Thermal tests on borehole heat exchangers for ground–coupled heat pump systems

Ioan Sarbu and Horia Bura

***Abstract*—** The mechanical vapor compression heat pumps are modern systems, recently used as an alternative to the thermal fossil fuel stations. The incorrect determination of the vaporization thermal power needed for the ground–coupled heat pumps with vertical closed–loops leads to unfavorable effects for this systems: under–sizing the catching system used for the vaporization of the refrigerant determines the reduction of heat pump nominal thermal power; over–sizing catching system leads to additional investments that puts under discussion the opportunity of using such systems. Therefore is very important to know the thermal conductivity of the soil and the thermal resistance of the vertical ground loop for establish the right number of loops to be realized, depending on energy to be transfered to the heat pump. For this purpose is needed to be made a ground thermal response test, using a prove borehole. In this paper is presented a working methodology and is developed an analytical model for evaluation of the soil thermal conductivity and the borehole thermal resistance, based on which can be calculated the vaporization thermal power that has to be assured from the ground and also the length of the vertical loops. Also, this paper presents an equivalent–time method to remove the effects of the interruption and estimate soil thermal conductivity, along with borehole resistance.

***Keywords*—** Heat pumps, Ground–source, Vertical loops, Vaporization thermal power, Thermal response test, Equivalent–time concept.

I. Introduction

Buildings are an important part of European culture and heritage, and they play an important role in the energy policy of Europe. Studies have shown that saving energy is the most cost effective method to reduce green house gas emissions (GHG). It has also pointed out that building represent the biggest and most cost effective potential for energy savings. The reduction of 26% energy use is set as a goal for buildings by the year 2020 which corresponds to 11% of the reduction of total energy use in European Union (EU) countries.

The buildings sector is the largest user of energy and CO_2 emitter in the EU, and is responsible for more than 40% of the EU's total final energy use and CO_2 emissions. At present heat use is responsible for almost 80% of the energy demand in houses and utility buildings for space heating and hot water generation, whereas the energy demand for cooling is growing year after year. There are more than 150 millions dwellings in Europe. Around 30% are built before 1940, around 45% between 1950 and 1980 and only 25% after 1980 [14]. Retrofitting is a means of rectifying existing building deficiencies by improving the standard and the thermal insulation of buildings and/or the replacement of old space conditioning systems by energy–efficient and environmentally sound heating and cooling systems.

In order to realise the ambitious goals for the reduction of fossil primary energy consumption and the related CO_2 emissions to reach the targets of the Kyoto–protocol besides improved energy efficiency the use of renewable energy in the existing building stock have to be addressed in the near future.

On 17 December 2008, the European Parliament adopted the Renewable Energy Directive. It is establishes a common framework for the promotion of energy from renewable sources. For the first time, this Directive recognises aerothermal, geothermal and hydrothermal energy as renewable energy source. This directive opens up a major opportunity for further use of heat pumps for heating and cooling of new and existing buildings.

Heat pump enables the use of ambient heat at useful temperature level need electricity or other auxiliary energy to function. Therefore, the energy used to drive heat pumps should be deducted from the total usable heat. Aerothermal, geothermal and hydrothermal heat energy captured by heat pumps shall be taken into account for the purposes provided that the final energy output significantly exceeds the primary energy input.

The amount of ambient energy captured by heat pumps to be considered renewable energy E_{res}, shall be calculated in accordance with the following formula:

$$E_{res} = Q_u\left(1-\frac{1}{\mathrm{SPF}}\right) \tag{1}$$

where: Q_u is the estimated total usable heat delivered by heat pumps; SPF – the estimated average seasonal performance factor for these heat pumps.

Only heat pumps for which SPF>1.15/η shall be taken into account, where η is the ratio between total gross production of electricity and the primary energy consumption for electricity production. For EU–countries Average η=0.4. Meaning that minimum value of seasonal coefficient of performance should be SPF=$COP_{seasonal}$>2.875.

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Heat pump enables the use of ecological heat (solar energy accumulated in the soil, water and air) for an economic and ecological heating. For practical use of these energy sources we have to respect the following criteria: sufficient availability, higher accumulation capacity, higher temperature, sufficient regeneration, economical capture, reduced waiting time.

By exchanging heat with the ground or surface water, ground–source heat pump sysems efficiently heat and cool buildings. Ground–coupled heat pumps systems often use vertical ground loops for the heat exchange with the ground [2], [7]. The design of the ground loops depends on the thermal conductivity of the surrounding soil and rock. A larger soil thermal conductivity allows the heat to be exchanged at a larger rate for a given borehole. For a given set of heat input rates during an annual cycle, the required borehole length decreases as the soil thermal conductivity increases.

Because the soil thermal conductivity is such an important parameter, in–situ tests are often performed on a test borehole for larger commercial installations. Reviews of the history and status of in–situ thermal conductivity tests have been written by Gehlin and Spitler [10] and Saner et al. [18]. Early portable test rigs were described by Eklöf and Gehlin [8] and Austin et al [3].

In this paper is presented a working methodology and is developed an analytical model for evaluation of the soil thermal conductivity and the ground loop thermal resistance, based on which can be calculated the vaporization thermal power that has to be assured from the ground and also the length of the vertical loops. Also, this paper presents an equivalent–time method to remove the effects of the interruption and estimate soil thermal conductivity, along with borehole resistance.

II. Operating Principle of Heat Pump

Heat pump is a thermal installation which is based on a reverse thermodynamic cycle (consumes action energy and produces a thermal effect).

Any heat pump takes heat E_S from a low potential thermal source, at temperature t_s and with an energy action E_A it raises the thermal potential and yielde this heat for a consu-mer at t_u temperature.

• *Heat source* can be*:*

– a gas or air (oudoor air, warm air from processes of cooling or ventilation, hot gases from industrial processes);

– a liquid called generic water: surface water (river, lake, sea), ground water (underground water, geothermal water), discharged warm water (domestic, recirculated in cooling towers, technological);

– soil (with the advantage of accessibility and the temperature constance at a depth of over 4 m, but with the disadvantage of low heat transfer).

• *Heat consumer.* The heat pump yields thermal energy at a higher temperature is depending on the application of heat consumer. This energy can be used to:

– spaces heating; heating with heat pump who will be related to heating systems that require low temperature: radiant systems (radiant panels, heating from floor), warm air or convective systems (ventiloconvectors);

– water heating (pools, domestic and technologic warm water);

– achivement of technological processes (drying, distillation of solutions, salt concentration).

It is recommended that whenever possible, the heat consumer to be associated with a cold consumer, in which case, the same installation will achieve both effects: the heat production and cold production. This can be performed with either a reversible (heating–cooling) or a double effect (it produces simultaneously heat and cold) installation.

• *Action energy*. Heat pumps can use to engage different forms of energy:

– electrical energy (electrocompressor installation);

– mechanical energy (mechanical compression installation, with the action energy produced with expansion turbines);

– thermal energy (installation with mechanical compression, absorption or ejection). In this case is required either a fuel feeding the thermal motor of compression installation with an internal combustion motor, or thermochemical compressor digester of absorption installation with direct combustion or a hot fluid (steam, condensate, hot water, warm gases) which supplies the digester of absorption installation or the ejector of ejection installation.

The most used heat pumps plants are those with mechanical compression and absorption.

A. Functional Scheme and Thermodynamic Cycle of Mechanical Compression Heat Pump

Mechanical vapor compression heat pump works by reversibile Carnot cycle, placed in the water vapor domain, but situate above ambient temperature. Figure 1 shows the functional scheme and theoretical thermodynamic cycle of undercooling heat pump. To reduce loss caused by the lamination irreversibility it recourses for inclusion of undercooler in the heat pump scheme with the role to reduce the temperature of saturated liquid refrigerant, below condensation temperature T_c.

Functional processes are the following:

1–2: isentropic compression in the compressor K, which leads to increased pressure and temperature from the values corresponding for vaporization p_0, T_0 to those of the condensation p_c, $T_2 > T_c$ (in the superheated vapour domain);

2–2’: isobar cooling in the condenser C at pressure p_c from the temperature T_2 to $T_{2'} = T_c$;

2’–3: isotherm–isobar condensation in the condenser C at pressure p_c and temperature T_c;

3–3’: isobar undercooling in the under cooler SR at pressure p_c from temperature T_c at $T_{sr}<T_c$;

3’–4: isentalpic lamination in expansion valve VL, leading the refrigerant from 3’ state of the undercooled liquid at p_c, T_{sr} in 4 state of wet vapor at p_0, T_0;

4–1: isotherm–isobar vaporization in the evaporator E at pressure p_o and temperature T_0.

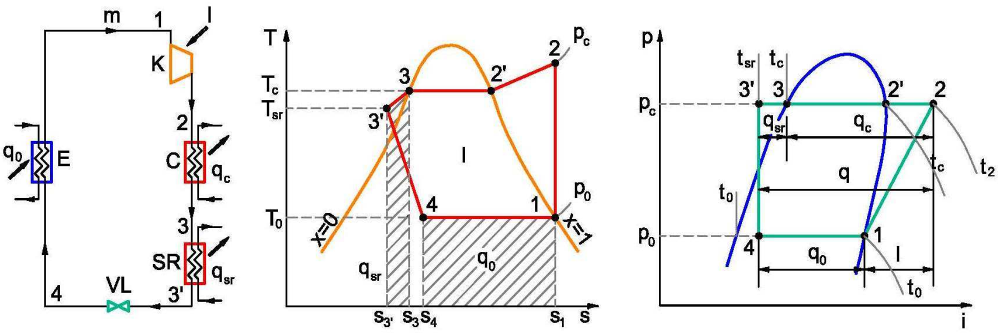


Fig. 1 Functional scheme and thermodynamic cycle of heat pump with undercooling

It result the following relationships and graphic meanings of energy exchanges of refrigerant:

– specific cooling power at the agent vaporization:

$$q_0 = i_1 - i_4 = T_0 (s_1 - s_4) = \text{area } s_4 41 s_1 \tag{2}$$

– specific heat load at condensation:

$$q_c = i_2 - i_3 = \text{area } s_1 22'3 s_3 s_1 \tag{3}$$

– specific heat load at undercooling:

$$q_{sr} = i_3 - i_{3'} = \text{area } s_3 33' s_{3'} s_3 \tag{4}$$

– specific thermal load of the refrigerant:

$$q = q_c + q_{sr} = i_2 - i_{3'} = \text{area } s_1 22'33' s_3 s_1 \tag{5}$$

– specific work of compression:

$$l = i_2 - i_1 = q - q_0 = i_2 - i_{3'} - (i_1 - i_4) \tag{6}$$

– thermal power of heat pump:

$$Q_{PC} = m\, q \tag{7}$$

where m is mass flow of the refrigerant.

– coefficient of performance COP or theoretical efficiency ε_{PC} of the heat pump is expressed trough the ratio:

$$\text{COP} = \varepsilon_{PC} = \frac{q}{l} = \frac{i_2 - i_{3'}}{i_1 - i_2} \tag{8}$$

In heat pumps the undercooling rank $\Delta T_{sr} = T_c – T_{sr}$ can be increased till the achievement of the ambient temperature of the refrigerant liquid, resulting a substantial reduction of loss caused by the irreversibility of the lamination process.

B. Earning Capacity Limit of Heat Pump with Electrocompressor

In this case interfers the global efficiency η_g as a product between the electric energy production efficiency η_p, its transportation efficiency η_t and the electromotor efficiency η_{em}:

$$\eta_g = \eta_p \eta_t \eta_{em} \tag{9}$$

Taking into account that the heat pump has an overunit theoretical efficiency, for the evaluation in which way is valued the consumed primare energy is using the sintethic indicator η_s, representing the product:

$$\eta_s = \eta_g \varepsilon_{PC} \tag{10}$$

which has to satisfy the condition $\eta_s > 1$ for justify the use of heat pump

Also, only if the real efficiency $\varepsilon_{r,PC} > 3$ the use of heat pump can be considered.

The maxim value, theoretic possible, of the efficiency ε_c can be obtained in reversibile Carnot cycle case depending only on absolute temperature level of the hot source T_c and the cold one T_0:

$$\varepsilon_c = \frac{T_c}{T_c - T_0} \tag{11}$$

The real efficiency $\varepsilon_{r,PC}$ of the heat pump is lower than the theoretical maximum one ε_c, representing 40...60% of its vaue. Results that for $\varepsilon_{r,PC}$ to have the value 3, ε_c must be at least 6...7.

If T_c = 70 °C, T_0 must be minimum 12...20°C, achieved condition by the major part of thermal waste. Only in case of the air heat pump, as energy source, we can talk about a limitation of use the plant during the coldest days of the year.

III. Ground–Coupled Heat Pump Systems

The ground has the property to accumulate and maintain solar energy for a longer period of time. That leads to a relatively constant level of heat source temperature for the whole year and thus to a high performance of the heat pumps.

The ground–coupled heat pump system is performed of:

– horizontal or vertical heat exchanger;

– equipment for the preparation of thermal agent using a vapour mechanic compression heat pump.

By exchanging heat with the ground, ground–coupled heat pump systems efficiently heat and cool buildings. These systems are further subdivided according to ground heat exchanger design: horizontal (spiral or flat coils) and vertical (closed–loops).

A. Horizontal Heat Exchangers

The heat pump takes the heat through a coil, which is placed in the ground from 0.5 to 2 m deep. At the level of the condenser, heat extracted from the ground is transmited through a heating circuite and/ or a circuit for domestic warm water.

Taking over the ground heat can be done either directly, when the coil in the ground, that is run by the refrigerant plays the rol of the evaporator, or indirectly by using an intermediary agent that can be a glycol water solution, which transmits the ground heat to the working agent at the level of the evaporator. The evaporator coil can be made either of cop-per, in the case of the direct systems, or of plastic materials in the case of indirect systems respectively.

Plastic tubes are placed, parallel to one another, 1.2 to 1.5 m deep, in the ground, and function of the chosen diameter of the tube at a distance of about 0.5…0.7 m, so that on each m^2 of rejection area 1.43…2.00 m of tube to be placed. The tube lengths must not exceed 100 m, because otherwise the pressure losses, and thus the heat pump power, could be too high. Heat that is transferred from the lower ground levels to the surface is only of 0.063…0.1 W/m^2 and can not be considered a heat source to be used. That is why the required ground area very much depends on the thermophysic ground properties and on the radiant energy, that is to say, on climate.

The accumulation capacity and thermal conductivity are greater function of the moisture content of the ground, the higher the quantity of minerals is, the lower the pore quantity. The values of the specific rejection power, q_E for ground are presented in Table I.

Table I. Specific rejection power for ground

No.	Type of ground	q_E [W/m^2]
1	Dry sandy ground	10…15
2	Moist sandy ground	15…20
3	Dry clay ground	20…25
4	Moist clay ground	25…30
5	Ground with underground water	30…35

The required ground area is determined function of the cooling (vaporization) power Q_0 of the heat pump, which is computed as the difference between the thermal power Q_{PC} of the heat pump and electric power absorbed P_A by the compressor:

$$Q_0 = Q_{PC} - P_A \quad (12)$$

At a specific rejection power q_E, there comes out the required ground area given by:

$$S_E = \frac{Q_0}{q_E} \quad (13)$$

B. Vertical ground loops

Because of the larger ground area requirement for the evaporator coil to be set, this is difficult to be done even for new buildings, because of the insufficient space. Especially in crowded halls, having small surfaces, space is limited. This is the reason why vertical ground loops are set, usualy of high–density polyethylene (HDPE) tubes that are introduced to a deapth of 50...150 m.

The vertical ground loop consists of a high–density polyethylene U–tube insered into vertical borehole (Fig. 2–a). With the U–tube in place, a grout mixture is pumped into the borehole to fill the space between the U–tube and the borehole wall. Low–permeability grout prevents water and contaminants from traveling along the vertical borehole.

The specific absorbtion power q_E, for ground loops are presented in Table II. For a specific absorbtion power q_E, that comes out the deapth (length) of the borehole where the loop is set:

$$H = \frac{Q_0}{q_E} \quad (14)$$

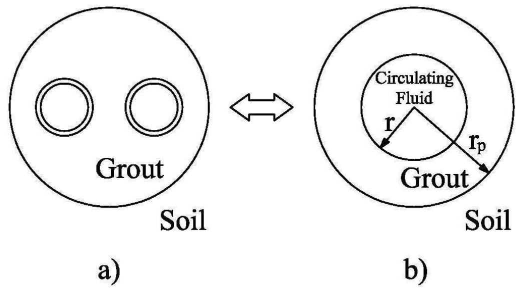


Fig. 2 Geometry of borehole
a) actual, b) calculation model

Table II. Specific absorbtion power for ground loops, q_E [W/m]

No.	Type of ground	Operating hours	
		1800 h/yr	2400 h/yr
1	Dry ground (λ<1.5 W/(m·K))	25	20
2	Ground of stable rocks and water filled sediments (λ<3 W/(m·K))	60	50
3	Rocks with high thermal conductivity (λ>3 W/(mK))	84	70
4	Gravel, dry sand	<25	<20
5	Gravel, acvifer sand	65...80	55...85
6	Gravel and sand with powerful underground water flux	80...100	80...100
7	Moist clay and/or loam	35...50	30...40
8	Chalkstone (massive)	55...70	45...60
9	Grit stone	65...80	55...65
10	Magmatic rock (granite)	65...85	55...70
11	Basaltic rock (basalt)	40...65	35...55
12	Gnais	70...85	60...70

The distance between two ground loops must be 4.5…6 m. Vertical loops are set, function of the model, with drilling equipments or with thrusting equipments by swinging. After the loop is properly set, the pressure and flowing tests are performed. Pressure test is performed at minimum 0.6 MPa and maximum 1 MPa. The duration of the test is 60 min and the accepted pressure drop is 0.02 MPa.

IV. Ground Thermal Responce Test

The incorrect determination of required vaporization thermal power of the vertical closed–loop ground–coupled heat pumps leads to unfavorable effects for these systems: under–sizing the catching system used for the vaporization of the agent determines the reduction of heat pump nominal thermal power; over–sizing catching system leads to additional investments that puts under discussion the opportunity of using such systems.

For a give heat taken from the air, water or ground (through horizontal heat exchangers), the parameters that determine the vaporization thermal power of the heat pump are easy to be measured, and the vaporization thermal power is easy to be solved. In the case of vertical closed–loop ground–coupled heat pump systems, the determination of the parameters to calculate vaporization thermal power that must be provided from ground is more laboriously to be done. The fact that ground–coupled heat pumps are the most widely spread (because of their fiability and efficency) determined the specialists in the domain to find the most efficient and precise methods to determine the parameters that leads to the most accurate calculation of the vaporization thermal power provided by the ground, the length of the ground loops respectively.

That is why it is very important to know the thermal resistance of a borehole heat exchanger in order to know how many loops must be set, function of the energy that must be given to the heat pump. If for the heat pumps with low thermal heating/cooling powers (< 30 kW) it is nor wrong to use estimated values of the thermal resistance of the ground loop, taking into account its caracteristics, as well as values of the soil thermal conductivity in the existing tables in technical literature, this thing is not possible for heat pumps with thermal heating/cooling powers greater than 30 kW. That is why the evaluation of the dimensioning parametres is absolutely necessary, as soil thermal conductivity and thermal resistance of the loop (borehole). In this respect it is necessary to make a ground thermal responce test, using a borehole for probation in which a simple ground loop is placed.

A. Physics Precepts of the Test

Thermal field surrounding loop is determined with line–source model, which represents the borehole as a line source of heat. The model ignores the details of the complicated geometry of the U–tube loop (Fig. 2–a) and the differences in thermal proprieties of the grout and soil. Instead, a borehole thermal resistance is used to represent the sum of all the thermal resistances inside the borehole between the circulating fluid and the soil.

Temperature rise surrounding loop is given by [26]:

$$\Delta T(r_p,\tau) = T_p - T_s = \frac{q_E}{4\cdot\pi\cdot\lambda}\cdot\int_{\frac{r^2}{4\cdot a\cdot\tau}}^{\infty}\frac{e^{-\beta^2}}{\beta}\,d\beta = \frac{q_E}{4\cdot\pi\cdot\lambda}\cdot E\left(\frac{r^2}{4\cdot a\cdot\tau}\right) \qquad (15)$$

in which: $\Delta T(r_p,\tau)$ is the difference of temperature around the loop in function of the borehole radius r_p and of the time τ; T_p – average temperature of borehole wall, in K; T_s – undisturbed soil temperature, in K; q_E – specific power of rejection/absorbtion, in W/m; λ – soil thermal conductivity, în W/(m·K); r – effective radius, in m; $a=\lambda/\rho c$ – thermal diffusivity of soil, in m^2/s; ρ – soil density, in kg/m^3; c – soil specific heat at constant pressure, in J/(kg·K); τ – time, in s.

Integral exponential E, for high values of the parameter ($a\tau/r^2$), may be approximated with the following relation:

$$E\left(\frac{r^2}{4\cdot a\cdot\tau}\right) = \ln\frac{4\cdot a\cdot\tau}{r^2} - \gamma \qquad (16)$$

with which the equation (15), for $\tau > 5r_p^2/a$ becomes:

$$\Delta T(r_p,\tau) = q_E\cdot R_s = \frac{q_E}{4\cdot\pi\cdot\lambda}\left(\ln\frac{4\cdot a\cdot\tau}{r_p^2} - \gamma\right) \qquad (17)$$

where: R_s is the thermal resistance of soil, in K/(W/m); γ – Euler constant approximately equal to 0.5772.

The temperature difference in borehole, that is between the average temperature of the circulating fluid within the tube $T_f=(T_i+T_e)/2$ and the borehole wall temperature T_p, is given by:

$$T_f - T_p = R_p\cdot q_E \qquad (18)$$

in which R_p is the borehole resistance, in K/(W/m).

Transforming the equation (17) of the thermal field, by introducing the borehole resistance $R_{p,}$ is obtained the equation of the temperature variation between the working fluid and the soil (Fig. 3):

$$\Delta T(r_b,\tau) = q_E\cdot(R_p + R_s) = q_E\left[R_p + \frac{1}{4\cdot\pi\cdot\lambda}\left(\ln\frac{4\cdot a\cdot\tau}{r_p^2} - \gamma\right)\right] \qquad (19)$$

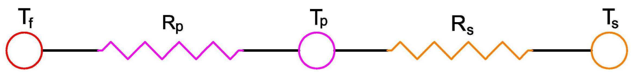
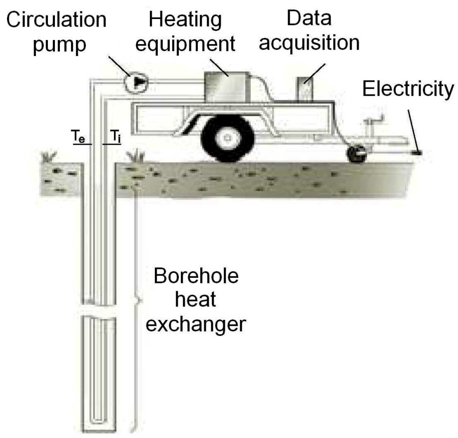


Fig. 3 Electric analogy of the heat exchange model between the loop and the ground

To get the smallest temperature differences within the borehole it is necessary that its thermal resistance to be as small as possible. This can be obtained by increasing the soil thermal conductivity, using adequate filling materials and/or by increasing the distance between the tubes of the vertical loop.

B. The Required Equipment for the Test

During an in–situ test, an above–ground electric heater usually provides heat to the fluid circulating (water or glycol water) through the ground loop, while the inlet (T_i) and outlet (T_e) fluid temperatures are measured (Fig. 4). The average of these two instantaneous temperature reading is usually taken to represent the average temperature in the vertical ground loop at a given time. In an ideal test, the measured circulating flow rate and the heat input rate remain constant throughout the test.



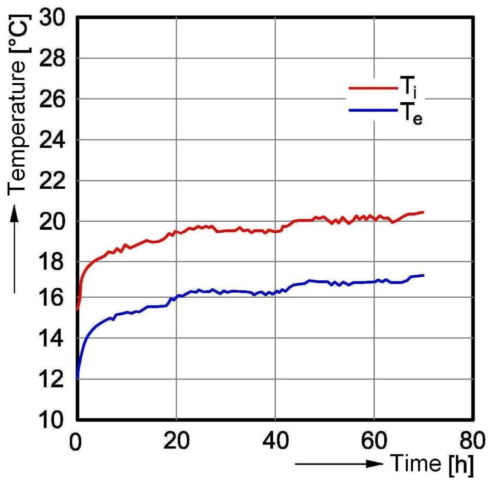


Fig. 4 Scheme of the equipment for thermal response test

Ground thermal responce tests, with this type of equipment, were done in Sweden, Norway, Germany, Swisserland, Canada and The United States. The first thermal responce test in Romania was performed by the company GEOTERM PDC from Bucharest [17], in July 2009. Figure 5 presents the testing equipment of this company.

Fig. 5 Testing equipment GEOTERM PDC

C. Data Analysis and Final Evaluation

While absorbtion/rejection a certain heat quantity in/from the ground a thermal transitory temperature is set up expressed by developing the equation (17), under the form:

$$T_f = \frac{Q}{4 \cdot \pi \cdot \lambda \cdot H} \ln(\tau) + \left[\frac{Q}{H} \left(\frac{1}{4 \cdot \pi \cdot \lambda} \left(\ln \frac{4 \cdot a}{r_p^2} - \gamma \right) + R_p \right) + T_s \right] \tag{20}$$

in which: Q is the rate of heat rejected to ground, in W; H – length or depth of borehole (vertical loop), in m.

Equation (20) can be simplified, by being written under liniar:

$$T_f = \alpha \cdot \ln(\tau) + n \tag{21}$$

where:

$$\alpha = \frac{Q}{4 \cdot \pi \cdot \lambda \cdot H} \tag{22}$$

$$n = \frac{Q}{H} \left(\frac{1}{4 \cdot \pi \cdot \lambda} \left(\ln \frac{4 \cdot a}{r_p^2} - \gamma \right) + R_p \right) + T_s \tag{23}$$

Soil thermal conductivity λ is obtained from relation (22) function of the late–time slope α, in a plot of the loop (circulating fluid) temperature versus the natural logarithm of time τ (Fig. 6):

$$\lambda = \frac{Q}{4 \cdot \pi \cdot \alpha \cdot H} \tag{24}$$

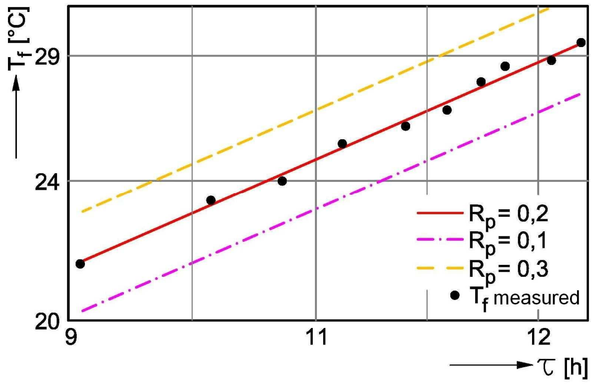


Fig. 6 Determination of borehole thermal resistance

The slope α of the interpolation straight–line of the measurements is independent from the borehole resistance R_p and, thus, allows for the determination of the real borehole resistance using the soil thermal conductivity λ estimated. Figure 7 shows the variation of the fluid average temperature T_f function of the time τ lapsed from the begining of the test.

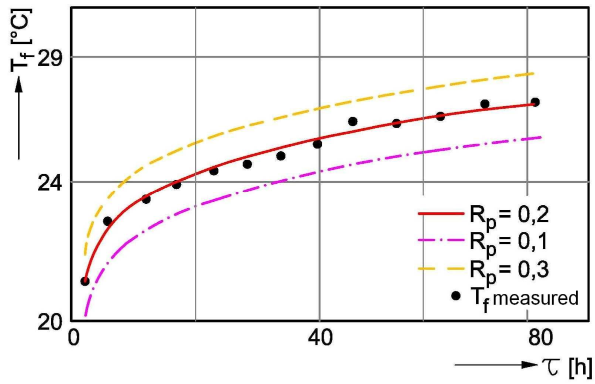


Fig. 7 Variation of fluid (loop) temperature in time

Replacing soil thermal conductivity, obtained in the relation (24), in the equation (17) it results the equivalent thermal resistance of the borehole:

$$R_p = \frac{1}{q_E}(T_f - T_p) = \frac{1}{q_E}(T_f - T_s) -$$

$$- \frac{1}{4 \cdot \pi \cdot \lambda} \left(\ln(\tau) + \ln \frac{4a}{r_p^2} - \gamma \right) \tag{25}$$

Equation (25) does not allow a proper evaluation of the equivalent thermal resistance of the well, this one being influenced of test duration through ln(τ).

In the same time, in equation (21) thermal diffusivity comes in a, as a ratio between thermal conductivity λ and volumetric thermal capacity C. While thermal conductivity λ is determined by the equation (24), the volumetric thermal capacity C, in $J/(m^3 \cdot K)$, can be only approximated with the relation:

$$C = \rho c = e^{\left[\ln\frac{4\cdot\tau\cdot\lambda}{r_p^2} - \gamma - \frac{4\cdot\pi\cdot\lambda}{q_E}\left(T_f - T_s - q_E \cdot R_p\right)\right]} \qquad (26)$$

where ρ, c are the density and specific heat of soil, and the borehole thermal resistance R_p is considered equal to 0.1 K/(W/m), for a standard borehole.

To determine the minimum duration $\tau_{\min}$ of test the following relation can be used:

$$\tau_{\min} = \frac{5r_p^2}{a} \qquad (27)$$

Austin et al [3] recommend a minimum duration of 50 hours based on their experiences with field data sets. Gehlin [9] suggests a minimum duration of 60 hours, but recommends using 72 hours. Smith and Perry [24] suggest that 12 to 20 hours may sometimes be sufficient, partly because if the test duration is too short, the estimated soil thermal conductivity is too low, which is a conservative estimate for the design of ground heat exchangers.

The graphic method described in the simplified analytical model of the line–source allows the determination of the soil thermal conductivity with an accuracy of 0.05 W/(m·K) and of the borehole thermal resistance with an accuracy of 0.005 K/(W/m). There are other more complex models to determine the thermal resistance of borehole as the analytical model of the cylindrical heat source or some numerical algorithms [3].

V. Equivalent Time for Interrupted Tests on Borehole Heat Exchangers

Electrical power outages, electric heater failures, or other unexpected events sometimes interrupot borehole tests before the test duration is sufficient to estimate soil thermal conductivity. One would like to restart the test immediately after the equipment problems are fixed, but the temperature distribution in the ground has been changed. Most analysis methods assume a spatially uniform ground temperature at the start of the test, and this assumption is invalid if the test is restarted quickly.

To return to the initially undisturbed ground conditions, Martin and Kavanaugh [16] recommend a 10 to 12 day waiting period before retesting a borehole after a completed 48 hour test . For an interrupted test, they suggest the waiting period can be reduced in proportion to the reduced test time [12]. Such time delays cost time and money when the equipment is on location and ready for restarting the test.

If the duration of the interruption is no more than a few hours, the best course of action may be to restart the test immediately after the problem is repaired, even with only standard analysis. Methods. As an example, Figure 8 illustrates temperature rise curves from both uninterrupted and interrupted tests.

During the interrupted test, electric power was shut off for a two–hour period, starting at approximately nine hours into the test.

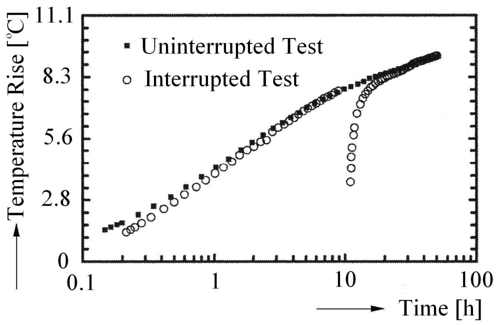


Fig. 8 Loop temperature curves

The loop temperature is greatly distorted immediately after startup, but the rising temperature eventually overlays on the uninterrupted curve with nearly the same late–time slope. The estimated thermal conductivity values from each test are within 2% of each other. In this case, the cumulative test time is 51 hours, including the interruption. Thus, restarting the test immediately after the power is restored is the best strategy in this case.

This paper applies the equivalent–time concept [1] to interrupted thermal tests on borehole heat exchangers. The method rescales the time coordinate on the horizontal axis in Figure 8 and removes the effects of the interruption on the temperature rise curve. The method shortens the required test time for estimating soil thermal conductivity.

A. Model for Interrupted Test

Consider the rate changes during an interrupted test, as shown in Figure 9, where the heat input rate is constant at the value of Q_1 between τ_1 and τ_2, but the electric power supply is interrpted, and the heat input rate suddenly goes to zero at time τ_1. Then, at time τ_2 the power is restored, and the heat input rate is restarted to Q_3, which may differ from the earlier value, Q_1. Supperposition may be used to take into account these rate changes and estimate the corresponing loop temperature.

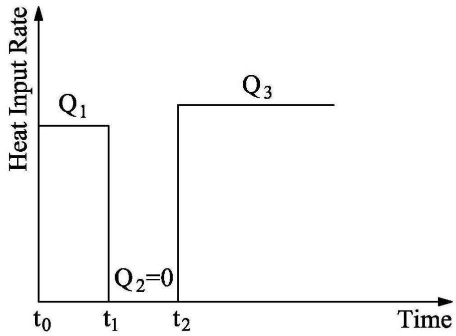


Fig. 9 Variations in heat input rate represented as discrete step changes during an interrupted test

One applies the constant–rate solution for each step rate change (Q_i–Q_{i-1}), which occurs at time τ_{i-1}. If the number of rate changes is given as n, the loop temperature is a sum of constant–rate responses [4]:

$$T_f(\tau) = \sum_{i=1}^{n} \frac{Q_i - Q_{i-1}}{Q_{ref}} T_u\left(\tau - \tau_{i-1}\right) \qquad (28)$$

in which: $\tau_{n-1} \le \tau \le \tau_n$, and $Q_0 = 0$ at τ_0. For the line–source model, T_u is set equal to T_f in equation (20), where Q is set equal to $(Q_i – Q_{i-1})$ for each step change. The reference heat input rate is set to the last input rate change, $Q_n – Q_{n-1}$. For the rate schedule in Figure 9, the reference heat input rate is $Q_3 – Q_2$.

B. Equivalent Time Method

Because the interruption in the heat input rate greatly distorts the temperature rise in Figure 8, there is a need for an alternative to the conventional line–source analysis. This is the motivation behind the equivalent time method. For a single interruption, equivalent time is given by [4]:

$$\Delta\tau_e = \left(\frac{\tau_2 + \Delta\tau}{\tau_2}\right)^{\frac{Q_1}{Q_3 - Q_2}} \left(\frac{\tau_2 - \tau_1 + \Delta\tau}{\tau_2 - \tau_1}\right)^{\frac{Q_2 - Q_1}{Q_3 - Q_2}} \Delta\tau \qquad (29)$$

in which: $\Delta\tau = \tau - \tau_2$ and $\tau_2 < \tau$. The temperature rise after the interruption may be expressed in terms of equivalent time as [4]:

$$T_f - T_2 = \frac{Q_3 - Q_2}{4\pi\lambda H}\left[\ln\left(\frac{4a\Delta\tau_e}{\gamma r_p^2}\right) + 4\pi\lambda R_p\right] \qquad (30)$$

Thus, equivalent time $\Delta\tau_e$ transforms the temperature rise in equation (30) into same mathematical formulation as the constant heat input rate case in equation (20). Equivalent time takes the place of the elapsed test time τ and T_2 takes the place of the undisturbed soil temperature T_s.

A comparison of equation (20) and (30) shows that the expression for soil thermal conductivity for the equivalent time method is a simple modification of equation (24):

$$\lambda = \frac{Q_3 - Q_2}{4\cdot\pi\cdot\alpha\cdot H} \qquad (31)$$

The slope α in equation (31) comes from a semilog graph with the natural logarithm of equivalent time on the horizontal axis.

Similary, the expression for borehole resistance based on equivalent time comes after some algebraic rearrangement of equation (30) to give:

$$R_p = \frac{1}{4\pi\lambda}\left[\frac{T_{f,1h} - T_2}{\alpha}\right] - \ln\left(\frac{4a\Delta\tau_{e,1h}}{\gamma r_p^2}\right) \qquad (32)$$

where the slope α is from a graph with the natural logarithm of equivalent time on the horizontal axis.

If the number of heat input rates in a field test is more than tree, the same techniques would apply. For the aplication with n heat input rates, the term $(Q_3 – Q_2)$ in equation (31) is replaced by $(Q_n – Q_{n–1})$ to estimate soil thermal conductivity. Likewise, the term T_2 is replaced by $T_{n–1}$ in equation (32) to estimate borehole resistance.

The equivalent time method differs from the practice of waiting for the temperature distribution in the ground to approach its undisturbed uniform temperature, which may require a long delay before restarting the test [16]. Such delays cost money if the equipment is repaired and ready for resumption of the test. The equivalent time method allows an analysis of test data if the test is restarted as soon as possible. The minimum additional time after restart is less than or equal to the minimum test duration for an uninterrupted test. Therefore, if one takes into account the interrupted heat input rates with equivalent time and restarts the test as soon possible, the additional test time is no more than for the case of stating a test with undisturbed soil temperature.

VI. Conclusions

In Europe, the ground thermal responce test became a standard instrument to investigate the necessary parameters for the proper designin of vertical loops.

Through the ground thermal responce test the length of the loops is properly determined, the operating performance of the system is provided, and suplimentary costs (extra loops, boreholes, glycol etc.) are avoided. This operation is performed using specialized software. From the cost efficency point of view, thermal test is generally efficient for those situations where 10 or more vertical loops are required.

Equivalent time provides a method to analyze interrupted tests by removing the effects of the interruption. This method has the potential of saving time and money if the test restarted immediately when electric power is restored.

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