Analysis of ground massif temperatures with horizontal heat exchanger

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Abstract

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The paper is aimed at specification of ground massif temperatures in the horizontal heat exchanger area in both the heating season and during heat exchanger stagnation. The energetic potential of the ground massif was evaluated using the difference of temperatures of the ground massif in the area of the heat exchanger at the beginning and at the end of the heating season. Specific heat rates of the ground heat exchanger were also determined, and the influence of the ground massif thermal resistance and coefficient of heat transfer between the inner wall of the heat exchanger pipe and the heat-carrying liquid were analysed.

Keywords: heat pump; heat conduction; heat transfer; heat transfer fluid; thermal power

In various fields of agricultural production the consumption of fossil fuels can be decreased or eliminated using heat pumps. The issue of the use of heat pumps in agricultural production, with respect to the high prices of energy and environmental protection requirements, is studied even abroad.

From the worldwide survey on use of geothermal energy for direct energy consumption (Lund et al. 2011), based on an inquiry in 78 countries, it follows that in 2009 the total installed power of geothermal resources was 48,493 MW. Totally 423,830 TJ per year of thermal energy was obtained from these resources. Approximately 47.2% of this energy was obtained using heat pumps earth-water. The energy was used for heating residential and industrial objects, for heating glasshouses (5.5%), heating of production ponds (2.7%), for agricultural drying (0.4%) and for cooling (0.2%). The implementation of energetic systems using geothermal heat is annually increased by 7%.

In Turkey, at the Gediz University and the Arge University (Gungor et al. 2011), a heat pump driven by a gas engine was verified during drying of medicinal and aromatic plants. The effectiveness of the system was increased by proven electrical energy savings and the possibility of using engine waste heat. The energetic effectiveness of this system varied within the range of 39.34 to 43.56%. At Seul National University (South Korea), mathematical models of energetic systems with heat pumps were prepared with the aim to decrease the quantity of energy consumption during the production of vegetables and flowers (Catton et al. 2011). The possibilities of using heat pumps not only for the heating of glasshouses, and heating of water in production ponds, but also for the drying of cereals and milk pasteurisation, are studied in cooperation at the Saint Mary's University in Halifax (Canada) and at the Hokkaido University, Japan (Tarnawski et al. 2009).

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Stable energy sources for heat pumps, minimally dependent on the temperature of the environment, are vertical and horizontal heat exchangers deposited in the rock or ground massif. The implementation of vertical rock heat exchangers is more demanding from the point of view of investments than of horizontal heat exchangers, but they require a minimum area. In agricultural farms and rural areas there are, however, sufficient areas for horizontal ground heat exchangers and therefore it can be supposed that their implementation will be preferred in these areas.

The aim of this paper is to determine temperatures and analyse the ground massif temperature changes with the horizontal heat exchanger used as a heat source for the heat pump. Also, to determine specific heat rates transferred from the ground massif and analyse the possibilities of increased heat rates.

Horizontal ground heat exchangers for heating (5.5 kW) and cooling (3.3 kW) were verified by the employees of Hokkaido University, Japan (Tarnawski et al. 2009). The consumption and costs of electric energy were compared to those of boiler heating for the combustion of fuel oil and using electrical resistance heating. During verification, low thermal degradation of the ground massif was proven in the heating season. Although there was a relatively high consumption of electrical energy in the heat pump, when compared to other heat sources from the point of view of investments and operating costs, the heat pump appeared to be the most advantageous alternative. The result of the verification was used as a base for area implementation in both the residential and commercial zone in northern Japan. In the Research Center for Energy and Environment in Lecce, Italy (Congedo et al 2012), horizontal ground exchangers in 3 configurations – linear, spiral and Slink type – were verified. In the year-long operation, ground massif temperatures and heat flows transferred by the exchanger were monitored. The results showed that the most important parameters, from the point of view of the heat flow transferred from the ground massif, are the coefficient of thermal conductivity of the massif and the flow rate of the heat-carrying liquid in the heat exchanger pipes. The role of the depth of exchanger deposition in the ground massif was not so important. From the point of view of geometric arrangement, the best results were for the horizontal spiral heat exchanger. At the University of Buffalo, USA (Rezaei et al. 2012), the influence of various coverage of the ground massif surface with the horizontal heat exchanger on the temperature distribution and on heat flows transferred from the ground massif was studied. A positive influence of the coverage was proved. E.g. the coverage of the massif surface with an insulating layer from recycled tyres caused the heat flow transferred from the ground massif in the winter season to increase by 17%.

MATERIAL AND METHODS

Theoretical reflection on the issue. The initial principle, during the theoretical analysis of the issue of the heat exchange in the ground massif with the horizontal heat exchanger, is the general Fourier-Kirchhoff equation of heat convection. In this case, it is suitable to use equations in the form of cylindrical coordinates, because it is a cylindrical symmetry issue.

$$\frac{\partial t}{\partial \tau} = a\left(\frac{\partial^2 t}{\partial r^2} + \frac{1}{r}\frac{\partial t}{\partial r} + \frac{1}{r^2}\frac{\partial^2 t}{\partial \varphi^2} + \frac{\partial^2 t}{\partial z^2}\right) \quad (1)$$

where:

t – temperature of ground massif (°C)
τ – time (s)
a – coefficient of thermal conductivity (m^2/s)
r – point distance from z-axis (m)
φ – angle of projection of the point connecting line to the plane x, y (°)
z – position of the point on z-axis (m)

During the stationary heat exchange $t = f(r, \varphi)$, the term $\partial t/\partial \tau = 0$. Non-stationary heat exchange can have three forms:

(a) $t = f(r, \varphi, \tau)$.

Eq. (1) is in the form:

$$\frac{\partial t}{\partial \tau} = a\left(\frac{\partial^2 t}{\partial r^2} + \frac{1}{r}\frac{\partial t}{\partial r} + \frac{1}{r^2}\frac{\partial^2 t}{\partial \varphi^2}\right) \quad (2)$$

Eq. (2) describes heat convection in a short time interval. The analytical solution of the equation is not known at present. For numerical solution, the Final Differences Method (FDM), Final Elements Method (FEM) or Final Volumes Method (FVM) are used.

(b) $t = f(r,z,\tau)$.

Eq. (1) is in the form:

$$\frac{\partial t}{\partial \tau} = a\left(\frac{\partial^2 t}{\partial r^2} + \frac{1}{r}\frac{\partial t}{\partial r} + \frac{\partial^2 t}{\partial z^2}\right) \quad (3)$$

Eq. (3) describes the heat convection from a long-term point of view (years).

(c) $t = f(r,\tau)$.

Eq. (1) is in the form:

$$\frac{\partial t}{\partial \tau} = a\left(\frac{\partial^2 t}{\partial r^2} + \frac{1}{r}\frac{\partial t}{\partial r}\right) \tag{4}$$

According to the theory of line source (Ingersoll et al. 1954), the analytical solution of Eq. (4) is Eq. (5), which is valid for $a\tau/R^2 > 20$:

$$t = \frac{q}{2\pi\lambda_z}\int_{\frac{r}{2\sqrt{a\tau}}}^{\infty}\frac{e^{-\beta^2}}{\beta}d\beta \quad (°C) \tag{5}$$

where:

λ_z – coefficient of thermal conductivity of ground massif (W/m.K)

q – specific heat flow through the pipe of the ground heat exchanger (W/m)

R – specific heat resistance of ground massif (m.K/W)

$$\beta = \frac{r}{2\sqrt{a(\tau - \tau_1)}} \quad (–) \tag{6}$$

When simplified, after substitution of the integral, Eq. (5) is in the form (Mogensen 1983):

$$t = \frac{q}{4\pi\lambda_z}\left[\left(\ln\frac{4a\tau}{r^2}\right) - \gamma\right] \quad (°C) \tag{7}$$

where:

$\gamma = 0.5772$ – Euler's constant

Thermal transfer from the ground massif to the heat-carrying liquid flowing in the heat exchanger pipe of the horizontal heat exchanger consists of heat exchange by conduction in the ground massif, conduction in the pipe wall and convection between the inner wall of the pipe and the heat-carrying liquid. The scheme of the processes of the heat exchange is presented in Fig. 1.

The total thermal resistance equals the sum of the thermal resistance of the ground massif R_z, thermal resistance of the pipe wall R_t of the heat exchanger and the thermal resistance by convection R_α on the inner wall of the pipe.

During the determination of the heat conduction through a semi-bounded massif with a row of pipes, it is suitable to use the method of sources and principle of superposition of temperature fields described by Šorin (1968). When using this method, it is supposed that the temperature fields formed by the sources of heat or heat consumptions and a marginal condition for the heat transfer on the ground massif surface can be expressed by linear differential equations. Then, the thermal resistance of the ground massif R_z can be calculated from Eq. 8 (Šorin 1968):

$$R_z = \frac{1}{2\pi\lambda_z}\times\ln\left[\frac{2s}{\pi d_2}\sinh\left(2\pi\frac{h}{s}\right)\right] \quad (m.K/W) \tag{8}$$

where:

d_2 – outer diameter of pipes of ground massif (m)

h – depth of deposition of pipes of the heat exchanger (m)

s – spacing of pipes of ground heat exchanger (m)

The thermal resistance of the heat exchanger pipe is determined according to the known formula for the heat exchange by conduction through the cylindrical wall:

$$R_t = \frac{\frac{1}{\lambda_t}\times\ln\frac{d_2}{d_1}}{2\pi} \quad (m.K/W) \tag{9}$$

where:

λ_t – coefficient of thermal conductivity of heat exchanger pipe material (W/m.K)

d_1, d_2 – inner and outer diameter of heat exchanger pipe (m)

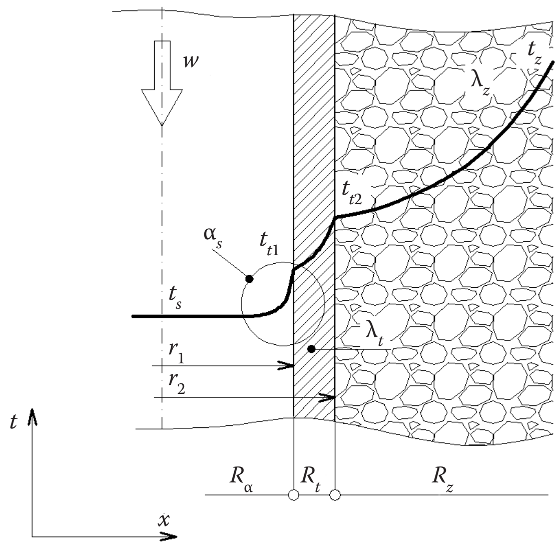

Fig. 1. The scheme of heat exchange processes between the ground massif and heat-carrying liquid

R_z – thermal resistance of the ground massif (m.K/W); R_t – thermal resistance of the pipe wall of the heat exchanger (m.K/W); R_α – thermal resistance by convection on the inner wall of the pipe (m.K/W); λ_t – coefficient of thermal conductivity of heat exchanger pipe material (W/m.K); λ_z – coefficient of thermal conductivity of ground massif (W/m.K); α_s – coefficient of heat transfer between the pipe wall and the heat-carrying liquid (W/m^2.K); r_1, r_2 – inner and outer radius of heat exchanger pipe (m); w – velocity of heat transfer fluid (m/s); t_{t1}, t_{t2} – temperature inside and outside the tube wall (°C); t_z – temperature of the ground massif (°C); t_s – temperature heat transfer fluid (°C)

The thermal resistance by convection R_α on the inner wall of the pipe can be expressed using the equation:

$$R_\alpha = \frac{1}{2\pi r_1 \alpha_s} \quad \text{(m.K/W)} \tag{10}$$

where:

$$\alpha_s = \frac{Nu \times \lambda_s}{d_1} \quad \text{(W/m}^2\text{.K)} \tag{11}$$

where:

α_s – coefficient of heat transfer between the pipe wall and the heat-carrying liquid (W/m^2.K)

Nu – Nusselt criterion (–)

λ_s – coefficient of thermal conductivity of heat-carrying liquid (W/m.K)

Measurement method. The measurement was performed within the VESKOM spol. s r.o. premises in Prague, Czech Republic. Veskom spol. s.r.o. implemented a large-scale experimental facility within its premises for verification of ground sources for heat pumps, including boreholes, horizontal ground heat exchangers and ground heat exchangers of the Slink type.

The horizontal ground heat exchanger, which is the subject of verification, was manufactured from PE 100RC 40 × 3.7 mm (Luna Plast a.s., Hořín, Czech Republic) polyethylene pipes resistant to point pollution and formation of cracks. It is not deposited in the sand bed. The pipes of the heat exchanger, with a total length of 335 m, are installed at the depth of 1.8 m in 3 loops with a spacing of 1 m. The ground massif to the depth of ca. 2 m is formed by dark sandy-clay soil, coarse gravel, scappling and broken bricks. The heat-carrying liquid flowing through the heat exchanger is a mixture of 33% of ethylalcohol and 67% of water. The scheme of location of temperature sensors PT 1000A (Greisinger Electronic GmbH, Regenstauf, Germany) and a heat consumption meter MTW 3 (Itron Inc., Liberty Lake, USA) is presented in Fig. 2.

Temperatures of the ground massive were measured using resistance temperature sensors and were recorded in 15 min intervals in a datalogger ALMEMO 2890-9 (Ahlborn Mess-und Regelungstechnik GmbH, Holzkirchen, Germany). Temperatures of environment t_e were measured at a height of 3 m above the soil surface at the distance of 20 m from the horizontal ground heat exchanger. The electronic heat consumption meter was used for the measurement of the heat flow transferred by the horizontal heat exchanger.

RESULTS AND DISCUSSION

Temperatures of the ground massif

The measurement was performed in the period from 01/03/2011 to 29/02/2012. The average temperatures of the ground massif recorded between 15:00 and 16:00 are presented in Fig. 3. The maximum temperature differences were t_{p02}–t_{z02} = 1.05 K and t_p–t_z = 1.5 K. In the chart in Fig. 3 there are only the temperature courses of t_{p02} and t_p. There is only a slight delay of the temperature course of t_{p02} at the depth of 0.2 m with respect to air temperatures above the ground massif t_e. A higher delay and a slight reaction to the ambient air temperature were seen for the temperature t_p at the depth of 1.8 m. The generally known fact that due to a low value of the coefficient of thermal conductivity of the ground massif and a high specific heat capacity, the amplitudes of temperature changes of the ground massif decrease with the depth of the ground massif when compared to the air temperature above its surface, is valid even during the transfer of the heat flow from the ground massif by the installed heat exchanger.

The temperature course t_p of the ground massif at the depth of 1.8 m can be divided into 3 phases. In the

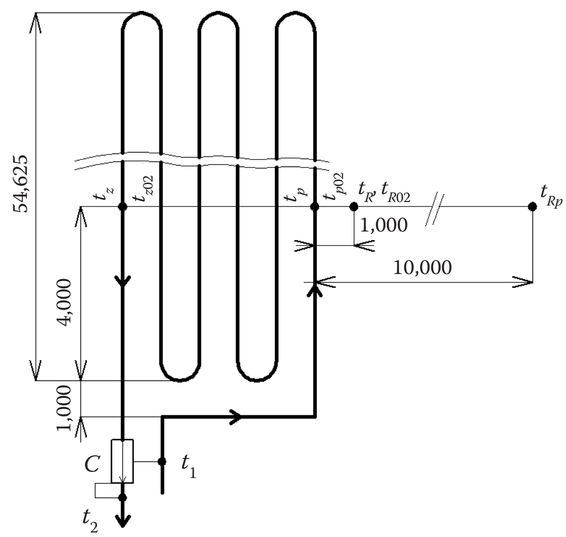

Fig. 2. Scheme of the horizontal ground heat exchanger and specification of measurement points

t_z – temperature sensor located in the depth of 1.8 m near pipes heading to the evaporator of the heat pump; t_p – temperature sensor located at a depth of 1.8 m near the pipes heading from the heat pump evaporator; t_{z02}, t_{p02} – temperature sensors located at a depth of 0.2 m above pipes; t_R – reference temperature sensor located at a depth of 1.8 m; t_{R02} – reference temperature sensor located at a depth of 0.2 m; t_{Rp} – reference sensor located at a depth of 1.2 m; C – electronic heat consumption meter first phase, there is an increase of the ground massif temperature at the end of the heating season. The second phase took place in the period from 12/03/2011 to 16/06/2011. The temperatures of the ground massif in this time period increased from 3.97°C to 13.7°C. The equation in this phase is as follows:

$$t_p = 3.74 \times 10^{-5}d^3 - 3.31 \times 10^{-4}d^2 + 1.04 \times 10^{-1}d + 3.63 \quad (R^2 = 0.998) \quad (°C) \qquad (12)$$

The second phase of the temperature regime in the ground massif takes place during the stagnation of the ground heat exchanger in the summer season from 17/06/2011 to 07/09/2011. The temperature of the ground massif in this time period varies within the range 13.94°C to 16.91°C. The equation in this phase is as follows:

$$t_p = -9.5 \times 10^{-10}d^6 + 2.19 \times 10^{-7}d^5 - 1.85 \times 10^{-5}d^4 + 6.96 \times 10^{-4}d^3 - 1.16 \times 10^{-2}d^2 + 1.23 \times 10^{-1}d + 13.8 \quad (R^2 = 0.985) \quad (°C) \qquad (13)$$

The last, third phase, takes place at the beginning and in the course of the heating season from 08/09/2011 to 27/02/2012. The ground massif temperature gradually decreases from 16.93°C to the minimum value of 3.78°C. The equation in this phase is as follows:

$$t_p = -6.2 \times 10^{-8}d^4 + 2.21 \times 10^{-5}d^3 - 2.53 \times 10^{-3}d^2 + 1.49 \times 10^{-2}d + 16.8 \quad (R^2 = 0.991) \quad (°C) \qquad (14)$$

From the point of view of energetic potential of the source and its operation life, it is very interesting to compare initial and final temperatures of the ground massif in the area of the heat exchanger during several heating seasons. The temperature difference of the ground massif t_p at the end of the heating seasons 2009–2010 (21/06/2010) and 2010–2011 (16/06/2011) was only 0.05 K. The temperature difference at the beginning of the heating seasons 2010–2011 (06/09/2010) and 2011–2012 (08/09/2011) was 0.04 K. The mentioned temperature differences varying within the measurement precision indicate that the ground massive can be regarded as a very stable energy source for a heat pump. This conclusion corresponds to the results of verification of the horizontal ground heat exchanger by Tarnawski (2009) and also to our verification at the locality of the South Bohemia (Adamovský et al. 2010).

Specific heat rates of the ground heat exchanger

The specific heat rates of the ground heat exchanger were determined based on the measurement volume flow and the temperature difference of the heat-carrying liquid on the input t_1 and on the output t_2 from the ground heat exchanger (Fig. 2). The circulating pump of the heat-carrying liquid was operated in two stages at 0.0000833 m^3/s and 0.000209 m^3/s. The thermodynamic variables necessary for calculations were determined based on literature (Jahoda et al. 2010). The example of the course of specific heat flows q in a typical day of the heating season, 06/02/2012, is presented in Fig. 4.

The course of the dependence $q = f(\tau)$ reacts on the stay of workers in production halls and offices. In time periods between 5:24 to 5:54 and 21:39 to 22:54 the circulation pump of the heat exchanger worked at lower level. Specific heat output of the exchanger varied between 2.79–3.56 W/m. In peak heat pump operation times between 6:09 and 14:39, the values of the heat flow q transferred from the ground massif using 1 m pipes of the heat exchanger varied within 7.85–11.50 W/m, which corresponds to the heat rate of 8.76–12.83 W transferred from the 1 m^2 area where the heat exchanger was installed. The found specific heat flows correspond to the values presented in the

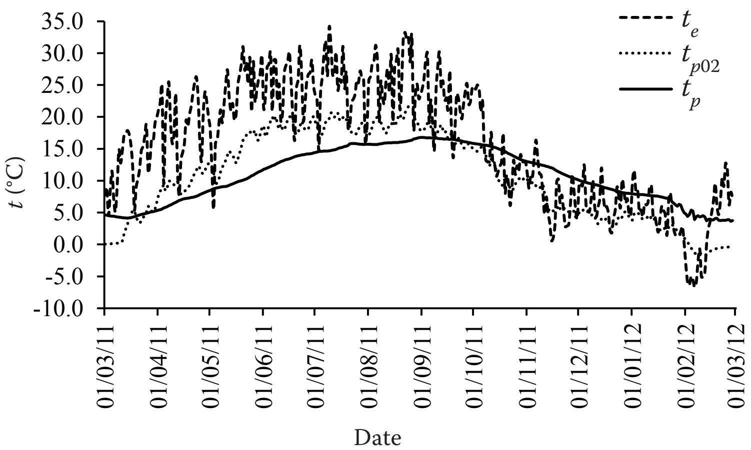

Fig. 3. Ground massif temperatures with horizontal heat exchanger in the period of 01/03/2011 to 29/02/2012

t_e – ambient temperature, tp_{02} – temperature of the ground massif at depth 0.2 m above the ground heat exchanger tubes, tp – temperature of ground massif at depth 1.8 m in the ground heat exchanger

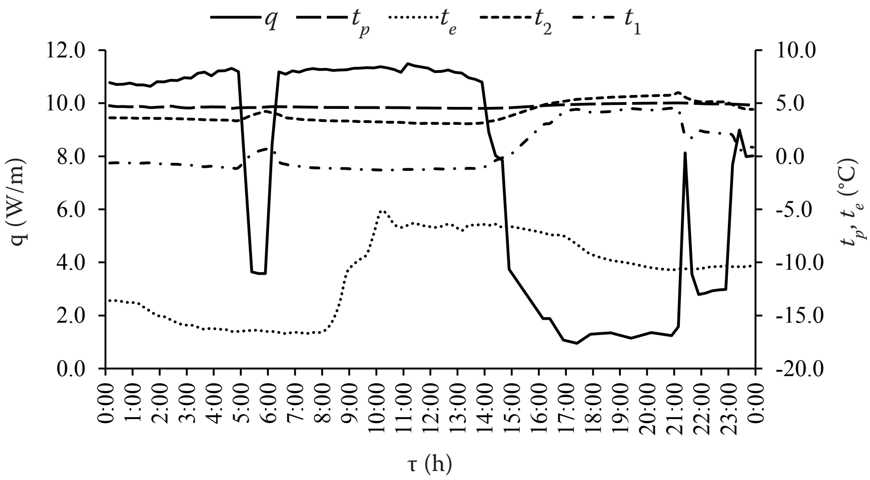

Fig. 4. Specific heat exchanger rates 06/02/2012

q – specific heat flow, t_e – ambient temperature; t_p – temperature of ground massif at depth 1.8 m in the ground exchanger; t_1, t_2 – temperature if the heat carrying liquid from the input and output of the heat exchanger (respectively)

literature (Brandl 2006). In the 16:09–21:09 time period the heat is not transferred from the ground heat exchanger. The ground massif temperature t_p at a depth of 1.8 m in the area of the heat exchanger is thus lower than the temperature of the heat-carrying media t_2 coming out of the heat exchanger.

The main external factors influencing the heat exchange in the ground massif with the horizontal heat exchanger are thermal characteristics of the massif, especially the coefficient of thermal conductivity λ (Congedo et al. 2012). It is especially influenced by the moisture, density, compression, portion of clay, underground and surface water. The process of the heat exchange can be influenced by the heat exchanger parameters, depth and spacing of the heat exchanger pipes, its diameter and the flow rate of the heat-carrying liquid.

The process of the heat exchange in the ground massif with the horizontal ground heat exchanger can be assessed in a simplified way using the value of thermal resistance of the ground massif R_z, which respects the three mentioned parameters of the heat exchanger. The lower the value of thermal resistance R_z, the higher the heat flow exchanged in the ground massif. It follows from Eq. (8) that the considered coefficient of thermal conductivity of the ground massif $\lambda = 1.7$ W/m.K and parameters of the verified ground heat exchanger, pipe diameter $d_2 = 0.04$ m, deposition depth $h = 1.8$ m and spacing of pipes $s = 1.0$ m, resulted in $R_z = 1.25$ m.K/W. The decrease of the deposition depth of pipes of the heat exchanger to $h = 1.5$ m would cause the decrease of the thermal resistance to $R_z = 0.94$ m.K/W, the increase of spacing of pipes to $s = 1.5$ m would cause the decrease of the thermal resistance to $R_z = 1.17$ m.K/W and the increase of the pipe diameter to $d = 0.1$ m would cause the decrease of the thermal resistance to $R_z = 1.17$ m.K/W.

The variable limiting the effect of the ground heat exchanger can be the coefficient of the heat transfer α_s between the inner wall of the heat exchanger pipe and the heat-carrying liquid. Regarding that the heat-carrying liquid is mostly formed by the mixture of water and glycerol, polypropylenglycol, ethylenglycol or, as in our case, ethylalcohol, it has a high cinematic viscosity and predominantly laminar flow is achieved. In the verified ground heat exchanger, Reynolds number was $Re = 598.62$ in the first stage of operation of the circulating pump, and $Re = 1,287.93$ in the second stage. In both cases there was a laminar flow. It is valid for the heat exchanger that the ratio of the length of pipes of the heat exchanger L and its inner diameter d_1 is equal to $L/d_1 >> 50$. So we can neglect the influence of the hydrodynamic start-up length on the Nusselt criterion (Sazima et al. 1989) and for the calculation of a mean value of the coefficient of heat transfer α_s in the total length L under given laminar flow of the heat-carrying liquid in the pipe we can use the equation (Sazima et al. 1989):

$$Nu = \frac{\alpha_s \times d_1}{\lambda_s} = \left[3.65 + \frac{0.0668 \times Re \times Pr \times \frac{d_1}{L}}{1 + 0.045\left(Re \times Pr \times \frac{d_1}{L} \right)^{2/3}} \right] \times \left(\frac{\mu_s}{\mu_{st}} \right)^{0.14} \quad (-) \qquad (15)$$

where:

λ_s – coefficient of thermal conductivity of heat-carrying liquid (W/m.K)
Re – Reynolds criterion (–)
Nu – Nusselt criterion (–)
Pr – Prandtl criterion (–)
Gr – Grashof criterion (–)
L – length of pipes of the ground heat exchanger (m)
μ_s – dynamic viscosity at the mean temperature of the heat-carrying liquid (Pa.s)
μ_{st} – dynamic viscosity at the mean temperature of the inner wall of the heat exchanger pipe (Pa.s)

Eq. (15) is valid within the limits:
Re < 2,300; Gr > 25,000; $10^4 > \mathrm{Re} \times \mathrm{Pr} \times (d_1/L) > 10^{-1}$
The validity conditions are fulfilled.

From Eq. (15) we can calculate the coefficient of the heat transfer α_s = 47.62 W/m^2.K for the first stage of operation of the circulating pump, and α_s = 50.58 W/m^2.K for the second stage. The coefficient of the heat transfer α_s, dependent on the flow rate of the heat-carrying liquid w, is always higher for the turbulent flow than for the laminar flow. However, the influence of the turbulent flow cannot be overestimated. Achievement of turbulent flow for non-freezing heat-carrying liquids induces a significant increase of the output of the circulating pump and thus decreases the energy effect of the whole system.

CONCLUSION

The result of our verification is knowledge of the temperature courses in the ground massif with the horizontal heat exchanger. Eqs (12) to (14) together with Fig. 3 specify the temperature courses of the ground massif in the area of the heat exchanger, both in the heating season and during stagnation of the ground heat exchanger.

The sufficient energetic potential of the ground massif with the horizontal heat exchanger was proved by a several year slight temperature difference of the ground massif at the beginning of the heating season. This fact is also supported by the results of comparison of temperatures of the ground massif in the area of the heat exchanger and temperatures of the ground massif on the reference areas.

The specific heat rates of the ground heat exchanger were determined (Fig. 4) and the influence of thermal resistance of the ground massif R_z and the coefficient of heat transfer α_s between the inner wall of the heat exchanger pipe and the flowing liquid on the process of retrieving of the heat rate using the horizontal ground heat exchanger were analysed.

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