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Heating Agent Pressure Losses in a Borehole Heat Exchanger

Andrzej Gonet, Tomasz Śliwa, Stanisław Stryczek

Faculty of Drilling, Oil and Gas, AGH University of Science and Technology, al. Mickiewicza 30, 30-059 Kraków, Poland

gonet@uci.agh.edu.pl, sliwa@uci.agh.edu.pl, stryczek@uci.agh.edu.pl

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ABSTRACT

One of the possible ways of exploiting thermal energy of rock is heat carrier circulation in a borehole heat exchanger. This method lies in the use of an already existing well, where a heat circulation enhancement system is installed. Heat is exchanged through the mechanism of convection and conduction with the rock mass (Rybach, 1998).

The most important parameter of a heat exchanger is the accessible heating or cooling power when the system operates as a heat storage. Another important factor is the flow pressures in the heat carrier circulation system. It has an effect on the unit cost of heat production or storage.

The methodology of determining heat carrier pressure losses during circulation is presented in the paper. Design characteristics of the heat exchanger were analyzed; variability of temperature was assessed through analysis of changes of density and viscosity of the energy carrier in the well – inside the insulating column and in the annular space.

INTRODUCTION

Heat exchangers are used for heat collecting or storing in the rock mass. In the first process, they are a source of low-temperature energy, which through the heat pumps is transmitted for heating purposes (Śliwa and Gonet, 2003).

Heat exchangers differ in their design. For borehole heat exchangers, there must be a borehole, the wall of which constitutes a heat-exchange surface between the circulating fluid and the rock mass. It is also a channel for heat energy transportation from the rock mass and the surface (Gonet and Śliwa, 2002).

Heat exchangers vary, depending on the depth of the borehole and whether the borehole was adapted or made especially for the heat exchanger purposes. As the BHE can have different designs, the heat carrier flow geometry will vary as well.

Fig. 1 shows the cross sections of BHE used in practice. The systems presented in figs. 1 b, c and d are applied for boreholes at small depths and in the case of boreholes made especially for heat exchanger purposes. The system depicted in fig. 1 a can be used for deeper and already existing boreholes. This is especially important that the casing of the existing well can be adapted for heat exchange purposes.

Circulation parameters are strongly influenced by the type of heat carrier used. The most frequently applied heat carrier is a liquid. This can be water or a solution of ethylene or propylene glycol (Kavanaugh and Rafferty, 1997).

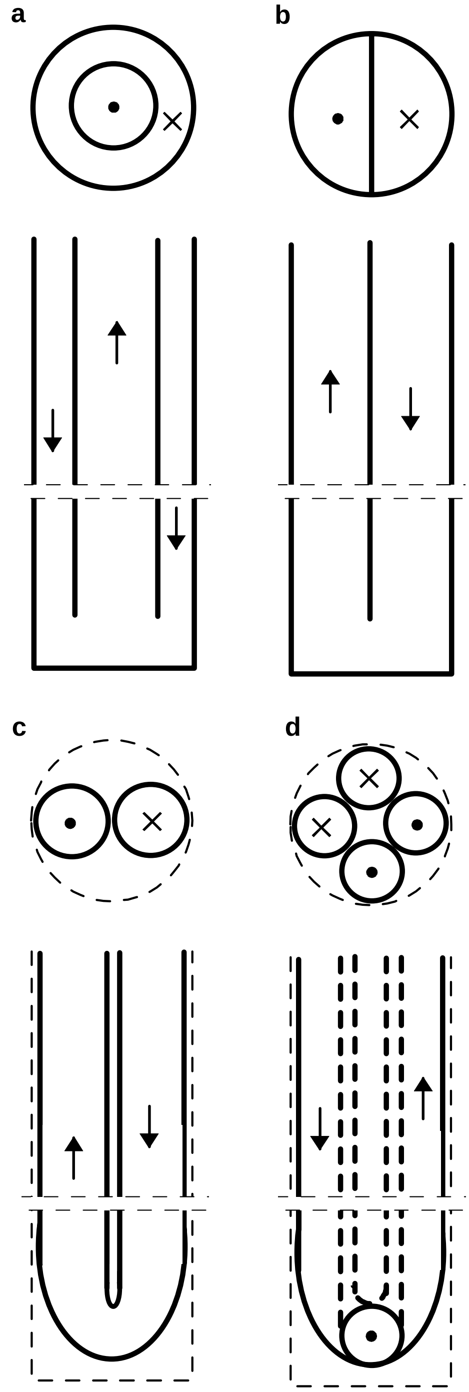

Figure 1: BHE designs, depending on the heat energy carrier flow geometry, a – co-axial system, b – half-system, c – U-tube, d – double U-tube.

The most important parameter of BHE is the obtained heating power (Śliwa and Kotyza 2003). To produce heat from BHE, a portion of energy has to be delivered to enhance the heat carrier circulation. Knowing the amount of this energy, one can balance the obtained heat and drive energy for enhancing heat carrier circulation. By taking these parameters into account and the quantity of energy needed to increase the state of heat energy to the required characteristic of the heating system, makes it possible to more precisely determine the unit cost of useful heat production.

ASSUMPTIONS

Geometry of heat carrier flow in a borehole presented in fig. 1a has been assumed for the considerations. This can be justified by a considerable number of boreholes in Poland (the Carpathians in particular) to be closed. This is caused the depletion of oil and natural gas resources in old Carpathian fields, exploited by tens of wells. Over 100 wells are closed in the flysch Carpathians area each year. A great number of them are in the urban areas, which creates the possibility to use the boreholes as heat sources instead of being closed down. The lack of warm formation waters flux to such boreholes causes that BHE are the only option for geothermal use.

The depth of the boreholes sometimes exceeds 1000 m, and the design presented in fig. 1a enables tripping of the internal column even to such a depth. The applicability of BHE to such a depth is conditioned by thermal insulation of the material, out of which the internal column has been made, and the heat carrier pressure losses.

It has been assumed in the methodic that a Newtonian fluid will be injected to the annular space. Flowing towards the bottom it will be heated by the rock mass. Then, heated, it will run to the internal column, going up towards the surface. The heat carrier temperature distribution will be the following for this operation mode (fig. 2).

Taking a simplifying assumption that the distribution of heat carrier is linear with the depth both in the annular space and inside the internal column, the following can be written:

$$T_a(h) = T_1 + (T_2 - T_1)\cdot\frac{h}{H} \tag{1}$$

and

$$T_i(h) = T_3 + (T_2 - T_3)\cdot\frac{h}{H} \tag{2}$$

The assumption of a linear geothermal gradient limits the applicability of the model to a borehole heat exchanger, where overburden formations do have equal thermal conductivities and do not have convections that would dominate part of the profiles.

Having assumed that a heat carrier is a Newtonian fluid in the form of pure water, we know that its density and viscosity change with temperature. In the Polish conditions, the heat carrier temperature during a long period of BHE exploitation will not exceed 20^{o}C (Śliwa, 2002). Fig. 3 illustrates the dependence of density, whereas fig. 4 shows the dependence of dynamic viscosity of water in a function of its temperature.

To further analyze the water density value with the changing temperature, determined empirically at correlation coefficient 0.999 for temperature ranging from 0 to 40^{o}C, the following assumption was made on the basis of data in (Bigg, 1967):

$$\rho(T) = 1000\cdot e^{\frac{(T-4)^2}{-147686}} \tag{3}$$

and dynamic viscosity as a function of temperature, according to the Arrhenius and Guzman relation:

$$\eta(T) = \eta_0\cdot e^{\frac{E}{R\cdot T}} \tag{4}$$

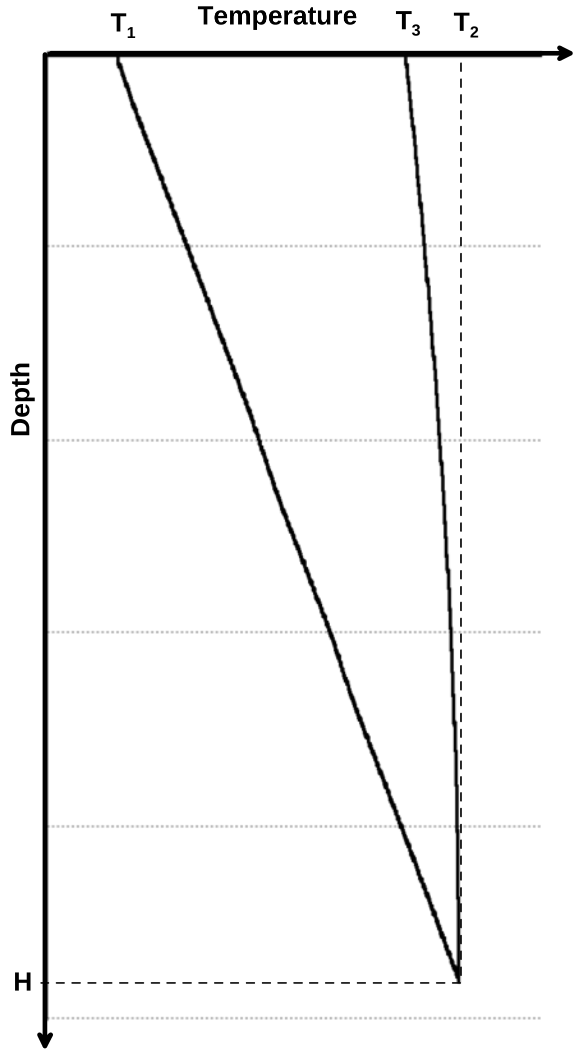

Figure 2: Temperature distribution during heating agent circulation in BHE presented in figure 1a.

Having assumed changeability of temperature in the BHE profile, the variability of density and dynamic viscosity can depend on depth. Then, two dependences are obtained for the inner column and two for the annular space:

$$\rho_i[T_i(h)] = \rho_i\left[T_3 + (T_2 - T_3)\cdot\frac{h}{H}\right] = 1000\cdot e^{\frac{\left\{\left[T_3+(T_2-T_3)\cdot\frac{h}{H}\right]-4\right\}^2}{-147686}} \tag{5}$$

$$\rho_a[T_a(h)] = \rho_a\left[T_1 + (T_2 - T_1)\cdot\frac{h}{H}\right] = 1000\cdot e^{\frac{\left\{\left[T_1+(T_2-T_1)\cdot\frac{h}{H}\right]-4\right\}^2}{-147686}} \tag{6}$$

and

$$\eta_i[T_i(h)] = \eta_i\left[T_3 + (T_2 - T_3)\cdot\frac{h}{H}\right] = \eta_0 \cdot e^{\frac{E}{R\cdot\left[T_3+(T_2-T_3)\cdot\frac{h}{H}\right]}} \quad (7)$$

$$\eta_a[T_a(h)] = \eta_a\left[T_1 + (T_2 - T_1)\cdot\frac{h}{H}\right] = \eta_0 \cdot e^{\frac{E}{R\cdot\left[T_1+(T_2-T_1)\cdot\frac{h}{H}\right]}} \quad (8)$$

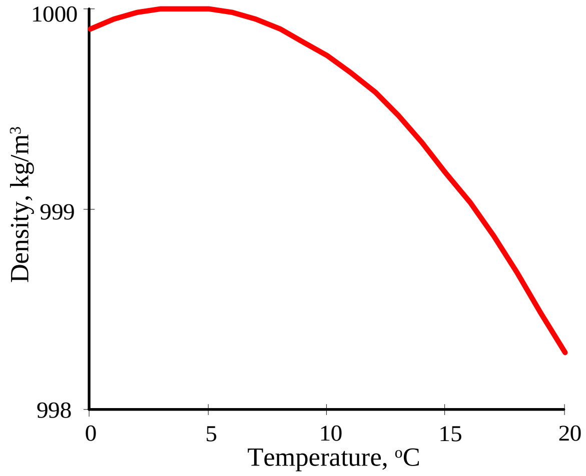

Figure 3: Dependence of water density on temperature.

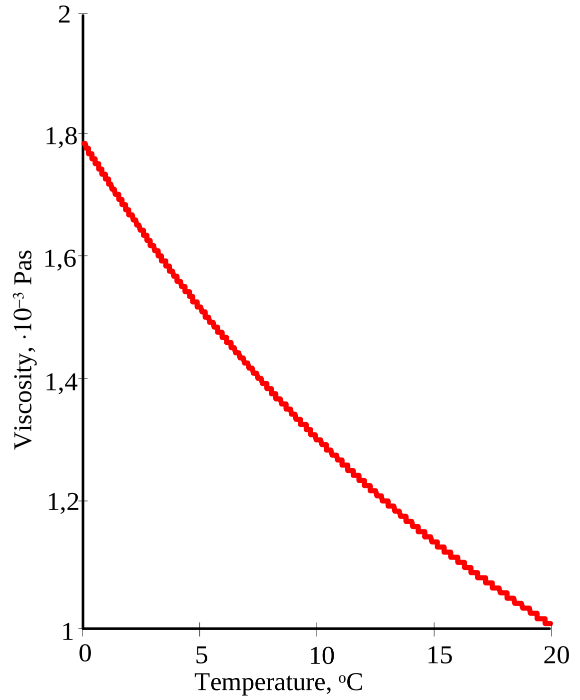

Figure 4: Dependence of water dynamic viscosity on temperature.

PRESSURE LOSSES

In view of the above assumptions, pressure losses of a heating agent in internal column of BHE can be determined from the Darcy-Weisbach formula, like pressure losses of a mud during drilling (Chilligarian and Voraburt, 1981):

$$p_i = \frac{8\cdot\lambda\cdot\rho\cdot H\cdot Q^2}{\pi^2\cdot d_i^5} \quad (9)$$

For a laminar flow (Re<2320) inside pipes, the friction losses coefficient is:

$$\lambda = \frac{64}{\mathrm{Re}} = \frac{64\cdot\eta}{v\cdot d_i\cdot\rho} = \frac{16\cdot\pi\cdot\eta\cdot d_i}{\rho\cdot Q} \quad (10)$$

Substituting (10) to (9) and simplifying it, the following is obtained:

$$p_i = \frac{128\cdot\eta\cdot H\cdot Q}{\pi\cdot d_i^4} \quad (11)$$

Assuming that viscosity changes with fluid temperature, and taking into account (7) we get:

$$p_i = \frac{128\cdot Q}{\pi\cdot d_i^4}\cdot\int_0^H \eta_i(h)\,dh \quad (12)$$

For a turbulent flow (2320≥Re>100000) in pipes the friction loss coefficient is:

$$\lambda = \frac{0{,}316}{\mathrm{Re}^{0,25}} = \frac{0{,}2975\cdot d_i^{0,25}\cdot\eta^{0,25}}{\rho^{0,25}\cdot Q^{0,25}} \quad (13)$$

After substituting (13) to (9) and simplifying the expression, the following is obtained:

$$p_i = 0{,}241\cdot\frac{\rho^{0,75}\cdot\eta^{0,25}\cdot H\cdot Q^{1,75}}{d_i^{4,75}} \quad (14)$$

Owing to the fact that flow pressure losses in such conditions are relatively high, flows in borehole heat exchangers have been ignored for the Reynolds number over 100,000. Consequently, the effectiveness of BHE as a source of thermal energy is lowered.

Having assumed that viscosity and density change with fluid temperature, and taking into account (5) and (7), the following can be written:

$$p_i = \frac{0{,}241\cdot H\cdot Q^{1,75}}{d_i^{4,75}}\cdot\int_0^H \rho_i^{0,75}(h)\cdot\eta_i^{0,25}(h)\,dh \quad (15)$$

For annular space the below formula holds true:

$$p_a = \frac{8\cdot\lambda\cdot\rho\cdot H\cdot Q^2}{\pi^2\cdot(D-d_o)^3\cdot(D+d_o)^2} \quad (16)$$

For a laminar flow (Re<2320) in an annular space, the friction loss coefficient is:

$$\lambda = \frac{64}{\mathrm{Re}} = \frac{16\cdot\pi\cdot\eta\cdot(D+d_o)}{\rho\cdot Q} \quad (17)$$

After substituting (17) to (16) and simplifying it, the following is obtained:

$$p_a = \frac{128\cdot\eta\cdot H\cdot Q}{\pi\cdot(D-d_o)^3\cdot(D+d_o)} \quad (18)$$

Taking into account that viscosity changes with fluid temperature, and dependence (8), we have:

$$p_a = \frac{128 \cdot Q}{\pi \cdot (D - d_o)^3 \cdot (D + d_o)} \cdot \int_0^H \eta_a(h)\, dh \tag{19}$$

For a turbulent flow (Re≥2320) in the annulus, the friction loss factor is:

$$\lambda = \frac{0{,}316}{\mathrm{Re}^{0{,}25}} = \frac{0{,}2975 \cdot (D + d_o)^{0{,}25} \cdot \eta^{0{,}25}}{\rho^{0{,}25} \cdot Q^{0{,}25}} \tag{20}$$

Substituting (20) to (16) and simplifying the formula the following is obtained:

$$p_a = 0{,}241 \cdot \frac{\rho^{0{,}75} \cdot \eta^{0{,}25} \cdot H \cdot Q^{1{,}75}}{(D - d_o)^3 \cdot (D - d_o)^{1{,}75}} \tag{21}$$

Assuming that viscosity and density change with fluid temperature, and taking into account (6) and (8), formula (21) can have the below:

$$p_a = 0{,}241 \cdot \frac{H \cdot Q^{1{,}75}}{(D - d_o)^3 \cdot (D - d_o)^{1{,}75}} \cdot \int_0^H \rho_a{}^{0{,}75}(h) \cdot \eta_a{}^{0{,}25}(h)\, dh \tag{22}$$

After taking into consideration (7) and (8), formulae (12) and (19) will take the following form:

$$p_i = \frac{128 \cdot Q}{\pi \cdot d_i^4} \cdot \eta_0 \cdot \int_0^H \exp\left\{ \frac{E}{R \cdot \left[T_3 + (T_2 - T_3) \cdot \frac{h}{H} \right]} \right\} dh \tag{23}$$

and

$$p_a = \frac{128 \cdot Q}{\pi \cdot (D - d_o)^3 \cdot (D + d_o)} \cdot \eta_0 \cdot \int_0^H \exp\left\{ \frac{E}{R \cdot \left[T_1 + (T_2 - T_1) \cdot \frac{h}{H} \right]} \right\} dh \tag{24}$$

Similarly, having additionally taken into account formulae (5) and (6), the dependences (15) and (22) will take the form:

$$p_i = 241 \cdot \frac{\eta_0 \cdot H \cdot Q^{1{,}75}}{d_i^{4{,}75}} \cdot \int_0^H \exp\left\{ \frac{\left[\left[T_3 + (T_2 - T_3) \cdot \frac{h}{H} \right] - 4 \right]^2}{-196915} + \frac{E}{4 \cdot R \cdot \left[T_3 + (T_2 - T_3) \cdot \frac{h}{H} \right]} \right\} dh \tag{25}$$

and

$$p_a = 241 \cdot \frac{\eta_0 \cdot H \cdot Q^{1{,}75}}{(D - d_o)^3 \cdot (D - d_o)^{1{,}75}} \cdot \int_0^H \exp\left\{ \frac{\left[\left[T_1 + (T_2 - T_1) \cdot \frac{h}{H} \right] - 4 \right]^2}{-196915} + \frac{E}{4 \cdot R \cdot \left[T_1 + (T_2 - T_1) \cdot \frac{h}{H} \right]} \right\} dh \tag{26}$$

An additional element influencing the increase of hydraulic friction when the energy carrier is flowing, are local losses. These are mainly pressure losses in the annular space, related with the use of centralizers of the insulating column, which can play the role of load. Depending on the volume of heat carrier, number and shape of centralizers, as well as geometry of flow, they may increase the total hydraulic losses in the annular space by ten or so percent (Knez and Śliwa, 2001). Moreover, the change of flow on the BHE bottom also results in an increase of heat carrier pressure losses.

By accounting for these factors, it is possible to correct hydraulic pressure losses in the annular space. General formulae for linear pressure losses in the annular space for a laminar flow (from eq. 24), can be written:

$$p_l = K_l \cdot Q^{m_l} \tag{27}$$

where

$$K_l = \frac{128 \cdot \eta_0}{\pi \cdot (D - d_o)^3 \cdot (D + d_o)} \cdot \int_0^H \exp\left\{ \frac{E}{R \cdot \left[T_1 + (T_2 - T_1) \cdot \frac{h}{H} \right]} \right\} dh \tag{28}$$

whereas, for a turbulent flow (from eq. 26) in the form:

$$p_t = K_t \cdot Q^{m_t} \tag{29}$$

where

$$K_t = \frac{241 \cdot \eta_0 \cdot H \cdot Q^{1{,}75}}{(D - d_o)^3 \cdot (D - d_o)^{1{,}75}} \cdot \int_0^H \exp\left\{ \frac{\left[\left[T_3 + (T_2 - T_3) \cdot \frac{h}{H} \right] - 4 \right]^2}{-196915} + \frac{E}{4 \cdot R \cdot \left[T_3 + (T_2 - T_3) \cdot \frac{h}{H} \right]} \right\} dh \tag{30}$$

The correction coefficient value *m* (Miska, 1979) can be determined through an experimental circulation of an heat carrier for two different volumes of its flow. The total pressure losses of the energy carrier should be measured and flow losses inside the insulating column should be calculated from the formulae (23) or (25), respectively. Then the correction coefficient value is calculated from (Gonet et al., 1988):

$$m = \frac{\lg \dfrac{p_{c2} - p_{i2}}{p_{c1} - p_{i1}}}{\lg \dfrac{Q_2}{Q_1}} \tag{31}$$

The value of *m* should be adjusted to the regime of flow in the annular space as m_l or m_t , depending on the Reynolds' criterial number.

Total pressure losses in the system are a sum of losses in the individual elements of its circulation, i.e.:

$$p_t = p_a + p_i + p_m \tag{32}$$

Power needed for enhancing heat carrier circulation can be described with the below formula:

$$P_c = p_t \cdot Q \cdot \eta_p \tag{33}$$

Knowing that the quantity of heating power imparted by BHE is described by the formula:

$$P_t = Q \cdot c \cdot \rho \cdot (T_3 - T_1) \tag{34}$$

the power effectiveness of BHE can be written in the form of an power balance:

$$\Delta P = P_t - P_c = Q \cdot [c \cdot \rho \cdot (T_3 - T_1) - p_t \cdot \eta_p] \tag{35}$$

Based on the formulae for hydraulic losses and temperatures obtained from BHE, construction and exploitation parameters of this source of heat can be optimized. Function of target can describe the above-mentioned energy or economic effectiveness. At present, environmental issues are in the focus, therefore ecological effectiveness may be of greater importance. However, it is hard to find a clear-cut formula defining ecological effectiveness and the quantitative representation in the economic effectiveness.

DEEP BHE IN POLAND

The turn of the 20th and 21st century is a time of a decline of oil exploitation in the south-east region of the Polish Carpathians. The exploitation of some wells is not effective for some time. Therefore, since 1991 the Polish Oil and Gas Company started to consequently close the wells. There are over 100 wells closed each year. The existing wells can be re-used which can result in a reduction of capital costs of geothermal energy production (mostly drilling costs). Closing of wells is very expensive, and the cost of closing may exceed the cost of adaptation of a well with the surface installment for the exchange and reception of heat (Śliwa, 2002).

Adaptation of wells for the needs of geothermal heat recuperation is very attractive, especially in the view of the fact that a great deal of them are sited in highly urbanized areas. Both formations and wells frequently coincide with developed city areas. This creates a possibility to manage the heat without constructing expensive large surface installments. One of the possible options of producing heat from the rock mass is the utilization of wells assigned for closing as BHEs.

One of the depleting oil and natural gas fields is the Iwonicz Zdrój fields. It was discovered in 1890. By the year 2000, 97.5% of natural gas and 97.74% of oil have been depleted. Production was carried out in 41 wells.

Elin 3 well was one of the wells that could be adapted to BHE. Thermal properties of the rocks near the Elin 3 well are presented in the paper (Śliwa and Gonet, 2003).

The internal diameter of BHE in the Elin 3 well will be 118 mm. The internal casing of an external diameter 90 mm and wall thickness of 25 mm can be tripped to a depth of 465 m. Fig. 5 shows the temperature profiles of the heat carrier after 1000 days of uninterrupted water circulation of 5 m^3/h and injection temperature 2 and 4^oC.

Table 1 lists temperature values in characteristic points of heat carrier circulation in BHE, based on Elin 3 well (Śliwa, 2002)

Table 1. Temperature values in characteristic points of heat carrier circulation in BHE.

Heat carrier temperature at the inlet to annulus	T_1	2,0 °C	4,0 °C
Heat carrier temperature on the BHE bottom	T_2	5,0 °C	6,6 °C
Heat carrier temperature at the outlet from the insulation column	T_3	4,8 °C	6,4 °C

Table 2 lists Re values and flow pressure losses in individual BHE spaces, determined on the basis of a relation describing hydraulic losses with and without accounting for the temperature profile of the heat carrier.

To determine the hydraulic power, the flow pressure loss values were increased by 10%, to cover the local hydraulic losses.

Re values were calculated for average ρ and η values, calculated from equations (3) and (4) for given temperatures in characteristic points of BHE from Table 1. The efficiency of the circulation pump was assumed to be 0.8.

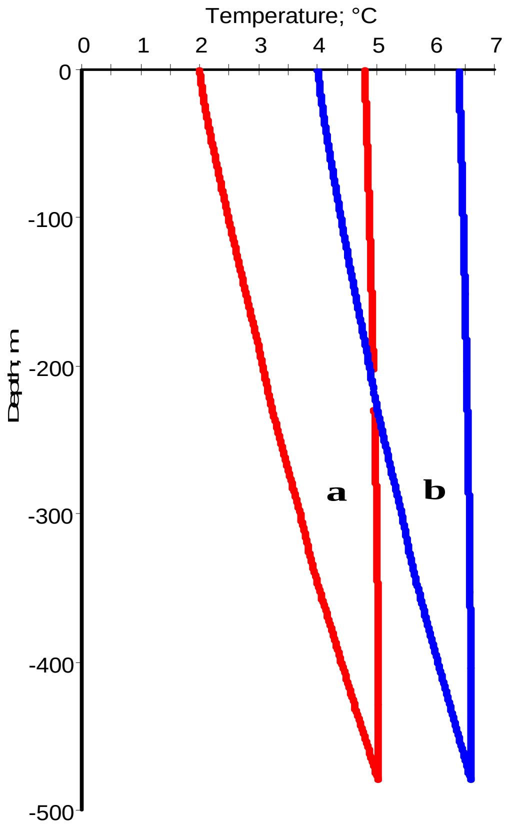

Fig. 5. Temperature profiles in BHE Elin 3 after 1000 days of heat production at the heat carrier flow rate 5 $m^3 \cdot h^{-1}$ and injection temperature of the heat carrier: a) 2^{o}C, b) 4^{o}C.

The efficiency of the system was defined as a ratio of useful heat power to operational power, i.e. power for driving heat pump compressors and circulation pump:

$$\varepsilon_{BHE} = \frac{P_u}{P_{hp} + P_c} \tag{36}$$

what can be also described as the function of heat pump`s coefficient of performance, heating power extracted from BHE and power for heating currier circulation:

$$\varepsilon_{BHE} = \frac{\varepsilon \cdot P_t}{\Delta P + \varepsilon \cdot P_c} \tag{37}$$

where

$$P_u = P_t + P_{hp} \tag{38}$$

and

$$\varepsilon = \frac{P_u}{P_{hp}} \tag{39}$$

It should be emphasized that there are also flow pressure losses related to the heat carrier transport from the well to the heat pump site and back, p_m. They additionally result in lowering of the total effectiveness ε_{BHE}, which lowers with the increasing distance between BHE and the customer. Surface pipelines of larger diameters increases the inwestment costs.

Table 2. Hydraulic parameters of heat carrier flow in BHE.

Parameter		Without accounting for heat carrier temperature profile		Accounting for heat carrier temperature profile	
		T_1=2 °C	T_1=4 °C	T_1=2 °C	T_1=4 °C
Re number in annular space	-	40497	42728	40498	42728
Re number inside the inner pine	-	28348	29909	28348	29909
Flow pressure losses in annulus, p_a	Pa	28088	27713	28239	27838
Flow pressure losses inside casing, p_i	Pa	179682	177284	178634	176427
Total flow pressure losses, p_c	Pa	207769	204997	206872	204265
Circulation pump power, P_c	W	361	356	359	355
Heat energy stream with BHE, P_t	W	16333	13999	16333	13999
Power effectiveness of BHE, ΔP	W	15972	13643	15974	13644
Useful heating power for ε=3, P_u	W	24500	20999	24500	20999
Effectiveness of the system, ε_{BHE}	-	2.873	2.855	2.874	2.855

Specific heat value for a heat carrier c=4200 $J \cdot kg^{-1} \cdot K^{-1}$.

CONCLUSIONS

1. Unit cost of energy production with BHE depends to a certain degree on costs related with the cost of energy needed for enhancing heat carrier circulation.

2. Energy for heat agent circulation enhancement depends on total hydraulic pressure losses in the closed circulation, consisting of BHE and the surface system.

3. In the borehole heat exchanger, the energy carrier temperature changes on the path of flow. Having assumed the variability of dynamic viscosity and density of energy carrier, total pressure losses can be determined more precisely. Energy carrier laminar pressure losses in the insulating column should be calculated from the formula (23), whereas for turbulent flow from (25).

4. The centralizers in the annular space of BHE cause an additional disturbance of flow and additional pressure losses. Owing to the different shapes of the centralizers and change of direction of heat energy flow in the BHE bottom area, the introduction of a correction coefficient *m* is the most favourable method for accounting these influences. Its value can be determined from the formula (31), after making empirical measurements of the object.

5. The selection of volume of energy carrier flow during circulation in the whole system should take into account both the analysis of condition of heat exchange with the rock mass and also analysis of pressure losses.

NOMENCLATURE:

c – specific heat, $J \cdot kg^{-1} \cdot K^{-1}$;
d_i – internal diameter of inner pipe, m;
d_o – external diameter of inner pipe, m;
h – depth, m;
m – correction coefficient, -;
p_{c1} – pressure losses in BHE at volume Q_1, Pa;
p_{c2} – pressure losses in BHE at volume Q_2, Pa;
p_i – pressure losses in internal column of BHE, Pa;
p_{i1} – pressure losses in internal column of BHE at volume Q_1, Pa;
p_{i2} – pressure losses in internal column of BHE at volume Q_2, Pa;
p_a – pressure losses in annular space of BHE, Pa;
p_m – pressure losses in surface circulation system of BHE, Pa;
p_t – total pressure losses in the system, Pa;
v – average flow rate, $m \cdot s^{-1}$;
D – internal well diameter, m;
E – activation energy for viscous flow, $J \cdot mol^{-1}$;
H – total depth of BHE, m;
P_c – power for heat carrier circulation in BHE, W;
P_t – heating power of BHE, W;
P_u – useful heat power of the system (heat pump), W;
P_{hp} – drive power of heat pump, W;
Q – flow rate of heating agent in BHE, $m^3 \cdot s^{-1}$;
R – universal gas constant, 8314 $J \cdot K^{-1} \cdot mol^{-1}$;
T – temperature, K;
T_1 – heat carrier temperature at the inlet to annulus, °C;
T_2 – heat carrier temperature on the BHE bottom, °C;
T_3 – heat carrier temperature at the outlet from the insulation column, °C;
T_i – heat carrier temperature in the insulating column, °C;
T_a – heat carrier temperature in annular space, °C.
ε - coefficient of performance (COP) of heat pump, -;
η – viscosity, Pas;
η_0 – constant for a given fluid, Pas;
η_p – efficiency of circulation pump, - ;
λ - friction losses coefficient, -;
ρ – density, $kg \cdot m^{-3}$.

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REFERENCES

Bigg P.H.: Density of water in S.I. units over the range 0-40 C, *Brit. J. Appl. Phys.*, **18**, (1967), 521-537.

Chilligarian G.V., Voraburt P.: *Drilling and Drilling Fluids*, New York Elsevier Scientific Publishing Company, Amsterdam, Oxford (1981).

Gonet A., Rzyczniak M., Stryczek S.: *Zadania do ćwiczeń z wiertnictwa (Exercises for classes in drilling)*, Wydawnictwa AGH, Kraków (1988) (in polish).

Gonet A., Śliwa T.: The utilization of boreholes for geothermal heat exploitation, New Methods in Drilling, Production, Transport and Underground Storage, *Conference Contributions*, XI International Scientific and Technical Conference, Technical University in Kosice, Podbanské, Slovakia (2002).

Kavanaugh S.P., Rafferty K.: *Ground-Source Heat Pumps, Design of Geothermal Systems for Commercial and Institutional Buildings*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (1997).

Knez D., Śliwa T.: Wpływ centralizatorów na hydrauliczne opory przepływu czynnika grzewczego w otworowym wymienniku ciepła (The influence of centralizers on hydraulic resistivities of a heat carrier in a borehhole heat exchanger)**,** *Zeszyty Naukowe AGH Wiertnictwo, Nafta, Gaz*, **18/1**, (2001), 121-126 (in polish).

Miska S.: *Drilling Optimization*, The University of Trondheim (1979).

Rybach L.: Market Penetration of BHE Coupled Heat Pumps – the Swiss Success Story, *Geothermal Resources Council Trans.*, **22**, (1998).

Śliwa T., Gonet A.: The idea of utilizing old production wells for borehole heat exchangers in the near depleted oil field in Iwonicz Zdrój, *Proceedings*, International Geothermal Conference: Multiple Integrated Uses of Geothermal Resources, Reykjavik, Iceland (2003).

Śliwa T., Kotyza J.: Application of existing wells as ground heat source for heat pumps in Poland, *Applied Energy*, **74**, (2003), 3-8.

Śliwa T.: *The Technical and Economic Issues of Exploited Wells Adaptation for Borehole Heat Exchangers*, doctorate thesis, Drilling, Oil and Gas Faculty, AGH University of Science and Technology, Kraków, (2002) (in polish).