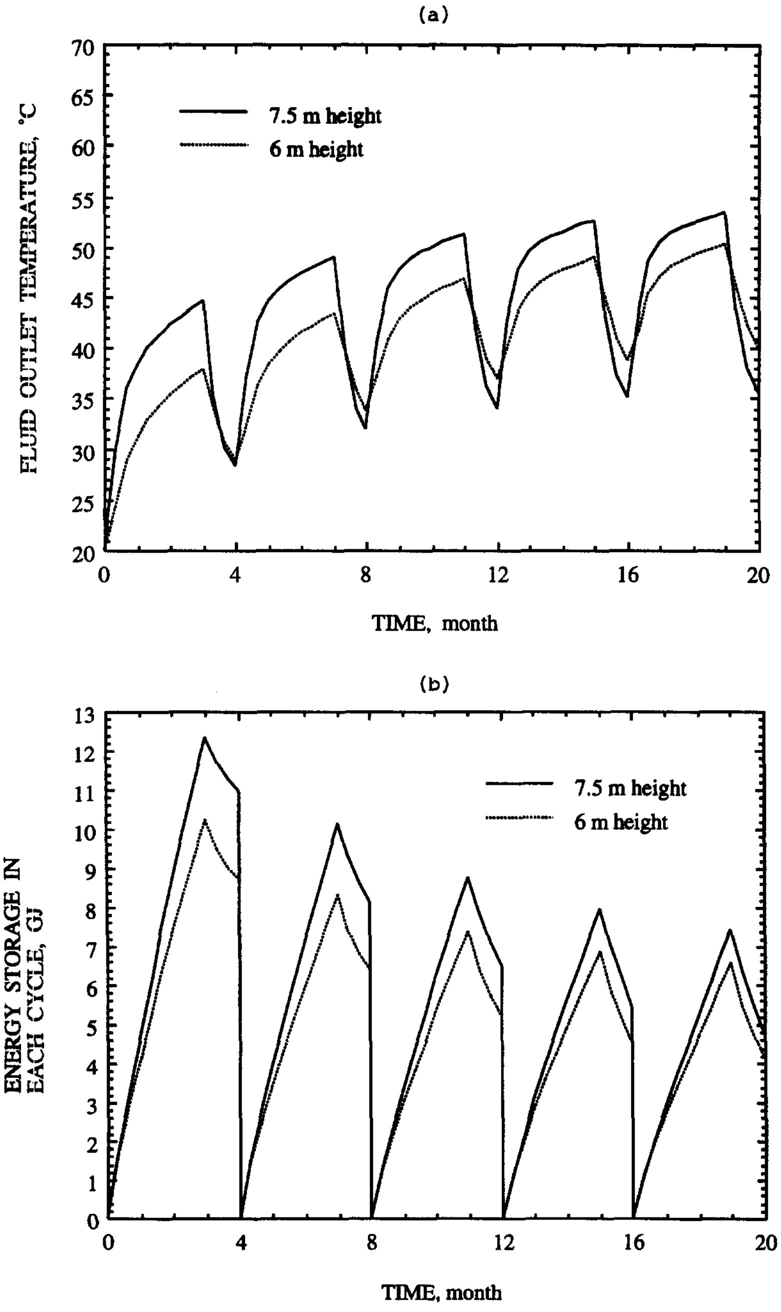


Fig. 11. Effect of the height of the heat exchanger on (a) fluid outlet temperature and (b) energy stored in the soil.

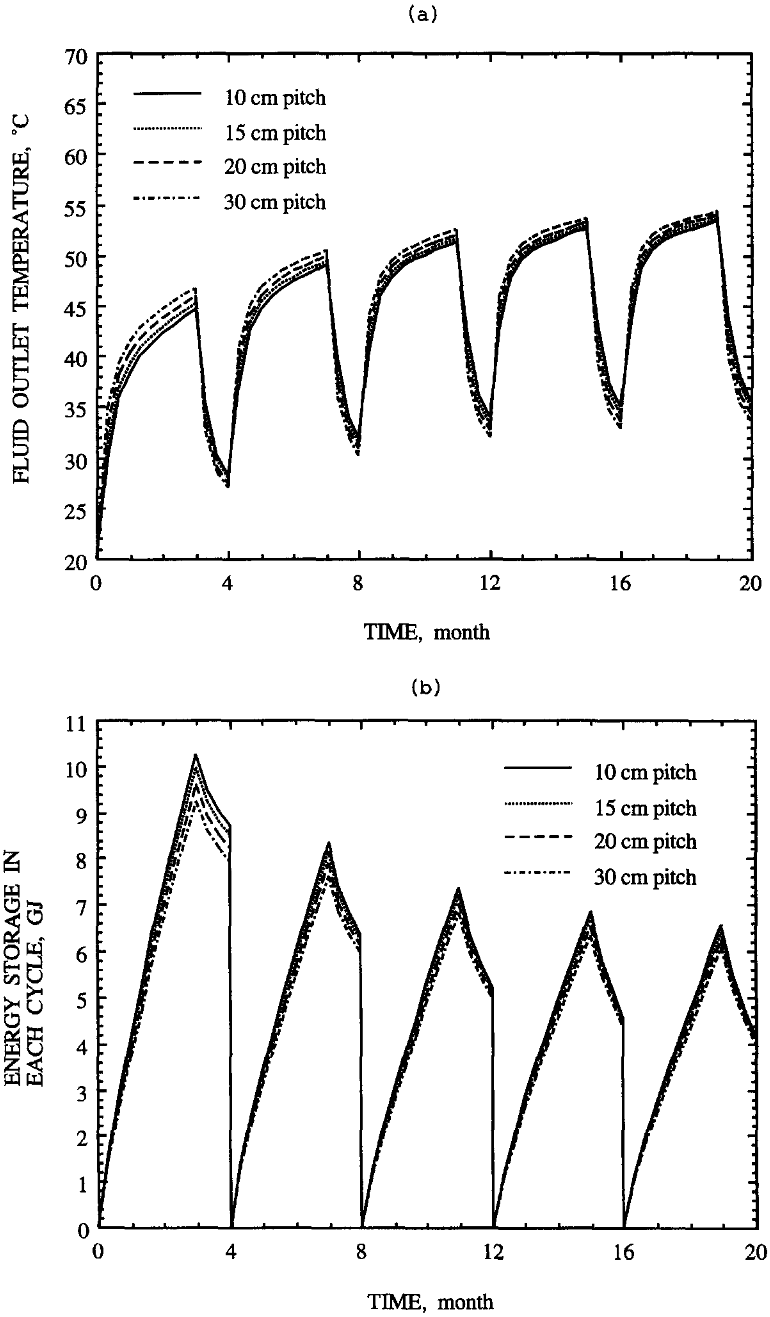


Fig. 12. Effect of the pitch distance on (a) fluid outlet temperature and (b) energy stored in the soil.

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REFERENCES

1. Papers presented at International Conferences on Thermal Energy Storage: (a) *International Conference of STES and CAES*, Seattle (1981); (b) *International Conference on Subsurface Heat Storage in Theory and Practice*, Stockholm (1983); (c) *ENERSTOCK '85—Third International Conference on Energy Storage*, Toronto (1985); (d) *JIGASTOCK 88—Fourth International Conference on Energy Storage*, Versailles (1988); (e) *THERMASTOCK '91—Fifth International Conference Thermal Energy Storage*, Scheveningen (1991) and (f) *CALORSTOCK '94—Sixth International Conference on Thermal Energy Storage*, Otaniemi, Espoo, Finland (1994).
2. A. Nir, C. Doughty and C. F. Tsang, Seasonal heat storage in unsaturated soils: example of design study, *Proceedings 21st Intersoc. Energy Conversion Engineering Conference*, San Diego, Vol. 2, pp. 669–675 (1986).
3. B. Bar-On, E. Korin, A. Nir and C. F. Tsang, Seasonal

storage of thermal energy in unsaturated soils: experimental procedures and technological development, *Proceedings of Thermastock '91, Fifth International Congress on Thermal Energy Storage*, pp. 4.3.1–4.3.6. Scheveningen, Holland, 13–16 May (1991).

4. J. Bensabat, C. Doughty, E. Korin, A. Nir and C. F. Tsang, Validation experiments of seasonal thermal energy storage models in unsaturated soils, *Proceedings of JIGASTOCK '88, Fourth International Conference on Energy Storage*, Versailes, pp. 121–124 (1988).
5. C. Doughty, E. Korin, A. Nir and C. F. Tsang, Seasonal storage of thermal energy in unsaturated soils: simulation and field validation—I, *STES* XII(1), XII, 1–5 (1990); II, *STES* XII(2), 5–8 (1990).
6. C. Doughty, E. Korin, A. Nir and C. F. Tsang, Seasonal storage of thermal energy in unsaturated soils: modelling simulation and field validation, *Proceedings of THERMASTOCK '91, Fifth International Congress on Thermal Energy Storage*, pp. 4.3.1–4.3.6, Scheveningen, Holland, 13–16 May (1991).
7. A. Nir, C. Doughty and C. F. Tsang, Validation of design procedure and performance modeling of a heat and fluid transfer field experiment in the unsaturated zone, *Adv. Water Res.* **3**, 153–166 (1992).
8. Y. Rabin, E. Korin and E. Sher, Simplified model for a helical heat exchanger for long-term energy storage in soil, *Proceedings of Eurotherm Seminar No. 18, Design and Operation of Heat Exchangers*, Hamburg, 27 Feb.–1 March 1991 (Edited by W. Roetzel, P. T. Heggs and D. Butterworth), pp. 304–314. Springer, Berlin (1992).
9. Y. Rabin, Long term energy storage in the ground using heat storage elements containing phase change materials, MSc Thesis, The Faculty of Engineering Science, Ben-Gurion University of the Negev, Beer-Sheva (1991) (in Hebrew).
10. J. C. Jager and H. S. Carslaw, *Conduction of Heat in Solids*, pp. 199, 397. Clarendon Press, Oxford (1959).
11. E. E. Korin and A. Nir, Development of seasonal thermal energy storage in unsaturated soil, Internal report no. BGUN-ARI-30-91, The Institutes for Applied Research, Ben-Gurion University of the Negev, Beer-Sheva, Israel (1991).