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Modeling of Joint Operation of a Ground Soil Heat Exchanger and a Thermal Pump Evaporator

Bogdan Sydorchuk[1], Oleksandr Naumchuk[2], Paweł A. Mazurek[3*]

[1] Department of Automation and Computer-Integrated Technologies, National University of Water and Environmental Engineering, Ukraine, 33028, Rivne, 11 Soborna St., Ukraine

[2] Department of Automation and Computer-Integrated Technologies, National University of Water and Environmental Engineering, Ukraine, 33028, Rivne, 11 Soborna St., Ukraine

[3] Faculty of Electrical Engineering and Computer Science, Lublin University of Technology, ul. Nadbystrzycka 38A, 20-618 Lublin, Poland

[*] Corresponding author's e-mail: p.mazurek@pollub.pl

ABSTRACT

The development of civilization is associated with an increase in demand for energy. Nowadays, the technologies using renewable energy sources are particularly important. The heating systems powered by heat pumps are increasingly used. The article presents an integrated mathematical model that takes into account the joint operation of the ground heat exchanger and the internal circuit of the heat pump. Modeling was done in Matlab environment. The modeling results allow optimizing the evaporator design in order to effectively use the heat pump.

Keywords: ground heat exchanger, thermal pump

INTRODUCTION

At present, the issue of using alternative sources of heat supply, one of which includes geothermal heat pumps, is particularly relevant. The main element of the heat supply system of a geothermal heat pump is the heat-exchange apparatus that transmits the low-potential heat from the soil to heat consumers. An important element of such a system is a circuit including a soil heat exchanger and a thermal pump evaporator. The heat withdrawn from the soil is transferred to a heat transfer agent of the soil heat exchanger and then to a refrigerant in the evaporator. In the said circuit, freon of a certain grade is usually used as a refrigerant. The objective of the study was to develop an integrated mathematical model that takes into account the joint operation of the ground heat exchanger and the internal circuit of the thermal pump.

The processes of heat energy transfer from soil to the heat transfer agent of the soil heat exchanger, and the processes of heat transfer in the thermal pump have been studied by many authors. In particular, in the article [Tarasova et al. 2011] the calculation of a non-stationary temperature field of a soil body and its influence on the thermodynamic modes of operation of a thermal pump during the year was presented, and in the article [Bomba et al. 2014] a model that allows calculating the temperature distribution in a heat exchanger tube with consideration of the convective diffusion transfer of heat was created. In the article [Vasilev 2006], the convective heat transfer in a soil heat exchanger was taken into account, and in the article [Eskilson and Claesson 1988] the temperature distribution of a wall by the depth of a soil heat exchanger was determined. The articles [Rej and Makmajkl 1982, Yantovskij and Pustovalov 1982] present the results of basic studies in the field of thermal pump calculation, and discuss various thermodynamic cycles of their operation. Additionally, several studies have been carried out over the last decades to recognize how the use insulation material can reduce the heat losses [Lewkowicz et al.

2018]. Modeling of individual segments of thermal pumps was performed in the articles [Chen et al. 2002, Cao and Faghri 1994, Nishikawara et al. 2014]. In the article [Chen et al. 2002], a modeling of compressors was carried out and in the articles [Cao and Faghri 1994, Nishikawara et al. 2014] a research on evaporators of thermal pumps was conducted.

FORMULATION OF THE PROBLEM

A common mathematical model can be considered that takes into account the joint operation of a ground heat exchanger and a thermal pump circuit. For this model, the subjects of research are an evaporator of the thermal pump and a soil heat exchanger (Fig. 1).

Along the evaporator (Fig. 1), a refrigerant (freon) flows which has a reduced temperature, and is heated by heat of an ethylene glycol solution from a ground reservoir, begins to evaporate into a gaseous state. In this case, the temperature of the ethylene glycol solution decreases. In our research, the evaporator as a tube-in-tube heat exchanger is taken as a basis in the assumption; then, the freon enters the evaporator in a liquid state and no heat loss to the environment occurs.

The material balance equation for the freon in the evaporator is written down as

$$\frac{\mathrm{dm}_f}{\mathrm{dt}} = Q_f - Q_{\mathrm{fr}} \tag{1}$$

where: m_f is the mass of freon in the evaporator,
Q_f, Q_{fr} is the consumption of freon in the liquid and gaseous states, respectively.

The energy balance equation for the freon in the liquid state is written down as

$$c_{\mathrm{pf}}\frac{d(m_f T_f)}{\mathrm{dt}} = \mathrm{kF}(T_k - T_f) + Q_f c_{\mathrm{pf}} T_{f0} - Q_{\mathrm{fr}} i_f \tag{2}$$

where: $k = V\frac{d\rho_f(T_f)}{\mathrm{dT}_f}$ is the coefficient of heat transfer through the heating surface,
V is the amount of freon in the evaporator,
L is the length of evaporator,
C_{pf} is specific heat of freon,
T_f, T_{f0} is the temperature of freon in the evaporator and its inlet, respectively,
χ_{2i} is the temperature of the ethylene glycol solution in the evaporator,
y_{01} is the coefficient of heat transfer through the heating surface,
y_{1i} is the heat transfer surface between the ground reservoir ethylene glycol solution and freon,
Q_{fr} is the amount of heat expended in the evaporation of freon,
i_f is specific enthalpy of freon formed in the gaseous state,
$i_f = \lambda + C_{pf}T_f$, λ is the specific heat of freon evaporation.

By differentiating the left-hand side of equation (1) as a function of two arguments, and by equating to (2), the following equation is obtained

$$c_{\mathrm{pf}} m_f \frac{\mathrm{dT}_f}{\mathrm{dt}} = \mathrm{kF}(T_k - T_f) - \\ - Q_f c_{\mathrm{pf}}(T_f - T_{f0}) - Q_{\mathrm{fr}}(i_f - c_{\mathrm{pf}} T_f) \tag{3}$$

A similar equation is obtained for the ground reservoir ethylene glycol solution passing through the evaporator:

$$c_{\mathrm{pk}} m_k \frac{\mathrm{dT}_k}{\mathrm{dt}} = \mathrm{kF}(T_f - T_k) - \\ - Q_k c_{\mathrm{pk}}(T_k - T_{k0}) \tag{4}$$

where: C_{pk}, m_k is specific heat and mass of the ethylene glycol solution,

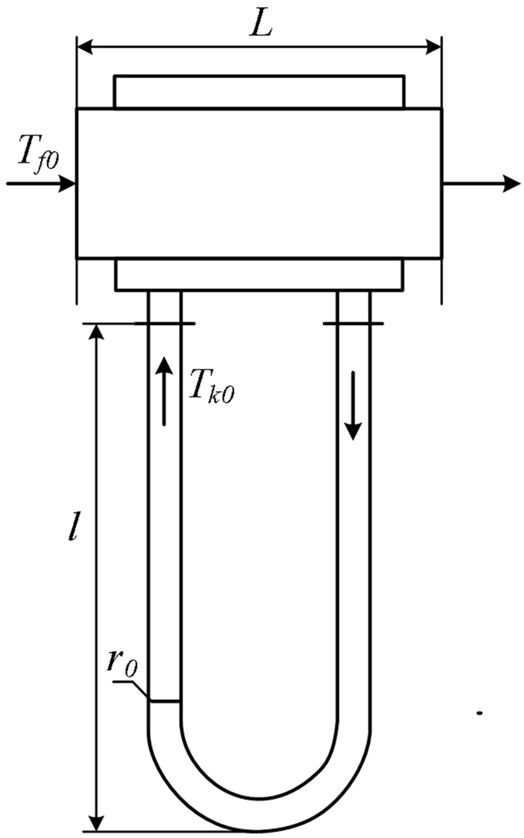

Fig. 1. Thermal pump evaporator and soil heat exchanger

T_{k0} is the temperature of the ethylene glycol solution at the evaporator inlet,

Q_k is consumption of the ethylene glycol solution.

Thus, the following system of equations is proposed to solve the above-mentioned problem:

$$\begin{cases} c_{pf} m_f \frac{dT_f}{dt} = kF(T_k - T_f) - Q_f c_{pf}(T_f - T_{f0}) - Q_{fr}(i_f - c_{pf} T_f), \\ c_{pk} m_k \frac{dT_k}{dt} = kF(T_f - T_k) - Q_k c_{pk}(T_k - T_{k0}) \end{cases} \quad (5)$$

Value T_{k0} can be determined according to [Bomba et al. 2014] by the model of two-dimensional non-stationary process of convective diffusion transfer of heat in a tube of the soil heat exchanger. In this case, in the tube having the length l and radius r_0, the convective diffusion process of heat transfer is considered, which is described by the following model problem:

$$a\left(\frac{\partial^2 T_g}{\partial x^2} + \frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T_g}{\partial r}\right)\right) - v(x)\frac{\partial T_g}{\partial x} = \frac{\partial T_g}{\partial t} \quad (6)$$

$$T_g(x.r.t)\big|_{t=0} = T_{g0}^0 T_g \cdot (x,r)\,, \qquad T_g(0.r.t) = T_g \cdot (r,t) \quad (7)$$

$$\left.\frac{\partial T_g(x,r,t)}{\partial x}\right|_{x=l} = 0\,,\quad \left.\frac{\partial T_g(x,r,t)}{\partial x}\right|_{r=0} = 0\,,\quad \left.\frac{\partial T_g(x,r,t)}{\partial x}\right|_{r=r_0} = -\alpha\left(T_g(x,r_0,t) - T_г(x,t)\right) \quad (8)$$

where: $Tg(x, r, t)$ is the temperature of the ethylene glycol solution in the tube of the soil heat exchanger at the point with coordinate (x, r) at the time point t,

$v(x)$ is the convective transfer rate,

α is the coefficient of thermal conductivity $(\alpha = \lambda/(\rho c)$,

λ is the heat transfer coefficient,

ρ, c are the density and heat capacity of the ethylene glycol solution, respectively),

l, r_0 are the length and radius of the soil heat exchanger,

$T_r(\chi,t)$ is the soil temperature in the location zone of the soil heat exchanger,

α is the heat emission coefficient.

We believe that all the functions appearing in assumptions (7), (8) are sufficiently consistent along the edges and angular points of the area.

$$D = \{(x,r,t): 0 \le x \le l, 0 \le r \le r_0, 0 \le t \le T_g < \infty\}$$

RESEARCH METHODS AND MATERIALS

The solution of equation (6) is obtained as an asymptotic series used to determine the temperature of the ethylene glycol solution $T_g(l,r,t) = T_{k0}$ at the inlet of the thermal pump evaporator [Bomba et al. 2014].

In order to model using the MatLab program, the system of equations (5) may be transformed as follows:

$$\begin{cases} \frac{dT_f}{dt} = \frac{1}{c_{pf} m_f}\left(kFT_k + Q_f c_{pf} T_{f0} - Q_{fr} i_f - T_f\left(kF + Q_f c_{pf} - Q_{fr} c_{pf}\right)\right), \\ \frac{dT_k}{dt} = \frac{1}{c_{pk} m_k}\left(kFT_f + Q_k c_{pk} T_{k0} - T_k\left(kF - Q_k c_{pk}\right)\right) \end{cases} \quad (9)$$

A numerical experiment was performed for the subject under study based on the modeling results. The soil heat exchanger tube with radius of $r = 0.02m$ is filled with the ethylene glycol solution having the following properties: heat transfer coefficient $\lambda = 0.43W/ms$, density $\rho = 1060kg/m^3$, heat capacity $c= 3.31kJ/kg.$ The transfer rate of a heat transfer agent in the heat exchanger tube is $v = 1\ m/s$. It is assumed that the initial temperature distribution in the tube is $T_{go}^0(x,r) = 5°C$, and the heat transfer agent enters the inlet with the temperature $T_g \cdot (r,t) = 2°C$. The length of the heat exchanger tube is $l = 100ml$. For the evaporator, the parameters of the heat transfer agent at points on the saturation line and condensation of the freon of $R22$ grade were determined by the results of the obtained value T_{k0}. Mass flow rate of the refrigerant is 0.06 kg/s. As a result, the model of joint operation of the ground reservoir and the thermal pump evaporator is obtained (Fig. 2).

This model was developed in the Simulink environment of the Matlab program using the method of reducing the order of derivative and allows studying the static and dynamic characteristics of the subject under investigation taking into consideration the heat transfer process in the soil heat exchanger tube.

As it passes through the evaporator, freon, being warmed up with a low-potential heat, starts evaporating, while cooling the ethylene glycol solution to a temperature of 4°C. This process is reflected in the graph (Fig. 3).

The evaporator transient characteristics were constructed, where the change rate of the heat transfer agent temperature was used as a control effect (Fig. 4).

As can be seen, when modeling the mode of start-up of the heat transfer agent, under the given

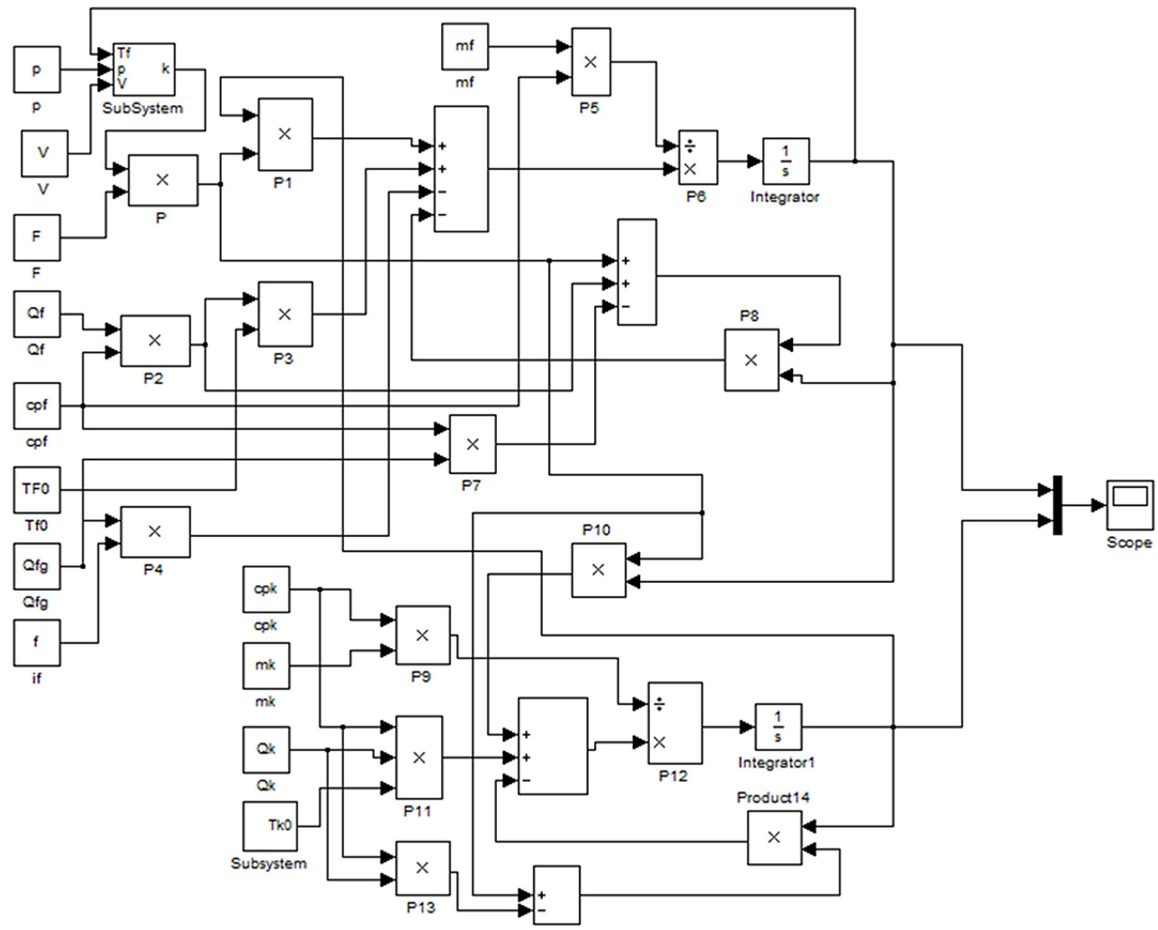

Fig. 2. Study model of joint operation of the ground reservoir and the thermal pump evaporator

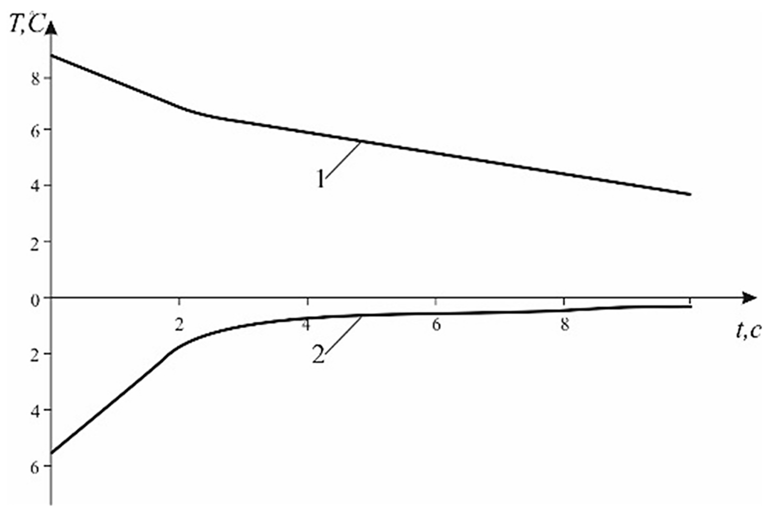

Fig. 3. Temperature change graph: 1 – T_k of ethylene glycol solution, 2 – T_f of freon in the evaporator, at T_{k0} = 6°C. T, °C t,c, °C

conditions, a higher gain ratio and less time constant are observed for $v_3(\chi)$.

Figure 5 presents the changes in the temperatures of the ethylene glycol solution and refrigerant at the evaporator outlet, depending on the temperature obtained from the soil heat exchanger.

As can be seen, as the temperature T_{k0} obtained from a soil heat exchanger decreases from 8 to 4.5ºC, no significant decrease in the

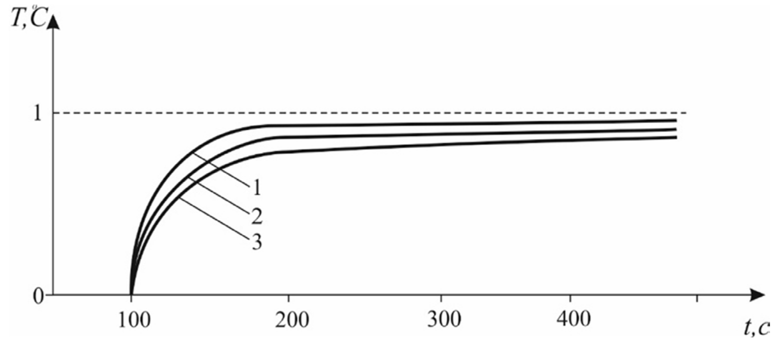

Fig. 4. Acceleration characteristics of the evaporator at different increments of the heat transfer agent temperature rate at the evaporator inlet: 1 – v_1 (x) 0 to 0,2 (m/s) ; 2 – v_2 (x) 0 to 0,5 (m/s) ; 3 – $v_3(\chi)$ 0 to 1,0 (m/s)

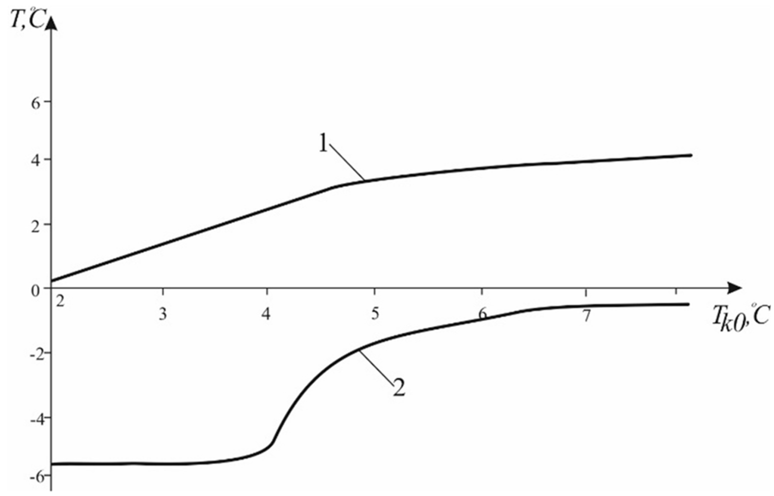

Fig. 5. Graph of temperature changes: 1 – ethylene glycol solution, 2 – refrigerant at the evaporator outlet

refrigerant temperature is observed. When changing T_{k0} from 4.5 to 0ºC, a significant decrease in refrigerant temperature is observed, which is associated with a decrease in refrigerant void fraction at the evaporator outlet.

CONCLUSIONS

An approach to modeling the joint operation of the soil heat exchanger and the thermal pump evaporator was proposed. When operating geothermal heat pumps on a long-term basis, the situations may occur where the soil temperature near the soil heat exchanger decreases during the heating season and in the summer period the soil does not have time to warm up to the initial temperature, i.e. there is a general decrease in the soil temperature potential. In this case, the energy consumption during the next heating period causes a further decrease in soil temperature, and its temperature potential decreases even more. According to the numerical calculation results, the following was established: in case of a decrease in the temperature obtained from the soil heat exchanger from 4.5 to 0ºC, a significant decrease in the refrigerant temperature is observed, which is associated with a decrease in the refrigerant void fraction at the evaporator outlet. Thus, studies have shown that in order to further use the thermal pump effectively, it is necessary to implement changes to the evaporator design for the residual evaporation of freon.

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